Abstract—In recent years there has been an increased need for testing antennas and radar systems at high power. Since absorbers work by transforming electromagnetic energy into thermal energy there is a danger that in the presence of high fields the absorber will reach temperatures that will cause it to ignite. In the present paper standard polyurethane absorber is illuminated by a conical horn antenna. Different field levels are used to illuminate the sample. The internal and surface temperature of the sample is measured. From these measurements a behavior of the temperature versus field can be determined. Absorbers with and without a rubberized surface coating are also investigated to study their thermal behavior. This becomes very useful in determining if higher power materials may be required for testing active arrays or radar systems. The effects of lowering the temperature of the absorber by using airflow across the tips are also studied.

I. INTRODUCTION

In the heating process of microwave absorbers under incident electromagnetic waves, two disciplines of physics are intertwined, i.e., electromagnetic waves behavior governed by Maxwell’s equations and heat transfer process dictated by laws of thermodynamics. The power density in the absorbers due to the electromagnetic field is given by

\[ p = \sigma |E|^2 = 2\pi \epsilon_0 \epsilon'' f |E|^2 \]  

(1)

where, \( E \) is the total electric field (V/m) in the material, \( \sigma \) is electrical conductivity of the material (S/m), \( \epsilon_0 \) is the free space permittivity \( (8.854 \times 10^{-12} \text{ F/m}) \), \( \epsilon'' \) is the imaginary part of the relative dielectric constant, and \( f \) is the frequency in Hz. This is point form of the Joule’s law, and is well understood by RF engineers. The EM behavior of the polyurethane absorbers can be numerically computed. The EM field acts as the heating source, and its distribution in the absorber can provide a good indication on the locations of hot spots.

Polyurethane foam is an excellent insulator, so the conductive heat loss may be minimal. The heat exchanges can be reasonably described by radiation and convection transfers. Radiation takes place in the form of EM wave, mainly in the infrared region. The net power transferred from a body to the surroundings is described by Stefan-Boltzmann’s law [1],

\[ p_{rad} = \epsilon \sigma A (T^4 - T_0^4) \]  

(2)

where \( A \) is the surface area, \( T \) is the surface temperature of the radiation body in K, and \( T_0 \) is the ambient temperature in K. Unfortunately, the conventional symbols used in heat transfer \( \sigma \) and \( \epsilon \) are not the same as those in Eq. (1). \( \sigma \) here is the emissivity or emission coefficient, and is defined as the ratio of the actual radiation emitted and the radiation that would be from a black body. \( \epsilon \) in Eq. (2) is the Stefan-Boltzmann constant \( (5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4) \). The context in the paper should make it clear which symbols the authors are referring to. Otherwise, we will make explicit references.

The convective heat transfer is due to the motion of air surrounding the absorbers. Two forms can take place, naturally or by forced air. The relationship is described by Newton’s law of cooling [1]:

\[ p_{conv} = h A (T - T_0) \]  

(3)

where \( h \) is the convection heat transfer coefficient in \( \text{W/m}^2\text{K}^{-1} \). \( h \) is often treated as a constant, although it can be a function of the temperature. Eq. (3) assumes that the ambient air is abundant, and is taken to be constant. This is a reasonable assumption, because the heating is typically confined to a small localized area in a relatively large anechoic chamber. Combining the two mechanisms of heat transfer, the total heat loss is given by

\[ p = \epsilon \sigma A (T^4 - T_0^4) + h A (T - T_0) \]  

(4)

It is possible to solve for the temperatures from coupled Maxwell’s and heat transfer equations. Realistic results require accurate electrical and thermal properties of the materials. It is often a non-trivial process to obtain the material properties in and of itself. Careful validation is warranted before we can have full confidence in the results. In this paper, we adopt a measurement approach instead. We conduct a series of experiments to measure the temperature both on the surface of the absorbers using an infrared imaging camera, and internally using thermocouple probes inserted into the absorbers. Temperature profiles versus applied E field are experimentally established. From the measured data, we curve fit to Eq. (4) or other mathematical functions. These functions are useful to calculate results at other field levels, e.g.,
extrapolating to a higher field where measurement results cannot be readily obtained.

II. FIELD DISTRIBUTION INSIDE THE ABSORBERS

Numerical analysis was performed using Ansys HFSS, a commercially available Finite Elements software package. As it was described in [2], symmetry is taken advantage of, so only one quarter of the pyramidal absorber is solved. The quarter pyramid is located inside a square cross section prism that bounds the computational domain. The structure is fed using a port located on the top of the geometry and the side boundaries of the domain are set as perfect electric conductor (PEC) or perfect magnetic conductor (PMC). The base is modeled as PEC. This is exactly the same approach taken in [2]. The structure of a CRV-23PCL-4 is analyzed at 12.4 GHz, the same frequency as used in the measurements. The resulting field is extracted at one plane. The plane is one of the two orthogonal planes that cut the pyramid in 4 sections. Fig. 1 shows the field distribution at 12.4 GHz. The curvature of the absorber profile has been added for clarity. The results are an approximation. The permittivity of the material is assumed to be fairly constant from 6 GHz to 12 GHz. The purpose of the numerical analysis is to check the expected field distribution in the pyramid, which we can use to compare with the infrared (IR) images of the absorbers taken during the measurements.

The field distribution data shows that most of the field exists on the upper third of the pyramid. It also shows that there is a region of high field existing in the valleys between the pyramids. The surface temperature profile from the IR pictures shows that this is an real phenomena. On the other hand, the field is higher at the very tip of the absorber. Measurements from the IR images seem to contradict this result. This can be explained. Since the tip is smaller, it cools faster to the surrounding ambient temperature.

III. EXPERIMENTAL SETUP AND DATA

Experiments were performed on ETS-Lindgren CRV-23PCL-8, and CRV-23PCL-4 absorbers at 12.4 GHz. Both types are 23” long from tips to bases. A piece has a base size of 2’ × 2’. A CRV-23PCL-8 piece consists of 8×8=64 pyramids, whereas a CRV-23PCL-4 piece consists of 4×4=16 pyramids. The two types are designed to have similar RF performances, but the CRV-23PCL-8 is made of slender pyramids to facilitate better heat transfers to the surroundings [2]. The absorbers are mounted on a particle board with metallic backings, and are placed in front a Ku band horn antenna with a circular aperture (the gain is approximately 20 dBi). A 300W amplifier is used, and the power to the antenna is monitored through a 40 dB directional coupler connected to a power meter. The test setup is shown in Fig. 2. The ambient temperature is at 23°C.

As a first step, a 200 V/m field is generated by leveling to a calibrated electric field probe. The distance from the probe to the antenna is 30”. At this distance, near field coupling is assumed negligible, and the incident wave uniform (numerical simulation also validated these assumptions). The power needed to generate 200 V/m field is recorded. Next, the field probe is replaced with the absorbers under test. The tips of the absorbers are placed at the same distance (30”) from the antenna. Other field strengths can be leveled by scaling from the power for 200 V/m.

A. Surface Temperature

Figs. 3 and 4 show two examples of the infrared images taken after the temperature reached equilibrium under a constant 700 V/m CW at f=12.4 GHz for the two types of absorbers described earlier. There is no forced airflow during the measurement. Table 1 summarizes the resulting temperatures on the absorber surfaces at different field levels. Tests were performed on two finishes of otherwise identical CRV-23PCL-8 absorbers, i.e., fully covered with rubberized paint, or with latex paint. The data indicates that the paint has minimal effects on absorber temperatures. Table 1 also lists data for the wider CRV-23PCL-4 absorbers (with latex paint).

B. Internal Temperature of the Absorber recorded by Thermocouples

Three thermocouples are inserted in the CRV-23PCL-8 which are painted with rubberized coating. They are inserted at distances of 4”, 6”, and 8” from the tip of the pyramid, as illustrated in Fig. 5. Fig. 6 shows the temperatures measured by the three sensors. The temperatures at 8” from the tip are consistently higher than at other locations. There is a gap in the data at 700 V/m because RF power was turned off briefly. Internal temperature reached 115 °C under 1.7 kW/m².
(800 V/m). Since the maximum allowed temperature for the polyurethane foam material is 125 °C, the incident power density is recommended to stay less than 1.7 kW/m$^2$ for CRV-23PCL-8 absorbers mounted vertically and with natural convection in a 23°C room.

After the temperature reached equilibrium under 800 V/m, additional airflow was introduced by turning on a 6” diameter fan at 45” in front of the absorbers. The airflow rate was measured to be approximately 80 ft/min at this distance. Note that this is a rather moderate airflow, which can arise naturally from air-conditioning vents in a chamber. As shown in Fig. 6, the internal temperature quickly dropped to 102°C from 115°C.

<table>
<thead>
<tr>
<th>E (V/m)</th>
<th>Power Density (kW/m²)</th>
<th>CRV-23PCL-8 rubberized (°C)</th>
<th>CRV-23PCL-8 latex (°C)</th>
<th>CRV-23PCL-4 rubberized (°C)</th>
</tr>
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<tbody>
<tr>
<td>200</td>
<td>0.11</td>
<td>24</td>
<td></td>
<td></td>
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<tr>
<td>300</td>
<td>0.24</td>
<td></td>
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<td></td>
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<tr>
<td>360</td>
<td>0.34</td>
<td>30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>400</td>
<td>0.42</td>
<td>35</td>
<td>36</td>
<td>43</td>
</tr>
<tr>
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<td>600</td>
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<tr>
<td>700</td>
<td>1.30</td>
<td>63</td>
<td>82</td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>1.70</td>
<td>72</td>
<td>73</td>
<td>93</td>
</tr>
<tr>
<td>950</td>
<td>2.17</td>
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<td>112</td>
</tr>
</tbody>
</table>

There is no visual change to the absorbers, nor are there noticeable odors during or after the entire test.

<table>
<thead>
<tr>
<th>Locations of thermocouples</th>
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<tbody>
<tr>
<td>4&quot;</td>
</tr>
<tr>
<td>6&quot;</td>
</tr>
<tr>
<td>8&quot;</td>
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</tbody>
</table>

Fig. 5. Locations of the thermocouple sensors. The sensors are inserted halfway into the middle of the absorber pyramid.

![CRV23-8 Internal Temperature](image)

Fig. 6. Temperature inside the CRV-23PCL-8 absorbers measured by the thermocouples.

### IV. Data Analysis

#### A. Surface Temperature

The surface temperature vs. the power is described by Eq. (4), or can be rewritten as

\[
p = \left( \frac{E^2}{120 \pi} \right) \cdot A = k_1 \sigma (T^4 - T_0^4) + k_2 (T - T_0)
\]

where E is the field strength (V/m) of the incident plane wave, and A is the area (m$^2$) of illumination. Since A is a constant, it can be dropped or merged into constants $k_1$ and $k_2$. The two unknown coefficients $k_1$ and $k_2$ in Eq. (5) are determined through least square curve fit to the measured data. For CRV-23PCL-8 with rubberized coating (Table I, column 3),

\[
k_1 = 0.6678, \quad k_2 = 27.93.
\]

A simple second order polynomial fit is also shown in Fig. 7. The second order polynomial fit yields a similar response as the model given by Eq. (5). From Eq. (5), the ratio of the radiation to the combined radiation and convection heat loss can be solved. Fig. 8 shows the ratio of heat loss due to radiation. It is interesting to note that the radiation heat loss constitutes a small but significant percentage of the total heat loss (approximately 15%).

From Table 1, it can be seen that the rubberized coating on the absorber has no impact on the temperature and the heat transfer (comparing column 3 and 4). The maximum surface
Fig. 7. Maximum surface temperature vs. incident electric field for CRV-23PCL-8 with rubberized paint. The chart shows the measured data, curvefit data to the heat transfer equation, and a second order polynomial curvefit.

Fig. 8. Ratio of the heat loss due to radiation to the total heat loss.

Temperatures of CRV-23PCL-8 with and without rubberized paint show almost identical results.

Table 1 also shows that the slender design of CRV-23PCL-8 exhibits lower temperatures under the same incident field as compared to the wider CRV-23PCL-4. The slender design indeed improves the power handling capability [2]. The results are seen in Fig. 9.

B. Internal Temperature

Radiation heat transfer is insignificant inside the absorbers. For temperatures in the absorbers, we can assume the incident power is linearly proportional to the temperature, i.e.,

\[ p = k(T - T_0) \]

where \( k \) is a constant. Fig. 10 shows the internal temperatures inside the CRV-23PCL-8 absorbers measured by thermocouples at the 8" position, as it measures the maximum internal temperature. Fig. 10 also shows a linear fit to the measured data. The linear model proves to be a reasonable assumption.

Fig. 10. Maximum internal temperature measured by thermocouples vs. incident power density.

V. Summary

Measured temperature data is presented for polyurethane absorbers under varying field strengths at 12.4 GHz, both on the surface and inside the absorbers. Mathematical models are validated based on the underlying heat transfer mechanisms. Coefficients of the models can be solved from curve fitting to the measured data. The models are useful to "predict" absorber temperatures under otherwise difficult to achieve field levels. Absorbers with slender profiles prove to have improved power handling capabilities. Applying forced air at a relatively low flow rate can significantly decrease the absorber temperatures. In addition, rubberized paint on the absorber surface does not have noticeable effect on the temperature of the absorbers.

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References
