THERMAL DESIGN AND TEST REQUIREMENTS FOR OUTSIDE PLANT CABLE TELECOMMUNICATIONS EQUIPMENT

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*Abstract*

*Shrinking thermal margins, driven by sophisticated but thermally sensitive components, greater heat dissipation and industry requirements to operate in more challenging environments, threaten the reliability of outside plant equipment in advanced broadband networks. Simply specifying an ambient temperature of -40°C to +60°C is no longer adequate for outside plant equipment because the effects of wind speed, installation type and solar radiation can eliminate the thermal margin.*

## THE PROBLEM

### More Capability, Higher Temperatures

The demand for advanced, bandwidth hungry telecommunications and entertainment services will force system operators to add more functionality to their networks. Equipment power consumption and associated heat dissipation will increase to support these new services. The additional heat dissipation will drive the operating temperatures of lasers and other temperature sensitive components up, particularly in outside plant (OSP) equipment where active cooling is unattractive.

The test methods commonly used to exercise OSP equipment over temperature do not replicate the real world environment accurately or even conservatively. This masks potential field problems caused by inadequate or negative thermal margin. Small or negative thermal margins reduce system availability.

### Thermal Margin

The performance of electrical components and other materials used in optical nodes, RF amplifiers and other OSP equipment varies over temperature. There is a temperature range for each component in a piece of OSP equipment over which that component will provide adequate performance for acceptable system operation. The maximum and minimum temperatures in this range define the application specific temperature limits for each component.

Component temperatures in fielded OSP equipment are affected by ambient air temperature, solar loading, altitude, wind speed and wind direction. The materials and design of enclosures and the installation of the OSP equipment in enclosures further affect the temperatures of components housed in vaults, cabinets and pedestals. Component heat dissipation, the dissipation of neighboring components and the total dissipation of the entire piece of OSP equipment are other significant factors that influence component operating temperature.

The difference between the operating temperature of a component and its application specific temperature limit is the thermal margin. The thermal margin is negative if the component operating temperature exceeds either of the application specific temperature limits.

Improvements in cable telecommunications technology will push component operating temperatures up while maintaining or reducing the maximum application specific temperature limit.

### Increasing Heat Dissipation

New services, improved transmission schemes and more capable network management will be implemented in HFC networks. These advancements will add hardware and heat to network equipment.

Incumbent Local Exchange Carriers (ILECs) have already begun to see the effects of network advancements. A conventional

POTS (Plain Old Telephone Service) line consumes about 2 watts of central office power. ADSL lines consume between 6 and 8 watts. A central office that consumed 2000 to 3500 amps of electricity ten years ago may use 10,000 amps today. An ILEC that had 10 central offices in a medium-sized city in 1990 may now have 10 central offices, 150 vaults and 1,000 cabinets[1]. Most power provided to telecommunications systems becomes waste heat. Since the cable telecommunications industry plans to provide similar services, a similar growth in heat dissipation is a likely outcome.

The computer industry is another example of increasing heat dissipation. Dissipation of computer CPU's has increased significantly with processing capability and data rates. Figure 1 and Figure 2 illustrate this point for Intel and AMD processors[2].

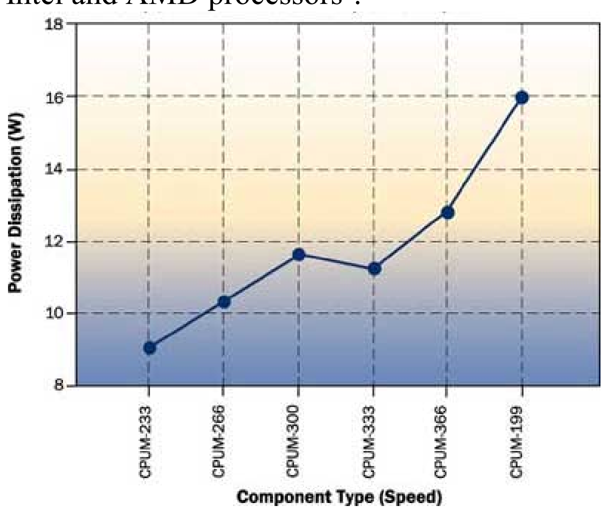


Figure 1 - Intel CPU Heat Dissipation vs. Clock Speed

The combined effects of new services, improved functionality and increased capacity will bring an increase in the heat dissipation of HFC network systems and their components.

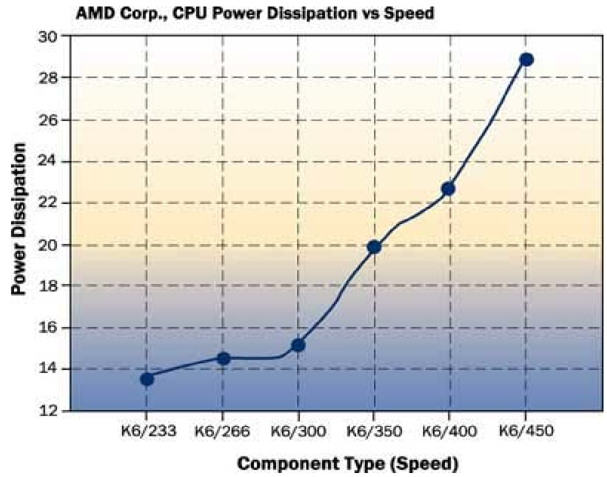


Figure 2 – AMD CPU Heat Dissipation vs. Clock Speed

Thermally Sensitive Components

RF amplifiers tested outside the normal operating temperature range for a short duration usually exhibit graceful performance degradation. The components being used or being considered for use in HFC networks today do not always degrade gracefully.

Optoelectronic components are necessary in the outside plant. Lasers are available with maximum case temperature ratings of 65°C to 85°C depending on the intended application. Lasers exhibit a sharp, ungraceful reduction in optical output power as case temperature increases past the maximum temperature limit. Lasers designed for use on the ITU wavelength grid also suffer from unacceptable wavelength drift at high temperatures. Both failure modes result in an unusable signal path.

Digital devices exhibit a variety of failure modes at extreme temperatures. The timing of critical high-speed signals may become skewed causing a breakdown in device to device communication. Random thermal noise can obscure low-level signals. Some devices will "lock-up" at temperatures outside the specification limits. These failures may cause degraded system performance or they can cause complete system failure.

The continued use of optoelectronics and the addition of digital electronics will maintain or lower the maximum application specific temperature limits for components used in OSP equipment.

First Order System Thermal Model

The remainder of this paper will focus on fiber optic nodes mounted in pedestal enclosures. The heat transfer mechanisms, design requirements and test methods presented are applicable to RF amplifiers and other OSP equipment.

Component operating temperatures in OSP equipment are a function of many variables. Figure 3 is a first order model that illustrates the relationships between key parameters for a node installed in a pedestal enclosure.

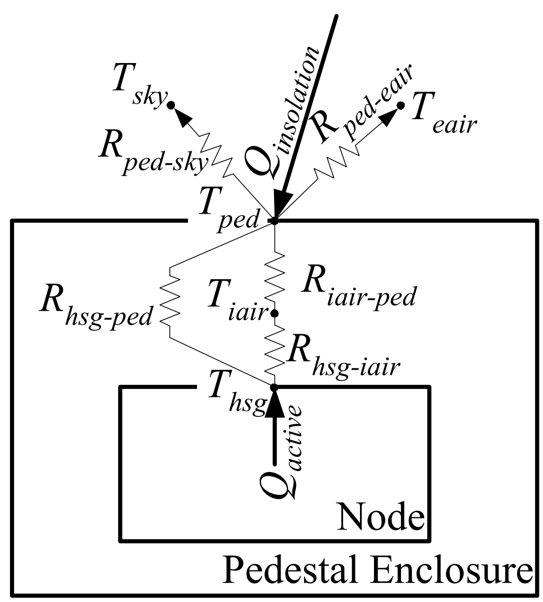


Figure 3 – First Order Thermal Model of a Node Installed in a Pedestal Enclosure with Local Wind Speed Equal to Zero

| | |
|---|---|
| $Q_{active}$ | Heat Dissipated by Node |
| $R_{hsg\text{-}iair}$ | Thermal Resistance from Node Housing to the Internal Air (Convection) |
| $R_{hsg\text{-}ped}$ | Thermal Resistance from Node Housing to Pedestal Enclosure (Radiation) |
| $R_{iair\text{-}ped}$ | Thermal Resistance from Internal Air to Pedestal Enclosure (Convection) |
| $R_{ped\text{-}eair}$ | Thermal Resistance from Pedestal Enclosure to External Air (Convection) |
| $R_{ped\text{-}sky}$ | Thermal Resistance from Pedestal Enclosure to Sky (Radiation) |
| $T_{hsg}$ | Average Node Housing Temperature |
| $T_{iair}$ | Pedestal Internal Air Temperature |
| $T_{ped}$ | Pedestal Enclosure Temperature |
| $T_{eair}$ | Ambient (External) Air Temperature |
| $T_{sky}$ | Sky Temperature |
| $Q_{insolation}$ | Heat Transferred to the Pedestal Enclosure by Direct Solar Radiation |

Equation 1 governs the temperature rise across the resistors in Figure 3.

$$\Delta T = QR_{thermal} \tag{1}$$

Q is the heat passing through each resistor. The sky and external air temperatures are not controllable. As Q increases, the temperature difference across each resistor increases.

The values of $R_{thermal}$ depend on the mode of heat transfer. Heat transfers by conduction, convection and radiation. Conduction is not a significant mode of heat transfer in pedestals. Convection can be free (natural) or forced. Thermal resistance for convection is determined from equation 2.

$$R_{Thermal} = \frac{1}{hA} \tag{2}$$

*h* is the convective heat transfer coefficient. *A* is the surface area of the node or other OSP equipment.

The value of $h$ is calculated differently for free and forced convection. $h$ under forced convection depends on air velocity, length in the direction of flow and several fluid properties. $h$ under natural convection depends on surface orientation with respect to gravity, the temperature difference between the surface and the surrounding fluid and several fluid properties. Analytical estimation of $h$ for real systems is complex and is usually done numerical techniques such as Computational Fluid Dynamics (CFD).

Heat transfer by radiation follows equation 3[3].

$$Q = \sigma f e A (T_{hsg}^4 - T_{ped}^4) \tag{3}$$

$\sigma$ is the Stefan-Boltzmann constant, $f$ is the view factor, $e$ is the emissivity factor and A is the surface area of the OSP equipment. For a first order approximation of a node in a pedestal, $e$ is approximately equal to the emissivity of the node surface and $f$ is approximately equal to one. CFD codes optimized for electronics cooling applications include a radiation analysis capability for better estimates of heat transfer by radiation.

The node housing temperature is the sum of the temperature differences across the resistors added to the external air/sky temperature.

Internal component operating temperatures (e.g. return transmitter lasers) are dependent on the node housing temperature. Components will approach and eventually exceed their application specific temperature limits if the heat dissipation of cable telecommunications equipment follows the trends in the computer and telephone industries. The consequences of this trend are shown in Figure 4.

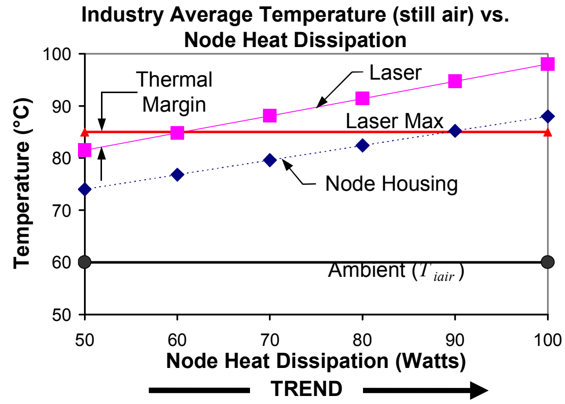


Figure 4 – Industry Average Node Temperature vs. Node Heat Dissipation in a Still Air Ambient Environment (Pedestal)

## The Real World vs. Environmental Chambers

Figure 4 depicts node housing and laser temperatures as a function of node heat dissipation in still air. Still air conditions can exist in pedestal enclosures, cabinets and underground vaults. Most high and low temperature OSP equipment tests are run in environmental chambers. Environmental chambers employ moving air to keep the chamber temperature uniform. $R_{hsg\text{-}iair}$ for a node in a chamber will be much lower than $R_{hsg\text{-}iair}$ for a node operating in still air. The reduction of $R_{hsg\text{-}iair}$ will lower the node temperature relative to the surrounding air. Figure 5 shows how testing a node in a chamber can hide the negative thermal margin that will exist when the same node is operating in a pedestal in the outside plant.

The laser that appeared to have adequate thermal margin in the environmental chamber (Figure 5) only has adequate thermal margin in a fielded pedestal if its node dissipates less than 60 Watts of heat (Figure 4).

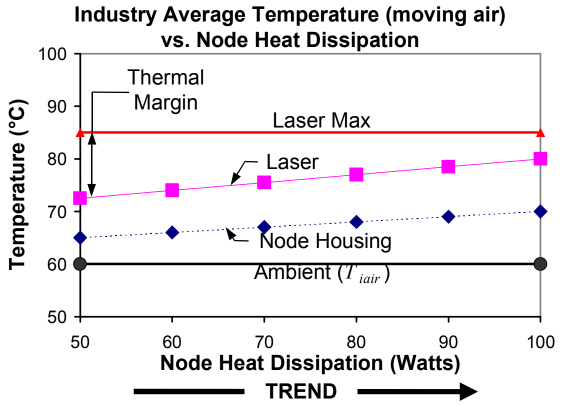


Figure 5 - Industry Average Node Temperature vs. Node Heat Dissipation in a Moving Air Ambient Environment (Environmental Chamber)

Increasing heat loads, constant or declining application specific component temperature limits and non-conservative test methods will result in negative thermal margins in future generations of OSP equipment. Design requirements that describe and test methods that reproduce the OSP environment more accurately are necessary to ensure that equipment delivered to the field is thermally robust.

## NODE HEAT TRANSFER IN A PEDESTAL

An understanding of pedestal heat transfer mechanisms is necessary to write design requirements and develop tests for a pedestal environment. Figure 3 models the effects of these mechanisms. The resistor values depend on many variables.

Natural convection and radiation cool a node operating in a pedestal when the external air is still. Natural convection moves heat from the node to the surrounding air. Heat from the node warms the adjacent air. The warm air rises to the top of the pedestal enclosure and moves from the center of the enclosure to the sides. The air cools as it comes in contact with the relatively cool sides of the enclosure. The cool air sinks to the bottom of the enclosure where it waits to be "sucked" past the node. The flow lines in Figure 6 illustrate natural convection airflow in a pedestal enclosure.

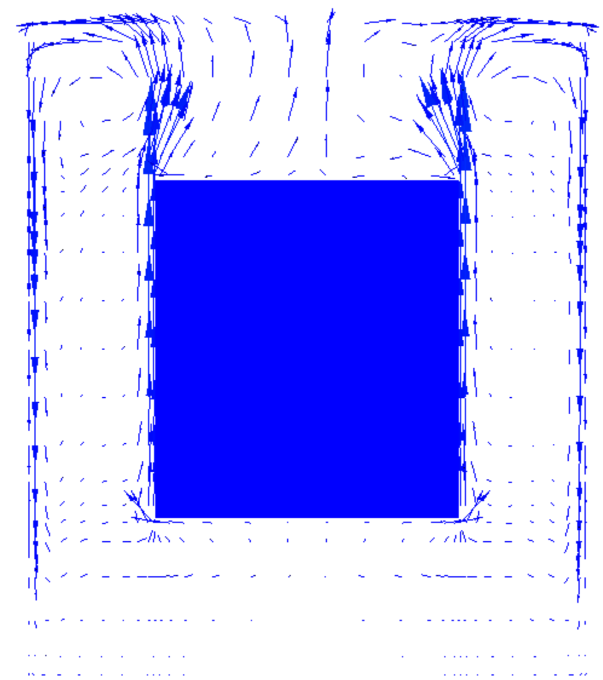

Figure 6 – Natural Convection Airflow around a simplified Node in a Pedestal without Solar Loading

Heat transfers by radiation from the node to the pedestal enclosure and from the node to the ground at the bottom of the enclosure when the node is at a higher temperature than the enclosure or the ground. Radiation heat transfer between the node and the pedestal enclosure can vary significantly depending on the emissivities of the surfaces and the temperature differences between the surfaces. If the interior surface temperature of the pedestal walls becomes greater than the temperature of the node surface, heat will transfer by radiation from the pedestal enclosure to the node. This could happen with certain pedestals under solar loading.

Figure 6 depicts a pedestal in still air with no solar loading. Solar loading changes the internal airflow pattern. Direct solar load will irradiate two vertical walls and the top of the pedestal depending on the orientation of the pedestal and the position of the sun. Other walls may receive solar radiation reflected from nearby walls, sidewalks and other terrain features. The warm air rising from the node must pass the non-irradiated walls for cooling if the internal temperature of the irradiated walls is equal to or greater than the temperature of the node. The asymmetrical and more restricted airflow will cause $R_{hsg\text{-}iair}$ to increase. The solar load on the pedestal will increase $T_{ped}$. The node housing temperature, $T_{hsg}$ will increase. The magnitude of the increase will depend on the materials used in the pedestal, the orientation of the sun, the pedestal size, the node size and the location of the node within the pedestal.

Wind will pass through the ventilation slots in the pedestal and mix with the internal air. The wind speed inside the pedestal will be less than the outside wind speed. The reduction in wind speed depends on the pedestal orientation, size, vent geometry and vent cleanliness. Local obstructions such as landscaping, buildings and fences also affect internal wind speed. Increasing internal wind speed will lower $R_{hsg\text{-}iair}$ and reduce $R_{iair\text{-}ped}$. A new parallel resistance, $R_{iair\text{-}eair}$, reduces the overall resistance between the housing and the external ambient air. See Figure 7.

Wind will result in lower internal air and node housing temperatures than still air. Still air should be assumed for design purposes because it represents a conservative but occasionally real field condition.

One consequence of pedestal fluid mechanics is higher air temperatures at the top of the pedestal than at the bottom. Temperature differences of up to 20°C have been measured between the top and the bottom of fielded pedestals.

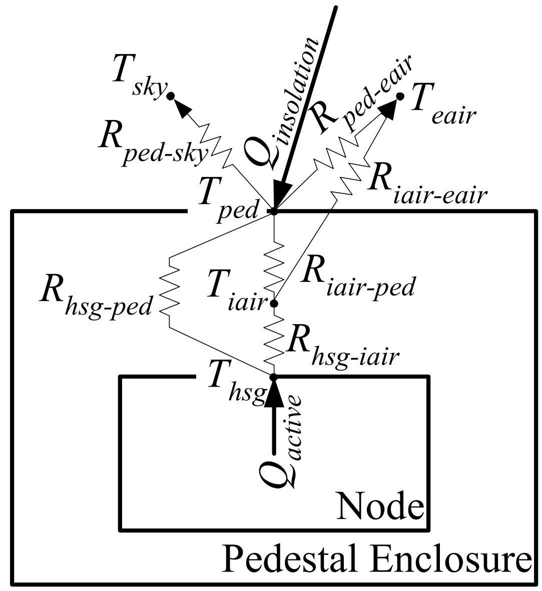


Figure 7 - First Order Thermal Model of a Node Installed in a Pedestal Enclosure with Local Wind Speed Greater than Zero

## DESIGN REQUIREMENTS

Thermal design requirements for any electronic product must address the three primary mechanisms of heat transfer; conduction, convection and radiation. Heat transfers from a heat source to a heatsink. Heatsink temperatures and parameters that determine the thermal resistance between the heat source and heatsink must be specified.

Conduction from a node to a pedestal or to the surrounding environment is a relatively minor means of heat transfer. Conductive heat paths through the mounting points, coaxial cables and optical fiber cables are long and filled with high resistance materials or interfaces. Assuming that no heat is transferred from the node by conduction is a slightly conservative assumption that is reasonable for design purposes.

Radiation could be significant if the emissivity of the node is high. The effect of radiation could increase or decrease the temperature of the node depending on the temperature and emissivity of the inside surfaces of the pedestal. If radiation is to be assumed as a means of heat transfer, pedestal surface temperatures, emissivities, dimensions and locations must be specified. The ground temperature and the node location with respect to the ground and the pedestal must be specified also. Radiation can be neglected as a mode of heat transfer if the emissivity of the node is low.

Convection is the primary mode of heat transfer for most electronic equipment operating in pedestals. Still air outside the pedestal should be assumed. Natural convection will be cooling the OSP equipment. The heatsink temperature for natural convection should be the temperature of the isothermal pool of air below the node (see the "Test Guidelines" section). Minimum pedestal size and maximum internal surface temperatures must be specified. Node location and orientation must be defined.

## TEST GUIDELINES

### Measurement Point for Node Ambient

Suppose a node is tested in a simulated pedestal environment (i.e. still air) where the temperature is 60°C at the top to 40°C at the bottom. If the node meets all of its performance requirements, should it be rated as acceptable for use up to 40°C or 60°C?

The air temperature inside a pedestal should be measured at the bottom of the pedestal at least 3 inches below the bottom of the node. The node draws its cooling air from the air at the bottom of the pedestal. The temperature of the air at the bottom has the strongest influence on the temperature of the node. Test repeatability is another reason for measuring temperature at the bottom of the pedestal. Simulations and tests show that temperature is uniform in the air at the bottom of the pedestal. The location of the ambient temperature sensors is not critical as long as they are mounted below the thermal boundary layer under the node.

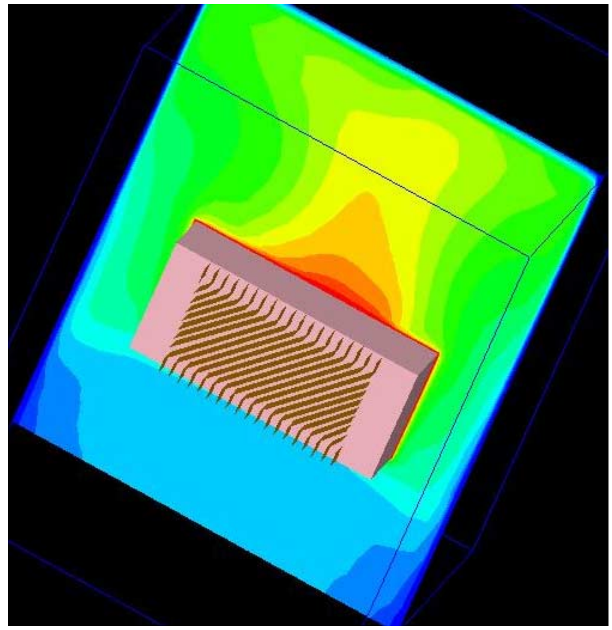

Figure 8 – Predicted Air Temperature Around a Node in a Pedestal in Still Air Without Solar Loading

The contour lines in Figure 8 are isotherms on a plane that passes through the center of a node in a pedestal. Large temperature gradients exist at the top while the temperature at the bottom is uniform. A slight shift in the location of a sensor at the top of the pedestal could result in a large change in the "ambient" temperature reported by that sensor. A sensor at the bottom could be moved without causing such a variation in the indicated temperature. Figure 8 was produced with CFD software.

### Test Procedure

OSP equipment performance testing at high or low temperatures is usually done in environmental chambers. The circulating air in a chamber makes the test environment different than in a fielded pedestal. A correction must be made to the chamber temperature to get the OSP equipment housing up to the temperature it will achieve in a pedestal.

Run a simulated pedestal test to estimate the temperature rise from the ambient below the unit under test (UUT) to the UUT housing.

Perform the test in an environmental chamber that is close to the size of an appropriate pedestal. Instrument the UUT housing with temperature sensors near the mounting points for modules and other heat sources. Several sensors should be placed in the air below the UUT to establish where the thermal boundary layer ends and to verify that the pool of isothermal air has been found. Put at least two sensors in the pool of isothermal air because these sensors will determine the reference for temperature rise calculations. Locate and orient the UUT as it would be in a pedestal.

The heat dissipation of the UUT should be as close as possible to its expected maximum. The expected maximum heat dissipation includes the heat dissipated by all application modules and power converters at their maximum operating temperatures. If the UUT is configured to pass current to other devices in the network, then include the ohmic losses ($I^2R$) created by the maximum passed current.

Run the simulated pedestal test with the chamber control system (heaters, chillers and fans) turned off and the chamber door closed. Verify heat dissipation throughout the test to ensure that all heat sources are working and that the total heat dissipation is correct. Monitor temperatures often enough to determine when the UUT has thermally stabilized with the isothermal pool of chamber air. Stabilization with the chamber air is achieved when the difference between the average UUT housing temperature and the average air temperature in the isothermal pool reaches a constant value.

$$\Delta T_{Natural} = \overline{T}_{UUT} - \overline{T}_{IsothermalPool} \tag{4}$$

Estimate the absolute housing temperature in a pedestal by adding the maximum rated temperature of the UUT to the temperature difference at stabilization. For example, if the UUT is rated to +60°C and the stabilized temperature difference is 25°C, then the estimated average housing temperature in a pedestal is +85°C.

After the UUT has stabilized and the temperature and power data have been checked, turn on the chamber control system. Adjust the chamber temperature to keep the UUT housing temperature where it was prior to activating the control system. The UUT heat dissipation and average housing temperatures should be the same as they were when the stabilized temperature difference was calculated previously. When the UUT is thermally stabilized with the chamber air, calculate the temperature rise from the chamber ambient to the housing.

$$\Delta T_{Forced} = \overline{T}_{UUT} - \overline{T}_{IsothermalPool} \tag{5}$$

<u>Data Reduction</u>

Calculate the thermal resistance between the UUT and the chamber air.

$$\Theta_{Natural} = \frac{\Delta T_{Natural}}{Q_{UUT}} \tag{6}$$

$$\Theta_{Forced} = \frac{\Delta T_{Forced}}{Q_{UUT}} \tag{7}$$

Calculate an equivalent chamber set temperature, $T_{Forced}$ for any real world pedestal internal temperature, $T_{Natural}$ and UUT heat dissipation.

$$T_{Forced} = Q_{UUT}\left(\Theta_{Natural} - \Theta_{Forced}\right) + T_{Natural} \tag{8}$$

Example:
A node consuming 70 true RMS watts of AC power and passing 15 amps of AC current is tested as described above. The measured temperature rise in still air (natural convection) is 20°C and in moving air (forced convection) is 8°C. The electrical resistance through the AC power bus is 44mΩ. What chamber temperature should be used for simulating a pedestal still air bottom temperature of 60°C?

1) Calculate the total heat dissipation:

$Q_{UUT} = Q_{Consumed} + Q_{Ohmic}$
$Q_{UUT} = Q_{Consumed} + I^2R$
$Q_{UUT} = 70W + (15A)^2(0.044\Omega)$
$Q_{UUT} = 80W$

2) Calculate the thermal resistances:

$$\Theta_{Natural} = \frac{\Delta T_{Natural}}{Q_{UUT}}$$

$\Theta_{Natural} = 20°C/80W = 0.25°C/W$

$$\Theta_{Forced} = \frac{\Delta T_{Forced}}{Q_{UUT}}$$

$\Theta_{Forced} = 8°C/80W = 0.10°C/W$

3) Determine the appropriate forced convection chamber temperature:

$$T_{Forced} = Q_{UUT}\left(\Theta_{Natural} - \Theta_{Forced}\right) + T_{Natural}$$

$T_{Forced} = 80W(0.25°C/W\text{-}0.10°C/W) + 60°C$

$T_{Forced} = 12°C + 60°C = 72°C$

Application

$\Theta_{Natural}$ and $\Theta_{Forced}$ will remain constant for a given housing and chamber combination. Changes to housing geometry or material could affect $\Theta_{Natural}$ and $\Theta_{Forced}$. Natural convection testing in a chamber of different interior dimensions, especially if the clearances between the UUT and the chamber walls are less than 6 inches, will change $\Theta_{Natural}$. Differences in chamber air speed and direction relative to the UUT will change $\Theta_{Forced}$. If the values of $\Theta_{Natural}$ and $\Theta_{Forced}$ are in question as the result of a design or chamber change, a test or some other evaluation is recommended.

A simple calculation using Equation 8 will produce corrected chamber temperature for standard forced convection performance tests. $\Theta_{Forced}$ usually varies over a small range for the same UUT tested in different chambers of similar size and air moving capacity. Nodes, amplifiers and line extenders can be tested once in a "pedestal sized" chamber and the resulting values for $\Theta_{Natural}$ and $\Theta_{Forced}$ can be applied to tests in similar sized chambers.

Limitations

This test method produces an approximation of pedestal based component operating temperatures in a standard environmental chamber. The test and data reduction procedure described above is a linear approximation of a non-linear system. Internal pedestal temperatures are dependent on many variables including pedestal material, vent design and the local environment. The test procedure has no provisions to simulate solar radiation load. Practitioners must evaluate the heat transfer mechanisms applicable to their situation and then determine the suitability of this approach.

## CONCLUSION

Market forces are driving the cable telecommunications industry to more sophisticated network equipment. The heat dissipation and internal temperatures of new equipment will increase if trends in the computer and telecommunications industries are realized. Reduced thermal margins require explicit thermal design requirements and accurate test methods. A test method is presented that allows OSP equipment to be tested in an ordinary environmental chamber in a way that reproduces the component temperatures experienced in a fielded pedestal.

References
[1] Branson, K., "More Power! More Power! But where will service providers get it?" Broadband Week, Jan 8, 2001.
[2] Azar, K., "The History of Power Dissipation," Electronics Cooling Magazine, Jan 2000.
[3] Steinberg, D., Cooling Techniques for Electronic Equipment, Second Edition, p128.