Thermal Test Chamber

Construction, Considerations and Constraints

Introduction

It is not uncommon for an experienced test professional to find the selection of a thermal test chamber to be a confusing process. Many factors must be considered in order to specify the right chamber to perform a test that ends with acceptable results. Among the factors that must be considered are the temperature extremes that the Unit Under Test (UUT) must experience, the speed at which the UUT must transition, the total mass contained within the chamber and the method used to perform the temperature conditioning.

There are many branches of science and engineering devoted to establishing a methodology for modeling the phenomenon of heat transfer between two bodies of differing temperatures. These include thermodynamics, heat and mass transfer and fluid dynamics. To properly apply these disciplines to temperature testing, one must have a fundamental understanding of the way a chamber works and how the operation of the chamber can be maximized for efficiency.

It is the purpose of this paper to provide such a discussion. The topics presented include the modes of heat transfer, a method of determining the load calculations, the effects that the blowers have on the efficiency, the methods and considerations for cooling and heating the chamber and some of the basic equations needed to engineer an appropriate chamber for a specific need.

Chamber Construction

A thermal test chamber is designed to produce a temperature change in a device. The magnitude and rate of this change should induce a sufficient thermal stress fatigue to identify sources of potential failure without causing catastrophic failure of all components.

A basic chamber consists of an insulated box, heaters, some sort of cooling device, a means of circulating the air within the chamber, and controller that regulates each of these devices to reach and maintain the desired temperature. Thermal Test chambers come in a variety of sizes and geometries. Additionally, temperature chambers can be designed for fast ramps, ultra high or ultra low temperatures. The design of the chamber should be driven by the requirements of the test being performed, the geometry of the device being tested, and the available resources of the facility where they are being performed.

Heat Transfer

Heat transfer can be described as energy in transit due to temperature difference[1]. If two bodies at different temperatures are brought into contact, energy (heat) will flow from the bodies of higher temperature (warmer body) to the body of colder temperature (colder body). Further, if the two bodies remain in contact for a sufficiently long period of time, they will reach the same temperature.[2] The rate of heat transfer between two bodies is a function of many factors including, but not limited to, the difference in temperature between them.

There are three methods of heat transfer: convection, conduction, and radiation. The following discussion will provide an introduction to each.

Conduction is the transfer of intermolecular energy from the more energetic to the less energetic

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particles within a substance. The heat flux rate equation that describes conduction is expressed as:

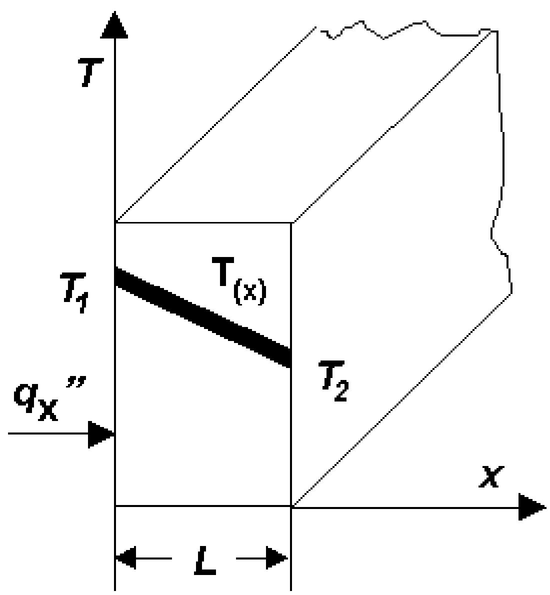
$$q_x'' = -k\frac{dT}{dx} = -k\frac{(T_2 - T_1)}{(x_2 - x_1)}$$

$$Where; q_x''(\frac{W}{m_2}); k(\frac{W}{m \bullet {}^\circ K}); T(\mathrm{o}K); x(m)$$

In the equation 1, q_x'' is the heat flux and is a measure of the heat transfer rate in the x direction perpendicular to the direction of transfer and it is proportional to the temperature gradient dT/dx in this direction. The transport property k is known as the thermal conductivity and is a characteristic of the conducing material.

Figure 1. Graphic Representation of Conduction Rate Equation

The minus sign is a consequence of the fact that the heat is transferred in the direction of decreasing temperature.



Convection is know as the transfer of heat by conduction from solid surface to a moving fluid. Convection heat transfer is a function of two mechanisms; random molecular motion (diffusion), and bulk, or macroscopic motion on the fluid. Convection can be further categorized as being forced or free flow. Forced convective heat transfer is caused by some external means such as a fan, pump, or wind, where free flow convective heat transfer is induced by the buoyant forces of the fluid. These buoyant forces arise from density variations caused by temperature differences in the fluid.

The governing rate equation for convective heat transfer is:

$$q'' = h(T_s \quad - T_\infty)$$

$$Where: q''\left(\frac{W}{m^2}\right); h\left(\frac{W}{m^2 \bullet \mathrm{o}K}\right)$$

In equation 2, q'' (W/m^2) is the convective heat flux, and h (W/m^2 **$**K) is the convective heat transfer coefficient, also known as the film conductance or the film coefficient. It encompasses all the effects that influence the convective mode of heat transfer.

Radiation heat transfer refers to the energy that is transferred by electromagnetic waves emitted by matter as a result of changes in the electron configuration of the constituent atoms.

The two primary methods of heat transfer that occur in a thermal test chamber are convection between the atmosphere surrounding the UUT and the bounding surface of the UUT, and conduction within the UUT.

Boundary Layers

A consequence of any fluid-surface interaction (as is found in convective heat transfer) is the development of a region in the fluid through which the velocity varies from zero at the surface to a finite value u_4 associated with the flow. This region is known as the hydrodynamic, or velocity boundary layer. If there is a difference in the surface and flow temperatures, there will also develop a region in the fluid through which the temperature varies from T_s at y=0 to T_4 in the outer flow (see Figure 2). This region, called the thermal boundary layer, may be smaller, larger or the same size as that through which the velocity varies.

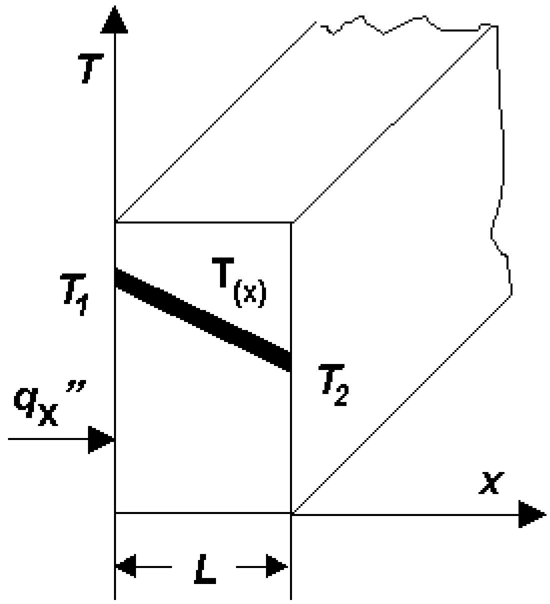


Figure 2. Boundary Layer Development in Convection Heat Transfer

It is often the case that a stagnant hydrodynamics boundary layer will act to insulate or resist the flow of thermal energy from the UUT to the surrounding fluid. The magnitude of the insulating effects are directly proportional to the heat transfer characteristics of the fluid and the hydrodynamic profile of the fluid flow. Generally, the more stagnant the flow the greater the insulating effects.

It is for this reason that the UUT should be placed into the airstream oriented so as to maximize the surface over which the stream flows and minimize any stagnant zones around the device. To demonstrate this, hold your hand a foot away from your face and blow on the back of your hand. Repeat the procedure, but blow onto the side of your hand. Observe the cooling effects generated by each orientation.

Load Calculations

Components that influence the loads include the mass and materials of the UUT, the chamber liner (inner shell) and any additional components and devices such as blowers, plenums or shelves. In addition, heat generating capabilities and energy losses must be considered to be minimum.

The energy required to change the temperature of any item can be calculated by considering the mass and specific heat of the item and the temperature extremes that the item must experience. Specific heat is defined as the amount of heat required to produce a desired temperature change in an exact amount of any substance and is expressed in J/(EC$g) or BTU/(F$Lbm). The mass of an item is the product of the volume and the density. The following table lists the specific heats of many materials commonly found in the thermal test arena.

Item	Temp (EC)	Specific Heat (J/EC$g)
Water Liquid	15	4.184
Water Solid	-11	2.030
Air	27	1.007
Aluminum	20	0.890
Stainless Steel	27	0.477

Table 1. Specific heats, common materials.

The calculation of the idealized (perfect efficiency) energy required to change a device temperature is readily available. The following illustrates the methodology:

A Sigma Systems Model M170 Thermal Test Chamber as interior dimension of 25.13" High x 20.5" Wide x 20.8" Deep. The interior wall material is stainless steel (30/1000" thick) with the physical properties listed above. The is an interior plenum and baffle system that has the same effective area as the back wall. There are two stainless steel blowers that weigh 2.3 Lbs. each. Not including wall losses, how much energy is required to transition this chamber +125EC to -40EC

Solution:

Stainless Steel Physical Properties:
Density (ρ): 0.29 Lb/in^3
Specific Heat: 0.477J/EC$g
Temperature Change: +125EC -(-40EC)=165EC

Mass Calculation:

Linear Mass =ρ $ volume
=ρ $ (thickness $ area)
=0.249 [Lb/in^3]
$30/1000
${2(25.13$20.08)
+2(20.5$20.08)
+3(25.13$20.5)}
=25.23 Lb.
Blower Mass =4.6 Lb.
Total Mass =29.83 Lb.
=13 530.66 g

Energy Calculations:

Energy =Mass$Specific Heat
$ ªT
=13 530.66 (g)$0.477(J/C $ g) 165 (C)
=1 064 930.63 J
= 1 009.55 BTU

Blowers

The effects blowers have on the operation of a thermal test chamber are important. The size and orientation of the blowers will influence the efficiency of the chamber and the heat transfer characteristics between the UUT and the atmosphere.

The insulating effect of the hydrodynamic and thermal boundary layer can be minimized if the thickness of the boundary layers can be decreased. This can be achieved by circulating enough air though the chamber to transition the flow from laminar (smooth motion in laminae or layers) to turbulent (random, three dimensional motion superimposed on the mean motion).

The chaotic vortices of a turbulent flow will effectively disrupt the hydrodynamic and thermal boundary layers and increase the efficiency with which the thermal energy can transfer from a UUT at a given temperature to the surrounding atmosphere that is at a different temperature. There is no unique point at which this transition will occur. However, it is generally accepted that the greater the air velocity through a given chamber, the greater the potential for turbulent flow.

There is a trade-off for achieving turbulent flow. The power required to move a given mass of air at a velocity sufficient for turbulent flow will impart greater frictional energy to the air, thereby increasing the kinetic energy of the atoms that make up the air column. This energy increase will give rise to an increase in the air temperature to the point where a desired low temperature may not be achievable, or additional cooling may be required to maintain set-point. It is for this reason that blowers must be carefully sized to induce the proper flow characteristics without imparting an excessive increase in the air temperature.

In any thermal test chamber, the effects of free flow convection will produce an undesirable temperature gradient if allowed to go unchecked. One method to curtail the formation of these temperature gradients is to design the chamber such that the air circulates from the top of the chamber to the bottom. This "counterflow" design will aid in mixing the warm air that tends to raise to the top of the chamber with the cooler air that will tend to settle on the bottom of the chamber.

Cooling

The most common methods of cooling thermal test chambers include mechanical refrigeration and the use of cryogenic (low temperature) fluids. Each method has its benefits and disadvantages.

Mechanical refrigeration is ideally suited to applications where cryogenics fluids are either not available, or prohibited from use. A mechanically refrigerated chamber is completely self contained and requires only an electrical connection to operate.

Mechanical systems are available in a variety of configurations. The following matrix may aid in determining the configuration that is right for your needs.

Attribute	Rep Time[H] from +125 to -40	Average Low Limit
1/2 Horsepower	60 Min.	-45EC
1/2 Hp x 1/2 HP Cascade	55 Min.	-70EC
1 Horsepower	40 Min.	-45EC
1 Hp x 1 Hp Cascade	20 Min.	-70EC
[H] Times taken from a Sigma Systems Supercycler Comparison Data Chart.		

Table 2. Slew Rate vs. Power in Mechanical Systems

The most common form of cryogenically cooled chamber uses the direct expansion of either liquid nitrogen (LN_2) or liquid carbon dioxide (LCO_2) as the refrigerant.

The temperatures of cryogenically cooled thermal test chambers are achieved by injecting the high pressure cryogenic refrigerant through a small diameter tube directly into the chamber interior. As the refrigerant is injected into the chamber, the injection capillary tube acts as a throttling device as the liquid transitions from high to low pressure. Throttling the refrigerant in this manner initiates a remarkable change of state. In the same way a pressurized fluid that passes through the nozzle of a spray bottle will atomize to a mixture of vapor and microscopic particles, so will the chamber refrigerant atomize as it passes through the injector tube. As LN_2 is throttled, it produces a vapor-fluid mixture, however, CO_2 throttles to a vapor-solid (vapor dry ice) mixture.

The physics involved with throttling refrigerant into a chamber or thermal platform is of considerable interest. To fully understand this process, a thermodynamic analysis of the expendable refrigerant must be made. For the purposes of this article, only CO_2 will be explored. However, most of the concepts presented can easily be applied to LN_2 (Note: LN_2 uses the evaporation of suspended droplets in place of the sublimation of dry ice in the following discussion).

CO_2 that is supplied in un-insulated "syphon" cylinders has a vapor pressure of 6,300 KPa (900 psia) at 23EC (73EF). Liquid CO_2 cooled to -17.75EC (0EF) with a vapor pressure of 2200 KPa (305 psia), is delivered in vacuum insulated containers generally referred to as "liquid cylinders."

Using $CO_{2,}$ most of the cooling is provided by the sublimation (change of state from solid to gas) of dry ice particles. The CO_2 vapor is merely the carrier. This is true because the cooling is taking place in the saturation region between the saturated solid line and saturated vapor line on the pressure enthalpy diagram (see Figure 3).

Figure 3 Generalized Pressure-Enthalpy Diagram for Carbon Dioxide

Since CO_2 vapor remains at the same temperature throughout the coolant path, the vapor gives up any absorbed heat to the dry ice particles as they sublimate. The following table shows the dry ice heat absorption in BTU/Lb. when liquid CO_2 is throttled to atmospheric pressure and temperatures:

Source	Temp	Energy Absorbed
900 psia	73EF	68 BTU/Lb
900 psia	0EF (subcooled)	113 BTU/Lb
900 psia	-40EF	131 BTU/Lb
305 psia	0EF	113 BTU/Lb
305 psia	-40EF (subcooled)	131 BTU/Lb
Low Limit for either 900 or 305 psia CO_2	69.9EF	145 BTU/Lb

Table 3 Heat Absorption of CO_2 as it Throttles to Atmospheric Conditions

Calculating a Chambers Cooling Requirements

An environmental test chamber operating below ambient temperature requires refrigeration to compensate for the following factors.

1. The heat energy flowing into the chamber through the walls
2. The heat generated by lamps, fans and blowers within the chamber
3. The heat generated from active loads

$$Q = \frac{A x\,(room\ temperature - chamber\ temp)}{R\ x\,(insulation\ thickness\ in\ inches)}$$

Heat flow through chamber walls can be calculated by the following formula.

Where

Q is the heat transfer in BTUs/hour
A is the total surface area of the test volume in square feet
R is the heat resistance of the insulating material per inch thickness

To illustrate, the cooling requirement at -40EF for an eight cubic foot cubical chamber with walls insulated with 1" thick fiberglass blanket is calculated here.

A=Wall area = 6 x 4 square feet = 24 square feet
R for fiberglass blanket per inch = 3.12
Temperature difference = 70EF -(-40EF) = 70+40=110EF

$$Q = \frac{24*110}{3.12} = 846.2\ BTUs/hr$$

If this chamber has a blower system that dissipates 0.150 kilowatt hours then the added energy in BTUs/hour is 0.15 x 3413 = 512.0 BTUs/hr.

The total energy that has to be extracted per hour would be 846.2+512.0=1,358.2 BTUs. Using the data for heat absorption in BTU/Lb. For a particular refrigerant, it is possible to estimate the amount of cryogenic refrigerant needed to transition a particular chamber between two temperatures. It is important to not that the calculations yield estimates only. The actual consumption is a function of the chamber efficiency.

Determining Response of Unit Under Thermal Test

The physics of heat transfer dictate that two bodes of differing materials and geometries must respond differently to equivalent thermal stimulus. How then, does one insure that a specific device will experience the proper thermal stress to induce a measure of component failure in a minimal amount of time? The answer lies in applying a known thermal stimulus and observing the response.

> It must be noted that the concepts and procedures provided here are given as an additional means to confirm accepted procedure and are not intended to be, nor should they be used in place of any required tests.

In most commercially available thermal test chambers, the mode of heat transfer between the UUT and the surrounding atmosphere is convection. Two major factors influence the rate at which this convective heat transfer occurs: 1) the resistance effects of the atmosphere film surrounding the UUT, or film factor, and 2) the thermal inertia, or rate at which the UUT will absorb the heat energy. In mechanical systems, inertia is modeled as capacitance and response to thermal stimulus is similar to current flow in an electrical circuit containing capacitance.

For simple thermal systems, the temperature change of the UUT over time is equal to the rate of convective heat transfer into the UUT. This can be described mathematically using Newtons' Law of

$$hA(T_e - T) = Mc\frac{dT}{dt}$$

Cooling, and Blacks' Heat Capacity Equation;

Where

$$(\frac{W}{m^2 * \circ K})$$

h= Convective Heat Transfer Coefficient
A=Surface Area (A)
T_e=Temperature of Chamber Environment (EK)
T=Temperature of UUT (EK)

$$c = UUT\ Heat\ Capacity\ (\frac{W * s}{Kg * \circ K})$$

M=Mass of UUT (Kg)
t=Time(s)

The mass [M] of the UUT can be represented as the product of density [ρ] and volume [v] allowing the equation to be written as:

$$(\frac{hA}{rvc})\,dt = (\frac{1}{(T_e - T)})\,dT$$

The term hA is the thermal conductance of the film envelope surrounding the UUT, and ρ*vc* is the thermal capacitance of the UUT. Because thermal resistance is the inverse of thermal conductance, the term on the left side allows the equation to be written as:

$$(\frac{1}{(\frac{rvc}{hA})})\,dt = (\frac{1}{(T_e - T)})\,dT\ ;(\frac{1}{(t\ )})\,dt = (\frac{1}{(T_e - T)})\,dT\ ;Where : (t\ ) = (\frac{rv}{hA}$$

For a system where the chamber environment is at a constant temperature different from that of the UUT, the initial boundary condition at t=0 are Te=Constant and T_{UUT}=Ti. Integrating Equation 3, with

$$T = T_e - [(T_e - T_i)* e^{(\frac{-t}{t})}]$$

the initial conditions presented, yields the following:
Where T_e=Temperature of Chamber Environment
T_i=Initial UUT Temperature

Equation 6 suggests the following: the response, T, of a device that is thermally driven by a specific environment as described by Te=Constant…Ti, will follow an exponential form whose growth rate magnitude is a function of 1) the difference between the initial UUT temperature and the temperature of the surrounding environment [Te-Ti] and 2) the time constant τ. The negative power of the natural number e in Equation 6 appears upon integrating Equation 3 with the initial and final boundary conditions; Equation 6 displays the classic form of exponential decay.

Figure 3. Typical Thermal Response for an Environment of Te=100, Ti=0 as Time Constant τ Varies

Figure 3. shows the family of curves generated by Equation 4 with Te= 100EC, Ti=0EC and a driving temperature (Te-Ti)=100EC as τ varies at 10, 40 and 80.

Additionally, Equation 6, $T=Te-[(Te-Ti)\$e^{-t/\tau}]$, suggest that when a UUT is subjected to a constant thermal step change (Te=C), then τ is the time required for the temperature difference between the environment and the element to be reduced to 1/e of the initial difference and 1 - 1/e = 0.63212. Therefor, τ is the time it takes for the UUT to reach 63.212% of the initial driving temperature difference. The longer time it takes for the UUT to transition to 63.212% of the driving temperature differential, the greater the value of τ.

This method of immersing a device into an environment of differing temperature to determining the time constant τ is useful when attempting to predict the response of a UUT under a variety of driving thermal produces. The same time constant that is derived for a step change is also applicable for a driving ramp change or a sine wave like profile. In the interest of brevity, the mathematics will be foregone to present the driving profiles and the governing equations.

$$\left(\frac{1}{t}\right) dt = \left(\frac{1}{(T_e - T)}\right) dt$$

Starting with the basic differential equation:

Table 2 presents some different driving produces and the corresponding governing equation:

Profile Descriptions	**The Described by**	**UUT Response**
Ramp	R=dT/dt	$T=Te-R\tau$
Sine Wave	$T_2 \sin(\omega t)$, Where T_2=Max Chamber Temperature ω=Frequency of Oscillations [radians/time]	Will lag Environment by phase angle θ: Where: $\theta_{lag} = Tan^{-1}(\omega\tau)$ $t_{lag}=Tan^{-1}(\omega\tau)/\omega$ $T_{max}=[1+(\omega\tau)^2]^{-2}$

Table 4 Governing Equations for Ramp and Sine Thermal Profiles

A method for determining the thermal time constant for a device under thermal test has been presented. The method of experimentally determining the thermal time constant for a particular Unit Under Test can prove to be a useful tool to confirm, and when applicable, adjust the test parameters to maximize the product throughput without compromising test effectiveness.

1. Incropera, Frank P., and David De Witt., *Fundamentals of Heat and Mass Transfer.*, 1985, John Wiley & Sons, Inc.. Pg. 2.

2. Haberman, William L., and James E. A. John., *Engineering Thermodymics*., 1980, Allyn and Bacon, Inc. Boston., Pg