Temperature

Heat Transfer and Air Flow Analysis
Environmental Chambers and Air Servos

Bemco manufactures a wide variety of Environmental Chambers and Portable Temperature Servo Conditioners that include air circulation, heating, and cooling systems. Standard temperature ranges vary from -184 to +538 C (-300 to +1000 F). Cooling systems include liquid nitrogen, and single stage or cascade mechanical refrigeration systems.

Description:

The tight interrelationship between air flow and temperature uniformity in Environmental Chambers is discussed in this Technical Paper.

Techniques for calculating "order of magnitude" values for required air flow, heating, and cooling systems to achieve specified temperature uniformities, and heating or cooling rates are discussed.

Examples of both properly designed and systems with problems are given.

Bemco manufactures environmental facilities to simulate Temperature, Humidity, Altitude, Vibration, Vacuum, Rain, Sunshine, Salt Spray, and Sand and Dust conditions. We also offer Space Simulation Systems, Walk-in Chambers, Drive-in Rooms, PAO Fluid Chillers, and Portable Air Servos.

Celebrating our 55th year, Bemco continues to develop new products and skills. With the most experienced engineering and production staff in the industry, we are proud of the quality equipment we make.

Environmental Test and Space Simulation Systems

Contact us for a free quotation or additional information.

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Bemco Chamber Design

Purpose

This document is intended as an aid in specifying a Bemco chamber that includes temperature conditioning or for use in selecting a Bemco Portable Temperature Servo conditioner (PTS) matched to your insulated enclosure.

Since we specialize in this type of work, we have automated programs that perform a very rigorous steady state and transient thermal analysis of both your test load and our environmental apparatus.

We are happy to provide this service to you at no charge. The information presented below is for your use in preliminarily analyzing a prospective system concept.

Many of the more esoteric aspects of heat transfer and system response that we routinely evaluate on your behalf are simplified so that a reasonably short explanation is possible.

Air Flow Calculations

The primary heat transfer media in an ambient pressure environmental chamber without a conductive baseplate, is air.

At sea level, air is composed of nitrogen (78%), Oxygen (20.9%), Argon (0.9%), Carbon Dioxide (0.03%) and trace gasses. On a weight basis, air is normally considered to be 75.5% nitrogen and 23.2% oxygen.

This composition has a specific heat of approximately 0.24 Btu/F-lb and a specific volume of 13.33 ft^3/lb, at sea level, at an ambient temperature of 68 F.

Specific heat is defined as the amount of energy (in British thermal units) required to raise a unit mass (1 pound), a unit of temperature (1 degree F).

Specific volume is similarly defined as the volume a substance occupies (ft^3) per unit mass (1 pound). You will quickly notice that specific volume is actually the reciprocal of density.

Combining these pieces of information with the "General Gas Law," a blending of "Charles' Law" (volume 1 divided by temperature 1 = volume 2 divided by temperature 2) and "Boyle's Law" (pressure 1 times volume 1 = pressure 2 times volume 2) we have:

$$P_1 * V_1 / T_1 = P_2 * V_2 / T_2$$

where:

P = pressure, psia or Torr

V = volume, ft3

T = temperature, degrees F

dT= thermal rise in the loop, degrees F

and:

degrees R = degrees F + 459.67

1 Atmosphere = 14.6956 psia

1 Atmosphere also = 760 Torr

Btuh = Btu/hr

cfm = ft3/minute

With this information we can derive a very useful relationship:

First, at sea level, at standard temperature (68 F), and pressure (14.6959 psia or 760 Torr):

where:

C_p = Specific Heat, Btu/F-lb

Wg = mass, lb

p = density = Wg/volume = lb/ft^3

1/p = Specific Volume, ft^3/lb

q = gas flow, ft^3/minute

Then solving for Btuh/F -cfm we have:

$$\text{Btuh/F-cfm} = C_p * 60 * (1 \text{ ft3/min} / 1 \text{ ft3/min}) * p$$

in numbers:

0.24 btu/F-lb * 60 minutes/hour * (1 ft^3/minute) / (1 ft^3/minute) / 13.33 ft^3/lb = 1.0803 Btuh/F-cfm

The number **1.0803 Btuh/F-cfm** is extremely useful in air flow calculations.

This number, when combined with the "General Gas Law" can quickly be rearranged into a very useful equation:

$$q * dT = \sum Q / (1.0803 \text{ Btuh/F-cfm}) * (T_2 + 459.67) / (68 + 459.67) * (P_2 / 14.6956) \text{ with } P_2 \text{ in psia}$$

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You can read this formula as:

Air flow in cfm times thermal rise in degrees F = total thermal load in the air loop in Btuh divided by 1.0803 times a correcting factor for the absolute temperature in the workspace in degrees R (Rankin) and times a correcting factor for the absolute pressure in psia.

This formula can be rearranged so that, if the answer is divided by the selected air flow, the result is the total thermal loop rise in degrees F.

It can also be rearranged so that, if the answer is divided by the total allowed thermal loop rise in degrees F, the result is the required air flow in cfm.

It is sometimes more convenient to rearrange the same equation so that P2 is in Torr, a unit of absolute pressure. The value of q * dT is the same in either case provided the equivalent pressures are used.

$q * dT = \sum Q / (1.0803 \text{ Btuh/F-cfm}) * (T_2 + 459.67) / (68 + 459.67) * (P_2 / 760)$ with P_2 in Torr

The key conclusion is:

Temperature uniformity in an environmental chamber is directly related to the total thermal load present and the amount of air circulated.

For this reason, all Bemco environmental chambers include relatively high volume air circulation systems matched to their thermal systems.

Thermal Calculations

Within an environmental chamber, thermal loads can be divided into two broad classes, steady state and transient.

Steady state loads:

Steady state thermal loads, or thermal loads that remain after complete thermal stabilization, are test specimen thermal dissipation, air movement friction, and enclosure and penetration thermal losses.

Test Specimen Dissipation are thermal loads caused by such items as heaters, heat dissipating devices, operating motors, and so on. Normally expressed in Watts, live loads in Btuh can be calculated by converting Watts to Btuh where:

Q_L = Test Specimen Dissipation, Btuh

L = Live load, Watts

Conversion is 1 Btuh = 3.41214 Watts

$Q_L = L * 3.41214$

Live loads can be either positive or negative. An example of a positive load is an operating heater used to maintain a minimum temperature inside an electronics package. An example of a negative load is the presence of fluid or gas passages in a test item cooled by an external chiller such as a Bemco PCL system or a portable air servo such as a Bemco PTS system.

Outside Air, Gas, or Fluid Flows into the workspace that carry air, gas, or fluid into and out of the chamber or your test load during steady state or transient temperature conditions can add or subtract from the chamber steady state load depending on chamber conditions and the temperatures of supply.

Man rated walk-in rooms often require the continuous injection of a small flow, typically 15 scfm, of fresh air per person. This air can enter the chamber workspace at ambient temperature and humidity creating a substantial thermal load for a walk-in chamber's conditioning system.

For example, if 50 scfm of air is drawn from outside a building, on a warm very humid day, where the outside conditions are 90 F and 70% RH, and injected into a walk-in test chamber holding at -20 F, approximately 3127 Btuh of energy must then be removed by the chamber's conditioning systems.

Since analysis of these types of energy flows are somewhat complicated, they are not covered in this document. If your test procedure involves this type of process, make sure that you bring this information to the attention of your Bemco Applications Engineer so that their impact can be evaluated.

Fan Air Friction is the thermal load created by inefficiency in the fan, movement of air within the conditioning system, over the test load, and within the workspace. Normally expressed in horsepower, it varies directly with air density. Fan load in Btuh can be calculated by converting horsepower to Btuh where:

Q_f = Fan Air Friction, Btuh

Hp = Fan Friction, horsepower

T_2 = Operating temperature, F

V_2 = Operating pressure, psia

Conversion is 1 horsepower = 2544.434 Btuh

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Q_f = Hp * 2544.434 * [(68 + 459.67) / (T_2 + 459.67)] * (V_2 / 14.6956)

Notice that Q_f increases as temperature goes below 68 F and increases as pressure goes above 14.6956 psia. This is because density increases as temperature goes down and increases as pressure goes up.

The word fan is meant as a general classification for air movers. Bemco uses both axial fans (bladed propellers) and forward curved, backward curved, and cast blowers to move air within various models. Axial fans are generally more efficient at moving large volumes of air but blowers are better when higher pressure drops are present in the air loop (with the exception of vane axial fans in tubes).

Thermal Transmission includes losses through the chamber walls, the thermal breakers around the door face, access sleeves, the fan motor shaft(s), and any conductive test load penetrations such as tubes or wires.

Thermal transmission is normally calculated in Btuh. A useful computation, that provides an approximate answer for thermal losses in an environmental test chamber is:

Q_{tw} = k/x * Ao * (T_2 - $T_{ambient}$)

where:

k = conductivity of the insulation = Btuh-in/F-ft^2

For fiberglass k = 0.27 Btuh-in/F-ft^2

For urethane foam k = 0.15 Btuh-in/F-ft^2

For stainless steel k = 105 Btuh-in/F-ft^2

Ao = Area Outside, ft^2

T_2 = Test Temperature, F

$T_{ambient}$ = Ambient Temperature, F

Note that $T_{ambient}$ is measured directly outside the chamber case.

For test chamber walls, a reasonable simplification is to take the outer wall area of the entire case including the conditioning section and the door face as the area. With the outside dimensions for height, width and depth (H, W, and D) of the insulated case of the environmental chamber given in inches , this computation is:

Ao = (2 * H * W + 2 * H * D + 2 * W *D) / 144 in^2/ft^2

Ao = ft^2 Outside Area

The chamber door thermal breaker typically adds a significant amount of additional losses due to its solid conductive path.

For the thermal breaker, a reasonable simplification is to take the outside door perimeter as the length and twice the thickness of the breaker as the width. Assuming that the door is on the front of the chamber, where

t = Breaker thickness, inches

and

Typical breakers are fabricated from 20 gauge or 18 gauge, 304 series stainless steel as well as nonmetallic materials like epoxy-glass laminate, not covered here.

For the stainless steels:

20 Gauge = 0.035 inches

18 Gauge = 0.048 inches

Then this computation is:

Ab = [(2 * H + 2 * W) * t *2] / 144 in^2/ft^2

and

Q_b = k/x * Ab * (T_2 - $T_{ambient}$)

To compute the breaker thickness value, x in the computation k/x, you must measure the distance from the inside of the chamber to the outside of the chamber along the breaker path.

Plug doors have longer breaker thickness values than flat doors and have an advantage in thermal losses. The above calculation ignores air thermal convection heat transfer in the gasket gap. Wide range Bemco chambers always have dual gaskets to mitigate thermal convection in this space.

Thermal Calculation Example

The Bemco wide range F, LDF, FTU, and FW chambers utilize plug type doors to increase the thermal breaker length. A side benefit of this type of design is that it also eliminates warpage caused by door expansion at high temperatures.

A detail of this type of construction on a Bemco F27-73/177C chamber case is shown on the top of the next page. The plug door on this chamber has two gaskets, a thickness of 6 inches, an 18 gauge stainless steel thermal

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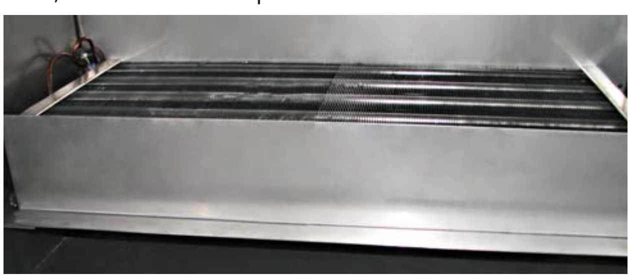
breaker, and fiberglass insulation. Even though the wall is 6 inches thick, the actual thermal breaker length, because of Bemco's standard door plug, is 9 inches.

This model chamber features a 12 inch diameter welded stainless steel fan that is factory pre-balanced for smooth operation at its designed speed of 1725 rpm. This fan is rated to produce 1288 cfm of air flow at a pressure drop of 0.25 inches of water, the design pressure drop in the Bemco F27-73/177C. When running at this pressure drop, the fan uses 0.140 horsepower.

The fan is shown without the standard Bemco machined stainless steel hub. In the chamber, it is vertically mounted to minimize fan shaft bending and is driven by a 1/3 horsepower externally mounted motor with an extended stainless steel shaft.

Workspace on a Bemco F27-73/177C is 36 inches high by 36 inches wide by 36 inches deep, or 27 cubic feet. The overall dimensions of the F27-73/177C are 50 inches high (there is 8 inches of insulation on the floor), 48 inches wide, and 63 inches deep.



The conditioning plenum at the rear of the chamber is 14 inches deep while the cooling coil, behind the baffle is 12 inches deep. The difference in depth is to allow the placement of a second baffle over the front of the coil to prevent the coil from coming in contact with the air separation baffle, preventing a cold spot from forming on the baffle.

With this information and the equations we have previously discussed, we can calculate the steady state thermal performance of this chamber.

Starting with fan horsepower, we know that that chamber has a temperature range of -73 C to 177 C or -100 F to +350 F. We can assume that ambient temperature is 68 F. For the purposes of this calculation we will evaluate the performance at sea level.

Q_{F-100} = 0.140 Hp * 2544.434 * [(68 + 459.67) / (-100 + 459.67)] * (14.6956 / 14.6956) = $\underline{522.6 \text{ Btuh}}$

Q_{F+350} = 0.140 Hp * 2544.434 * [(68 + 459.67) / (350 + 459.67)] * (14.6956 / 14.6956) = $\underline{232.2 \text{ Btuh}}$

Max. Fan Horsepower = 0.140 Hp * [(68 + 459.67) / (-100 + 459.67)] * (14.6956 / 14.6956) = 0.205 Hp @ -100 F

Note that since friction heats, the fan heat at high temperature results in a reduction in the amount of heat that must be added to maintain process conditions.

For wall losses on the Bemco F27 model with the same ambient temperature:

Ao = ((2 * 50 * 48) + (2 * 50 * 63) + (2 * 48 * 63)) / 144 ft^2

Ao = 119.1 ft^2

Wall thickness x = 6 inches

Conductivity = 0.27 Btuh-inch/F-ft^2 (fiberglass)

Q_{tw-100} = 0.27 / 6 * 119.1 * (-100 - 68) = $\underline{900.4 \text{ Btuh}}$

Q_{tw+350} = 0.27 / 6 * 119.1 * (+350 - 68) = $\underline{1511.4 \text{ Btuh}}$

For thermal breaker losses on the Bemco F27 model with the same ambient temperature:

Ab = (2 * 50 + 2* 48) * (0.048 * 2) / 144 ft^2

Ab = 0.131 ft^2

Breaker Length x = 9 inches

Conductivity = 105 Btuh-inch/F-ft^2 (18 gauge stainless)

Q_{bw-100} = 105 / 9 * 0.131 * (-100 - 68) = $\underline{256.8 \text{ Btuh}}$

Q_{bw+350} = 105 / 9 * 0.131 * (+350 - 68) = $\underline{431.0 \text{ Btuh}}$

The total thermal losses at -100 F are therefore 522.6 Btuh for the fan plus 900.4 Btuh for the chamber walls plus 256.8 Btuh for the thermal breaker around the door. The sum of which is 1679.8 Btuh.

In the same manner, **the total thermal losses at +350 F** are minus 232.2 Btuh for the fan plus 1511.4 Btuh for the

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chamber walls plus 431.0 Btuh for the thermal breaker around the door. The sum of which is 1710.2 Btuh.

Remembering the number 1.0803 Btuh/F-cfm and correcting for both temperature and pressure, the maximum temperature gradient at +350 F and -100 F where:

$q * dT = \sum Q / (1.0803\ \text{Btuh/F-cfm}) * (T_2 + 459.67) / (68 + 459.67) * (P_2 / 14.6956)$ with P_2 in psia

and

$\sum Q_{+350}$ = 1710.2 Btuh
$\sum Q\text{-}_{-100}$ = 1679.8 Btuh
q = 1288 cfm

Therefore:

$dT_{+350} = (1710.2 / 1.0803 * (350 +459.67) / (68 +459.67)) / 1288$

dT_{+350} = 1.89 F or 1.89 F / 1.8 C/F = **1.05 C**

$dT_{-100} = (1679.8 / 1.0803 * (-100 +459.67) / (68 +459.67)) / 1288$

dT_{-100} = 0.82 F or 0.82 F / 1.8 C/F = **0.46 C**

At this point in our calculation we know that the Bemco F27-73/177C Wide Range Temperature Chamber is designed for a gradient of 1.05 C total at 177 C and 0.46 C total at -100 C. We also know that it circulates 1288 cfm.

Since we also know that the workspace is 36 inches wide by 36 inches deep we can compute the average air velocity in the workspace at:

Cross Sectional Area = 36 * 36 / 144 = 9 ft^2
Average Air Velocity = 1288 ft^3/minute / 9 ft^2
Average Air Velocity = 143.11 ft/minute

Looking at the bulletin for the wide range F (please request a copy from your Bemco representative), the rated live load holding capacity is 1580 watts at -55 C (-67 F).

Re-running the calculations for -67 F:

Q_{F-67} = 0.140 Hp * 2544.434 * [(68 + 459.67) / (-67 + 459.67)] * (14,6958 / 14.6956) = 478.7 Btuh

Q_{tw-67} = 0.27 / 6 * 119.1 * (-67 - 68) = 723.5 Btuh

Q_{bw-67} = 105 / 9 * 0.131 * (-67 - 68) = 206.3 Btuh

$\sum Q$ = 1408.67 Btuh + Live Load of 1580 Watts * 3.41214 Btuh/Watt = 6799.9 Btuh

$dT_{-67} = (6799.9 / 1.0803 * (-67 +459.67) / (68 +459.67)) / 1288$

dT_{-67} = 3.64 F or 3.64 F / 1.8 C/F = **2.02 C**

We now know that the theoretical average gradient in the Bemco F27-73/177C Wide Range Temperature Chamber, with the maximum rated heat load in place in the workspace, is 3.64 F total or 2.02 C total. This number is a distributed average and you should expect local hot spots around your test load where heat is being generated.

A useful approximate equation for calculating the forced convection heat transfer coefficient on the surface of a plane surface in air is given in the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) in their Handbook Chapter on Heat Transfer as:

$h = 0.5\ (V)^{0.8}$

h = Btuh/F-ft^2 (Heat Transfer Coefficient)

Where V (Velocity of Air) is 16 to 100 ft/second (fps)

Note that 16 ft/second = 960 ft/minute is a velocity that is much higher than normal in an environmental chamber's workspace. If the standard chamber we are discussing, the Bemco F27, had a 16 fps average velocity in the workspace

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a fan generating 16 fps * 9 ft2 face area = 8,640 cfm would be required.

High velocity and high air volume systems are available from Bemco for simulating vehicle movement. Maximum velocities for these types of systems are in the range of 120 miles per hour or 10,560 ft/minute and 176 ft/second.

We have already noted that the average air velocity in the Bemco F27 is 143.11 ft/minute.

Converting this velocity to ft/minute:

$V = 143.11$ ft/minute / 60 seconds/minute = 2.39 fps

$H = 0.5 * (143.11 / 60)0.8 = 1.002$ Btuh/F-ft^2

Since the actual average velocity in the chamber is less than 16 fps, another simplified equation offered for forced convection (forming the lower limit of consideration) is:

$H = 0.99 + 0.21 * V = 0.99 + .21 * 2.39 =$ **1.49 Btuh/F-ft2**

Realizing that the lower velocity equation yields the larger number, it is safe to assume that the heat transfer coefficient inside the chamber is at least 1.49 and may be higher at points due to induced blockage by the placement of the test item.

Surface heat transfer coefficients can be difficult to calculate. In general, they vary with the velocity, specific heat, viscosity, and conductivity of the gas or fluid and the shape of the space they are passing over or through. They also vary sharply when flow is smooth rather than turbulent.

Transient loads:

Transient thermal loads are computed based on the thermal energy required to change the chamber, the air within the chamber, the chamber conditioning equipment, the shelves or product supports, and your test items between the starting temperature and the ending temperature of your test cycle. At the same time, the "Steady State Loads" previously computed vary with time.

Computations of this type are fairly complicated and in most cases, should be made by your Bemco Applications Engineer using information provided by you.

For **Transient Thermal Analysis, the general equation used is:**

$Q_{tr} = \sum CpWg * dT/d\Theta + Q_{ss}$

Where:

Q_{tr} = Transient Load, Btuh

C_p = Specific Heat, Btu/F-lb

W_g = Mass due to gravity, pounds (lb)

dT = Change in Temperature, F

$d\Theta$ = Change in Time, hours

$\sum CpWg$ = Sum of all Cp * Wg for all components that make up the load.

As a complication, please note that all loads do not necessarily track together, since their surface heat transfer rates vary, their density concentration varies, their conductivity varies, and their temperatures as the transition proceeds vary. The density concentration refers to how homogeneous the material is versus its exposed air side surface area and mass. These factors are not covered in this explanation but can be evaluated by your Bemco Applications Engineer.

A short but useful list of C_p (Specific Heat) in Btu/F-lb, density, and conductivity is given in a table on the last page of this document.

As noted above, **heat transfer to a test object is limited by the available surface area versus the test object's mass**. The maximum energy that can be transferred per unit time is given by the general equation:

$Q_h = H_o * A_o * (T_s - T_{ca})$

Where:

Q_h = Surface Heat Transfer, Btuh

H_o = Surface Heat Transfer Coefficient, Btuh/F-ft^2

A_o = Surface Area, ft2

T_s = Average Test object surface temperature, F

T_{ca} = Average Test chamber air temperature, F

As an example of the use of the Transient Thermal Analysis equations, assume that your test load weighs 100 pounds, is made from steel, is a closed hollow box with air inside and has an outside surface area of 10 ft^2. You want to cool this box from 68 F to -67 F in 60 minutes. How much energy will it take?

Looking at the table on the last page, we know:

Steel $C_p = 0.12$ Btu/F-lb

Load $W_g = 100$ lb

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$C_pW_g = 0.12 * 100 = 12.0$ Btu/F

$dT = 68$ F - (-67 F) = 135 F

$d\Theta = 60$ minutes / 60 min/hr = 1 hour

$Q_{tr} = 0.12 * 100 * 135 / 1 =$ 1620 Btuh

We also want to know if there is enough surface area to exchange this amount of heat at the rate we need so that our load tracks the chamber's air temperature.

p = Steel density = 490 lb/ft^3

Our unit has an area of 10 ft^2:

100 lb / 490 lb/ft^3 * 1728 in3/ft^3 = 352.65 in^3

352.65 in^3 / (10 ft^2 * 144) in^2/ft^2 = 0.24 inches thick

$A_o = 10$ ft^2

We know from our previous calculation that in an F27 the outside heat transfer coefficient is at least:

$H_o = 0.99 + 0.21 * V = 0.99 + .21 * 2.39 = 1.49$ Btuh/F-ft^2

Therefore, evaluating:

$Q_h = H_o * A_o * (T_s - T_{ca})$

for the value Q_h = 1635 Btuh, with a little algebra, we have:

$(T_s - T_{ca})$ = 1620 Btuh / (10 ft2 * 1.49 Btuh/F-ft^2)

$(T_s - T_{ca}) = 108.72$ F

What does this answer mean? It tells you that the test load you have selected will probably not track the temperature change you have in mind since it does not have enough surface area. In fact, it will not start to track at the rate you have in mind until the temperature difference exceeds 108.72 F.

If you again look at the equation for Transient Thermal Analysis and think about the added term Q_{ss}

$Q_{tr} = \sum CpWg * dT/d\Theta + Q_{ss}$

you will quickly realize that if your test load dissipates heat while cooling in excess of the amount of heat that can be carried away by convective heat transfer:

$Q_h = H_o * A_o * (T_s - T_{ca})$

then your test object will actually heat rather than tracking the cooling offered by the air circulating within the environmental test chamber.

Bemco LDF, F, and FTU Series Wide Range Temperature Chambers are available with a number of custom or upgraded conditioning systems, air flow patterns, and test fixturing modifications to match your needs. Bemco Applications Engineers can help you evaluate your testing problem. Assistance of this type is free.

Testing Specifications

Many military and commercial specifications contain requirements with respect to temperature control and temperature uniformity. For example:

MIL-STD-810F Says:

Department of Defense Test Method Standard for Environmental Considerations and Laboratory Tests.

5.2 Tolerances of Test Conditions.

Unless otherwise specified, adhere to the test condition tolerances shown below for the following parameters. . .

a. Test section air temperature. Surround the test item totally by an envelope of air (except at necessary support points). To ensure that the test item is bathed in the required air temperature, place verification sensors at representative points around the entire item and as close to the test item as possible but not so the airstream temperature is affected by the test item temperature. Keep these temperatures within + or - 2 C (3.6 F) of the required test temperature. Ensure the air temperature gradient across the item does not exceed 1 C (2 F) per meter or a maximum of 2.2 C (4 F) total (test item non-operating). Wider temperature tolerances are acceptable in situations such as:

(1) For large items with a volume greater than 5 m3, the temperature tolerance can be + or - 3 C. Justify any larger tolerance and obtain approval for its use from the procuring activity.

(2) For required temperatures greater than 100 C, the temperature tolerance can be + or - 5 C. Specify the actual tolerance achieved.

Similar statements can be found in many other Military Specifications.

In Method 507.3 of MIL-STD-810 on Humidity, there is a paragraph (II-1.1b) that says: "The flow of air anywhere within the envelope of the air surrounding the test item

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shall be maintained between 0.5 and 1.7 meters per second (98 and 335 ft/min)."

Recalling our previous analysis of the Bemco F27-73/177C Wide Range Temperature Chamber, we know that it is designed for a gradient of 1.05 C total at 177 C and 0.46 C total at -100 C. We also know that it circulates 1288 cfm and that its Average Air Velocity is 143.11 ft/minute.

We also know that with a test load dissipating its maximum live load, that the theoretical average gradient in the Bemco Chamber is 3.64 F total or 2.02 C total.

All of these numbers meet or exceed the requirements of all known Military Specifications. ***Unfortunately, many manufacturer's similarly sized equipment do not meet these Military or commercial specifications.***

We strongly advise our clients to carefully analyze any offer of equipment by asking detailed questions to allow an understanding of the actual air circulation volume and conditioning systems included in any chamber considered for use in testing to rigorous specifications.

In our case, our motto: "Bemco chambers really simulate the environments expected" is the literal truth. Every product we make is carefully designed, checked for performance, and backed by our 24 hour a day, 7 day a week, service group.

A Contrasting Story

As a cautionary tale, we offer the following example for your consideration. We recently received two identical competitors machines for repair of their refrigeration systems.

We often accept other competitor's products for upgrade or modification. In this case the units were sent in for modification of the refrigeration system since they were originally manufactured without the ability to modulate cooling capacity, instead depending on the "bucking of heat" to overcome a continuously operating cascade refrigeration system.

The original owner wanted to add the ability to program. We added automatic hot gas bypass and suction cooling to the low stage and replaced the low stage refrigerant with a more modern and environmentally friendly fluid.

Although the original manufacturer of these systems is a major company in our industry, they will remain nameless, since our comments are unfavorable. In their defense, theses chambers were made in mid 1980's. Hopefully, their current designs have corrected the performance and quality deficiencies noted.

The primary reason for offering this information is to sensitize our customers to potential problems with other competitor's designs and to illustrate the value of the calculation techniques outlined in this document.

The two environmental chambers in question have a workspace of 30 inches high by 30 inches wide by 30 inches deep or 16 cubic feet. Insulation is 6 inches of fiberglass on the door and walls and the thermal breaker is made from 20 gauge stainless steel, 6 inches long.

Outside dimensions are 42 inches high by 42 inches deep by 50-1/2 inches deep.

The real problem begins with the airflow system. These chambers are equipped with two 7 inch diameter, pressed, plated carbon steel fans driven by externally mounted motors turning at 3000 rpm.

These fans are rated at 365 cfm at 0.25 inches of water pressure drop when turning at 3000 rpm. At these conditions they consume 0.03 horsepower each or a total of 0.06 horsepower. Total rated air flow in the workspace is therefore 730 cfm with two fans in operation.

Both chambers are equipped with cascade refrigeration systems (one refrigeration system cools another refrigeration system which in turn cools the workspace) of size and capability similar to a standard Bemco F16-73C/177C Wide Range Temperature Chamber.

Their previous problem with ability to modulate refrigerant flow has been corrected by our service department.

The evaporative cooling coil in these chambers is located behind a screwed down stainless steel baffle with the two fans almost in contact with the back wall. This is an

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unusual design since fan manufacturers normally recommend some clearance, in front of, and behind the blades to achieve their rated flow. Unfortunately, the entire air plenum depth on this chamber is 7-1/2 inches and a secondary air plenum and fan ring, housing the two fans, is only 2 inches deep.

Based on what we see, air flow must be de-rated from the hoped for 730 cfm, previously mentioned, to about 584 cfm, a reduction of 20%. This reduction is caused by a loss of efficiency due to their restricted mounting combined with an increase in pressure drop caused by their very close placement to the chamber inner rear wall which forces the circulating air to turn almost immediately.

Frankly, we have a similar problem with air flow reduction due to close placement of an axial fan on our FB1.5V, a one and one half cubic foot chamber where we also use a pressed fan due to the chamber's compact size. In our case we use an all aluminum fan to prevent corrosion.

A photo of our standard FB1.5 fan next to the fan temporarily removed from the other manufacturer's 16 cubic foot chamber is shown to the left. The smaller, 7 inch diameter, 4 blade fan is the one from the 16 cubic foot chamber.

Using the information provided above and the equations we have already discussed to analyze the 16 cubic foot chamber, referred to below as the COM16:

Starting with fan horsepower on the COM16

Q_{F-100} = 0.06 Hp * 2544.434 * [(68 + 459.67) / (-100 + 459.67)] * (14,6956 / 14.6956) = 224.0 Btuh

Q_{F+350} = 0.06 Hp * 2544.434 * [(68 + 459.67) / (350 + 459.67)] * (14.6956 / 14.6956) = 99.5 Btuh

Max. Fan Horsepower = 0.06 Hp * [(68 + 459.67) / (-100 + 459.67)] * (14.6956 / 14.6956) = 0.088 Hp @ -100 F

For wall losses on the COM16

Ao = ((2 * 42 * 42) + (2 * 42 * 50.5) + (2 * 42 * 50.5)) / 144 ft^2

Ao = 83.4 ft^2

Wall thickness x = 6 inches

Conductivity = 0.27 Btuh-inch/F-ft^2 (fiberglass)

Q_{tw-100} = 0.27 / 6 * 83.4 * (-100 - 68) = 630.5 Btuh

Q_{tw+350} = 0.27 / 6 * 83.4 * (+350 - 68) = 1058.4 Btuh

For thermal breaker losses on the COM16 model with the same ambient temperature:

Ab = (2 * 42 + 2* 42) * (0.035 * 2) / 144 ft^2

Ab = 0.082 ft^2

Breaker Length x = 6 inches

Conductivity = 105 Btuh-inch/F-ft^2 (20 gauge stainless)

Q_{bw-100} = 105 / 6 * 0.082 * (-100 - 68) = 241.1 Btuh

Q_{bw+350} = 105 / 6 * 0.082 * (+350 - 68) = 404.7 Btuh

The total thermal losses at -100 F on the COM16 are therefore 224.0 Btuh for the fan plus 630.5 Btuh for the chamber walls plus 241.1 Btuh for the thermal breaker around the door. The sum of which is 1095.6 Btuh.

In the same manner, **the total thermal losses at +350 F** are minus 99.5 Btuh for the fan plus 1058.4 Btuh for the chamber walls plus 403.0 Btuh for the thermal breaker around the door. The sum of which is 1363.6 Btuh.

Remembering the number 1.0803 Btuh/F-cfm and correcting for both temperature and pressure, the maximum temperature gradient at +350 F where:

q * dT = ΣQ / (1.0803 Btuh/F-cfm) * (T_2 + 459.67) / (68 + 459.67) * (P_2 / 14.6956) with P_2 in psia

and

ΣQ_{+350} COM16 = 1095.6 Btuh

$\Sigma Q\text{-}_{-100}$ COM16 = 1363.6 Btuh

q COM16 = 584 cfm

Therefore:

dT_{+350} COM16 = (1363.60 / 1.0803 * (350 +459.6956) / (68 +459.6956)) / 584

dT_{+350} COM16 = 3.32 F or 3.32 F / 1.8 C/F = **1.84 C**

dT_{-100} COM16 = (1094.68 / 1.0803 * (-100 +459.67) / (68 +459.67)) / 584

dT_{-100} COM14 = 1.18 F or 1.18 F / 1.8 C/F = 0.66 C

At this point in our calculation we know that the Competitor's Temperature Chamber (COM16) is designed for a gradient of 1.84 C total at 177 C and 0.66 C total at -100 C. We also know that it circulates about 584 cfm. Note the 1.84 C at 177 C as compared to the Bemco F27 where the similar number, previously calculated is 1.05 C.

Since we also know that the workspace on the COM16 chambers is 30 inches wide by 30 inches deep we can **compute the average air velocity in the workspace** at:

Cross Sectional Area = 30 * 30 / 144 = 6.25 ft^2
Average Air Velocity = 584 ft^3/minute / 6.25 ft^2
Average Air Velocity = 93.44 ft/minute

Note that, if these units were humidity chambers, they do not have enough velocity to meet the minimum air flow requirements of MIL-STD-810 paragraph (II-1.1b), where the minimum is 98 ft/minute. Usually requirements of this type are selected by the writers of Military Specifications to assure that air velocity is sufficient to assure reasonable heat transfer coefficients (Ho), but not so high that the test yields results that are not typical of real world conditions.

Importantly, the height, width and depth of the COM16 is 30 inches or 0.76 Meters cubed. On a similar basis the Bemco F27 is 36 inches or 0.91 Meters cubed. Another Bemco standard model, F16 has the same dimensions as the COM16.

The competitor's chambers we are discussing have the same horsepower refrigeration systems as both our standard Bemco F16-73/177C and our Bemco F27-73/177C Wide Range Temperature Chambers. The Bemco F27 is the Bemco chamber we have already analyzed. As refrigeration systems get smaller, there are fewer horsepower choices.

Both the Bemco F16 and F27 also utilize the same size air circulation system. This subsystem must be matched to the refrigeration system to remove the energy available. As a result, all of the performance numbers we have so far derived for the Bemco F27-73/177C are slightly better on the F16-73/177C, since it is smaller.

Referring again to the F series Technical Bulletin, both the Bemco F16 and F27 models are rated for a **live load holding capacity of 1580 watts at -55 C (-67 F)**. Applying this live load to the competitor's COM16 unit and re-running the calculations for the COM16 at -67 F we immediately see one of the problems with cutting the air flow by more than 50%:

Q_{F-67} = 0.06 Hp * 2544.434 * [(68 + 459.67) / (-67 + 459.67)] * (14.6956 / 14.6956) = 204.1 Btuh

Q_{tw-67} = 0.27 / 6 * 119.1 * (-67 - 68) = 506.7 Btuh

Q_{bw-67} = 105 / 9 * 0.131 * (-67 - 68) = 192.9 Btuh

$\sum Q$ = 903.7 Btuh + Live Load of 1580 Watts * 3.41214 Btuh/Watt = 6299.4 Btuh

dT_{-67} = (6299.4 / 1.0803 * (-67 +459.6956) / (68 +459.6956)) / 584

dT_{-67} = 6.81 F or 6.81 F / 1.8 C/F = **3.78 C**

Contrasting Story Conclusions:

Recalling that MIL-STD-810F requires that test . . ."temperatures (be) within + or - 2 C (3.6 F) of the required test temperature. Ensure the air temperature gradient across the item does not exceed 1 C (2 F) per meter or a maximum of 2.2 C (4 F) total (test item non-operating)." We see a problem with the COM16.

As temperatures get higher than 220 F the Competitor's COM16 chambers no longer meet the 1 C per meter gradient specification of MIL-STD-810F. The Bemco F27-73/177C system meets this specification over the full temperature range and the Bemco F16-73/177C system easily passes since it has the larger fan and cooling systems of the Bemco F27 in a 16 cubic foot chamber.

In a COM16 chamber, as heat is applied in the workspace and the air temperature approaches 1580 watts, the thermal rise in the air loop will approach 6.8 F or 3.8 C between the supply and return vents in the conditioning package making it progressively more difficult to achieve the conditions specified by MIL-STD-810F.

As illustrated in this example, one of the most important things to watch out for when selecting an environmental chamber for testing where air temperature uniformity is important is:

Sufficient air flow so that the energy lost through the chamber walls and thermal breaker and the energy added by any live loads within the chamber workspace

Bemco Chamber Design

are smoothly carried back to the chamber conditioning system.

Air flow volume is critical both for temperature uniformity and heat transfer between the test load and the air passing over it. On rapid response chambers, Bemco progressively increases air flow to assure proper heat transfer between the conditioning apparatus and your products.

Our claim of: "Bemco chambers really simulate the environments expected. We take your specifications and requirements literally. Our equipment does what we promise and you specify. We are truly focused on Excellence" is the actual truth. All of our equipment is carefully designed, tested for compliance with both your specifications and our promises, and supported by our conscientious service staff.

We are ready to help you with any questions you may have on our products or services.

Our assistance in preparing your specifications and analyzing your needs is free.

Material Properties for Heat Transfer

Test Item	Density	Density	Cp	Conductivity	Conductivity	Viscosity	Prandtl #
Material	lb/in3	lb/ft3	btu/F-lb	btu-in/hr-F-ft2	btu/hr-F-ft	lb/ft-second	None
Metals							
Aluminum	0.098	169.344	0.214	1540.000	128.33333		
Brass	0.308	532.000	0.092	672.000	56.00000		
Bronze	0.313	540.000	0.082	180.000	15.00000		
Copper	0.322	556.416	0.095	2680.000	223.33333		
Silver	0.379	655.000	0.056	2856.000	238.00000		
Steel	0.284	490.752	0.120	460.000	38.33333		
Stainless Steel	0.286	494.208	0.122	105.000	8.75000		
Non-Metallics							
Concrete	0.083	144.000	0.200	6.480	0.54000		
Delrin	0.051	88.128	0.350	1.600	0.13333		
Fiberglass Insulation	0.002	4.000	0.120	0.270	0.02250		
Glass	0.101	174.528	0.120	7.500	0.62500		
Phenolic	0.046	79.488	0.400	1.000	0.08333		
Polyethylene	0.035	60.480	0.550	2.300	0.19167		
Polystyrene	0.038	65.664	0.320	0.850	0.07083		
Rubber	0.044	76.032	0.440	1.100	0.09167		
Urethane Foam	0.001	2.000	0.300	0.150	0.01250		
Assemblies							
Electronics	0.069	120.000	0.300	5.000	0.41667		
Fluids and Liquid Solids							
Ice	0.033	57.000	0.460	15.360	1.28000		
Mercury 50 F	0.490	847.000	0.033	56.400	4.70000	0.00107	0.02700
Water, 32 F	0.036	62.400	1.010	3.828	0.31900	0.00120	13.67774
Water, 70 F	0.036	62.300	0.998	4.164	0.34700	0.00066	6.81286
Water, 200 F	0.035	60.100	1.000	4.728	0.39400	0.00021	1.87310

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