# Development of a Bench-Top Time Proportional Humidity Generator with Chilled Mirror Hygrometer Reference Standard

Jeff Hawkins & Ken Soleyn
GE General Eastern Instruments
500 Research Dr
Wilmington, MA 01887
(978) 203-1900
Ken.Soleyn@ge.com
Jeff.A.Hawkins@ge.com

## Abstract

*Divided-flow volumetric mix ratio techniques for the creation of controlled humidity environments have been in use for many years. These techniques typically operate under flow rate control whereby a continuous stream of saturated air (wet) is mixed in selected proportion with a continuous stream of dry air. The resultant mixed air stream embodies a partial water vapor pressure somewhere between zero and the saturation vapor pressure, dependent upon the selected volumetric mixing ratio. The design approach for this bench-top humidity calibration chamber utilizes volumetric mixing of dry air and saturated air via time proportioning of a three-way solenoid activated valve. An integrated NIST traceable chilled mirror hygrometer provides humidity measurement for feedback control of the test environment via microprocessor controlled time proportioning of the wet and dry air streams. System design considerations and validation data chronicling the chamber's accuracy, stability, and response time are presented.*

## 1. Introduction

GE General Eastern has commercialized several products utilizing the time proportioned divided-flow technique as described in United States Patent 5,056,547. The humidity generator illustrated in this paper is the latest embodiment of this technology and differs from previous design iterations as follows:

- Increased chamber volume over a previous "portable" humidity generator.
- Fully integrated chilled mirror hygrometer and precision temperature reference standard.
- Self-contained apparatus for generating a dry air supply.
- Integrated heat exchanger for temperature control and stabilization.
- Material and component selection for enhanced corrosion prevention, improved response time, and long-term reliability.
- Enhanced system ergonomics and user interface including "ramp and soak" profile programming.

## 2. Theory of Operation

Divided-flow volumetric mix ratio humidity generation typically operate under flow rate control whereby a continuous stream of saturated air (wet) is mixed in selected proportion with a continuous stream of dry air. The resultant mixed air stream embodies a partial water vapor pressure somewhere between zero and the saturation vapor pressure, dependent upon the selected volumetric mixing ratio. In the described system, the dry air contains a partial water vapor pressure (Vp) essentially equal to zero, while the saturated (wet) air stream has a partial water vapor pressure equal to the saturation vapor pressure (Vpsat) at a specific temperature. The resultant mixture of the wet and dry air becomes a mixed air stream containing a partial water vapor pressure somewhere between zero and Vp(sat), dependent on the ratio of the mixed amounts. If the mixed gas stream is maintained at the saturation temperature of Vp(sat), the percent relative humidity (%RH), of the mixed gas streams can be calculated as follows:

(1) $$\%RH = \frac{fd \times Vp(dry) + fw \times Vp(sat)}{(fd + fw) \times Vp(sat)} \times 100$$

*Where:*
*fd= flow rate dry air*
*fw = flow rate saturated air*
*Vp= partial water vapor pressure, dry*
*Vp(sat) = partial water vapor pressure, wet*

Assuming the dry air stream is devoid of moisture, (Vp=0, %RH=0), equation (1) can be simplified to:

(2) $$\%RH = \frac{fw}{(fd + fw)} \times 100$$

It is important to note that equation (2) remains valid in a dynamic system only if the dry air remains "completely dry" (0% RH), the wet air remains "completely saturated" (100% RH), no water vapor is added or subtracted from the gas stream after mixing, the relative flow rates of the wet and dry air streams remain constant, and the pressure and temperature of the mixture remains the same as the pressure and temperature of saturation of the wet air stream.

As a further refinement, relative humidity is defined as:

(3) $$\%RH = 100 \times {e}/{e_s}$$

*Where:*
*e = water vapor partial pressure*
*$e_s$= water vapor saturation pressure*

The water vapor partial pressure, e, is given as:

(4) $$e = 6.1121e^{\left(17.502Td/240.9+Td\right)}$$

*Where:*
*Td = dew point temperature, °C*

The water vapor saturation pressure, $e_s$, is given as:

(5) $$e_s = 6.1121e^{\left(17.502T/240.9+T\right)}$$

*Where:*
*T = gas dry bulb temperature, °C*

In considering equations (3), (4) and (5), it is apparent that relative humidity is both a function of the water vapor partial pressure and water vapor saturation pressure and is therefore dependent on the dry bulb temperature of the gas as well as its moisture content. As such, temperature stabilization becomes an important design attribute of the system.

## 3. System Overview

In practice, precise control of flow rates, water vapor content of pre-mix streams, saturator and test chamber temperatures and pressures can be quite complex and costly. In order to overcome these inherent variances of the system, closed-loop microprocessor based control of the test chamber environment is employed.

In this arrangement, the imbedded chilled mirror and dry bulb temperature sensors continuously monitor the relative humidity levels within the test environment. This reference data is then utilized by the humidity generator control system to adjust the wet/dry gas mix ratio via time-proportioned duty cycling of a three-way solenoid valve. The system can be described in more detail with the assistance of the schematic shown in Figure 1.

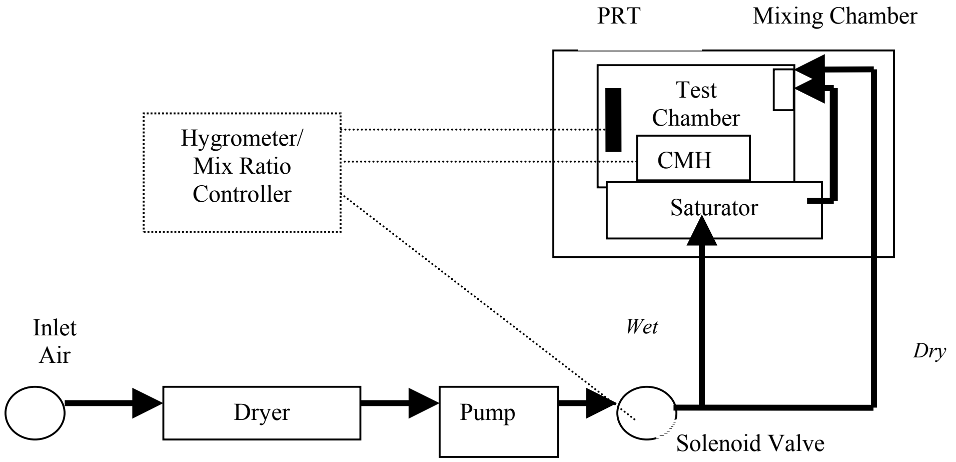


Figure 1- System Schematic

Inlet air is drawn through the air dryer via a diaphragm pump, creating a dry air source. This dry air stream is then introduced into a three-way solenoid valve, where the air stream is split into two proportioned streams: (1) dry air for introduction into the mixing chamber and; (2) dry air for introduction into the mixing chamber through the saturator, thereby creating the wet air stream. After passing through the mixing chamber, the mixed air stream enters the test chamber where it comes in direct contact with the chilled mirror hygrometer (CMH) and dry bulb temperature sensor. These sensors provide input into the mix ratio controller, which adjusts the duty cycle of the 3-way solenoid valve, thereby providing closed-loop control of the test chamber relative humidity.

## 4. Design Considerations

### *4.1 Dry Air Supply*

In order to provide a consistent flow of dry air with a relative humidity of “0% RH”, the inlet air for the system is drawn though a desiccant cartridge by the pump prior to introduction into the mixing chamber and/or saturator. The system’s dryer cartridge contains 560 grams of calcium sulfate ($CaSO_4$) desiccant impregnated with cobalt chloride ($CoCl_2$) as a visual indicator of moisture content.

At the system design flow rate, the calcium chloride desiccant provides conditioned air containing .005mg of water per liter of air (.022% RH at 25°C and 1 ATM pressure). This desiccant adsorbs approximately 8.8% of its own weight in water, about 50 grams for the system. When operating in an ambient environment of 25°C and 50% RH, the desiccant cartridge provides approximately 25 hours of continuous operation before requiring regeneration or replacement. Extended operation of the system can be accommodated via an external supply of conditioned dry air to the desiccant cartridge inlet.

To neutralize differential temperature effects on the mixed gas stream relative humidity, the dry air stream flows through a heat exchange coil prior to introduction into the mixing chamber. This heat exchange coil resides within the water jacket adjacent to the test chamber’s external walls and saturator and is sized to equilibrate the dry air gas stream temperature to the temperature of the test chamber and saturator bath. At the ambient operating range of the system (20…30°C), the effects of temperature variation on the dry air stream’s inherent relative humidity are negligible. However, the effects of mixing the dry air gas stream with a saturated gas stream at a different temperature can have significant effects on the resultant relative humidity and are therefore addressed via this design feature.

### *4.2 Wet Air Supply*

The wet gas stream is created by sparging an air stream through the water jacket bath, thereby creating a saturated “100% RH” air stream for introduction into the mixing chamber.

The system’s wet gas sparger consists of two sintered stainless steel air stones positioned within the water jacket. In order to obtain maximum efficiency (≅100% RH) from these elements, they are sized based on the gas exit velocity from the porous sparger surface. As a general rule, lower exit velocities at the sparger surface will create smaller air bubbles, thereby increasing the ratio of bubble surface area to bubble volume and the resultant saturation efficiency of the gas stream.

Considering the system design flow rate, water jacket depth, operating temperatures, and physical space limitations, the required sparger dimensions and porosity were determined to optimize the gas exit velocity.

After flowing through the saturator sparging elements, the diffused air bubbles travel up though the water jacket until they reach the sealed headspace at the top of the tank, where they form the wet air stream prior to introduction into the system mixing chamber. Since the saturator sparging elements reside within the water jacket, the conditioned wet gas is at the desired temperature, and there is no need for additional temperature conditioning apparatus.

### *4.3 Dry and Wet Air Stream Mixing*

Prior to introduction into the test chamber, the dry and wet gas streams, having previously been conditioned for moisture content and temperature, are blended within the system’s mixing chamber.

The mixing chamber consists of a stainless steel cavity where the wet and dry gas streams are physically “intermixed” in order to minimize water vapor pressure gradients within the test chamber environment. The exit gas stream from the mix chamber contains air at the desired relative humidity by virtue of the time-proportioned flows through the wet and dry air circuits at a single equilibrium temperature.

*4.4 Temperature Stability*

As discussed, the saturation vapor pressure is a function of the dry bulb temperature, which affects the resultant relative humidity within the test chamber. Therefore, several design elements are incorporated into the system to maximize temperature stability.

The primary temperature stability feature of the system is the water jacket that surrounds the test chamber on five sides. Water has a relatively high specific heat and conducts heat energy fairly well. When in intimate contact with the stainless steel test chamber walls, the dry air stream heat exchange coil, and the saturator sparging elements, an isothermal environment is created for the dry and wet gas streams and test chamber environment. To further enhance the temperature stability of the water jacket and test chamber, closed-cell foam insulation surrounds the water jacket tank.

As an enhancement to the system, an integrated, closed-loop heat exchange coil is incorporated within the water jacket. An external circulating water bath can be connected to this coil to provide additional temperature stability as well as control.

A further thermal consideration for the system relates to the integration of the dew point sensor. The chilled mirror sensor installed in the test chamber is a source of heat since the chilled mirror dissipates power to cool the mirror through the use of thermoelectric cooling. When operated at low % RH levels, heat dissipated by the chilled mirror can affect thermal stability of the chamber.

To minimize these affects, the chilled mirror sensor is installed within an aluminum alloy heat sink selected for its relatively high thermal conductivity. This aluminum block is in intimate contact with the water contained within the water jacket, thereby conducting the heat to the large thermal mass of the water bath.

The thermal effects of the chilled mirror self-heating are modeled with Femlab multiphysics modeling software. A steady-state temperature slice plot of this study is shown in Figure 2.

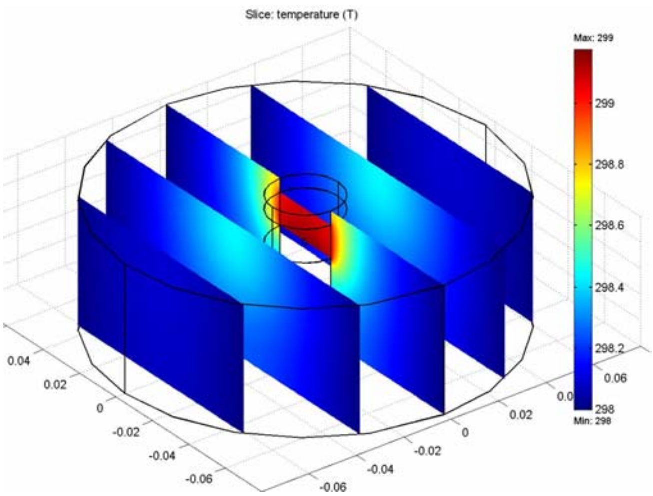


Figure 2- Thermal Modeling Output Plot

The thermal model predicts a heat sink surface temperature increase of 0.4°C maximum when the system is operated in steady-state operation at low % RH levels.

*4.5 System Response Time*

The system's dynamic humidity response is governed by the exchange rates of the air within the test chamber environment. In general terms, the water content (mass) within the test chamber at any point in time can be modeled as a first order differential equation:

$$\frac{dy}{dt} = (input\_rate) - (output\_rate) \tag{6}$$

For the system at the design flow rate, and fixed chamber volume, equation (6) becomes:

$$y = .3086 - .305518e^{-.22388t} \tag{7}$$

*Where:*
*y = mass of water, grams*
*t=time, minutes*

Per equation (7) the theoretical first time constant (63% delta) is 4 min 30 s for a step change from 0% RH to 100% RH.

It is important to note that the dynamic behavior described above is derived from *ideal conditions* and does not consider the hygroscopic nature of the chamber materials or items placed within the chamber. In addition, equation (6) and (7) are based upon air exchanges at full dry or saturated conditions. In practice, the system exchanges air at a specific time-proportioned cycle between the wet air and dry air streams, and not at full wet or dry settings. As such, actual response times can be 10 to 20% longer.

### *4.6 System Operation and Closed Loop Control*

In open-loop operation, the system responds to a user-defined %RH set point by adjusting the wet and dry gas stream duty cycle of the 3-way solenoid valve to a pre-determined level. In this mode, the system will reach an equilibrium % RH level without the interaction of feedback from the integrated dew point and temperature sensors.

Theoretically, the 3-way solenoid valve duty cycle settings should match the %RH settings per equation (2). For example, a 70%RH user set point would command a 70:30 wet/dry stream duty cycle.

Normal manufacturing variations in pump flow rate, saturator efficiency, and system piping backpressures can affect individual system performance at specific 3-way solenoid valve duty-cycle settings. In order to minimize these effects, each system is individually "calibrated" in open-loop operation.

In this calibration process, the system steps through a pre-programmed % RH timed profile, acquiring the actual % RH reading from the dew point and temperature sensors. These actual % RH readings are then incorporated into the system controller, which adjusts the duty cycle to the 3-way solenoid valve accordingly. For example, a post calibration user set point of 70%RH could command a 73:27 wet/dry stream duty cycle to the 3-way solenoid valve in order to obtain the desired 70%RH equilibrium.

In closed-loop operation, the system responds to a user-defined %RH set point by adjusting the duty cycle of the 3-way solenoid valve to the post calibration settings contained within the controller electronics. After a set period of time, the system will compare the actual %RH readings from the chilled mirror dew point and dry bulb temperature sensors, and makes a proportional adjustment to the duty cycle of the 3-way solenoid valve. The system continues to make proportional corrections to the 3-way solenoid valve duty cycle at specific timed intervals in order to maintain the user % RH set point.

Closed-loop control via the integrated dew point and dry bulb temperature sensors provides real-time correction to the chamber environment. Utilizing closed-loop control minimizes errors associated with temperature variations, chamber load, and other potential test chamber disruptions, as well as variations in dry and wet air stream conditions.

## 5. Performance Data

### *5.1 System Accuracy*

The % RH accuracy of the system is determined by the accuracy of the dew point temperature measurement, the dry bulb temperature measurement, and the dry bulb temperature gradient within the test chamber. Given the NIST traceable chilled mirror accuracy of +/- 0.15°C dew point, the precision temperature sensor accuracy of +/- 0.15°C, and an estimated temperature gradient of +/- 0.15°C the system uncertainties are presented in Figure 3.

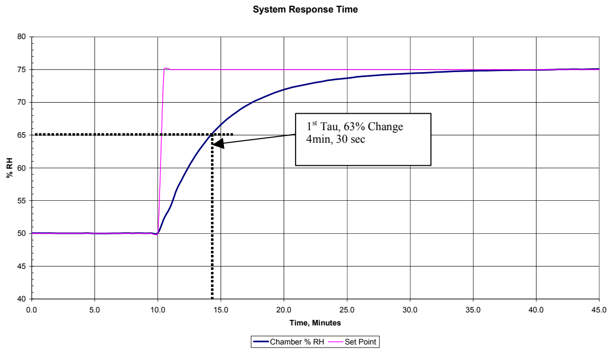
| **% RH** | **Dry Bulb Nominal Temp, °C** | **Dew Point Temp Error, % RH** | **Dry Bulb Temp Error, % RH** | **Temperature Gradient, % RH** | **RMS Uncertainty** |
|---|---|---|---|---|---|
| 10 | 20 | .14 | .09 | .09 | .19 |
| 10 | 25 | .13 | .09 | .09 | .18 |
| 10 | 30 | .13 | .09 | .09 | .18 |
| 50 | 20 | .51 | .47 | .47 | .83 |
| 50 | 25 | .49 | .45 | .45 | .80 |
| 50 | 30 | .47 | .43 | .43 | .77 |
| 90 | 20 | .85 | .84 | .85 | 1.47 |
| 90 | 25 | .82 | .81 | .82 | 1.41 |
| 90 | 30 | .79 | .78 | .79 | 1.36 |

Figure 3- Uncertainty Calculation Data

To analyze the cumulative effects of these error sources, each of the errors is converted to a %RH effect. Note that the relationship of dew point temperature, dry bulb temperature and dry bulb temperature gradient are not linear with respect to %RH. The probability that all of the errors will manifest in the same direction is small, therefore the square root of the sum of the squares (or root mean square) is reported as the RMS uncertainty or cumulative error. This uncertainty is less than 1.5% RH for the system across the operating range of 10 to 90%RH at 20 to 30°C chamber temperature.

### *5.2 System Response Time*

In operation, the system will be commanded to a % RH set point from a prior equilibrium % RH state. The graph in Figure 4 profiles the system's dynamic closed-loop control response to a "ramp-and-soak" % RH profile programmed into the system controller. This profile consists of six % RH set points with 90-minute dwell periods.



1st Tau, 63% Change
4min, 30 sec

Figure 5 – Response Time

As illustrated, stable system equilibrium conditions are obtained within 40 minutes for each step.

The first time constant for the system is 4 minutes 30 seconds for the 50% RH to 75% RH adsorption step depicted in Figure 5 and as predicted in the analysis described in section 4.5.

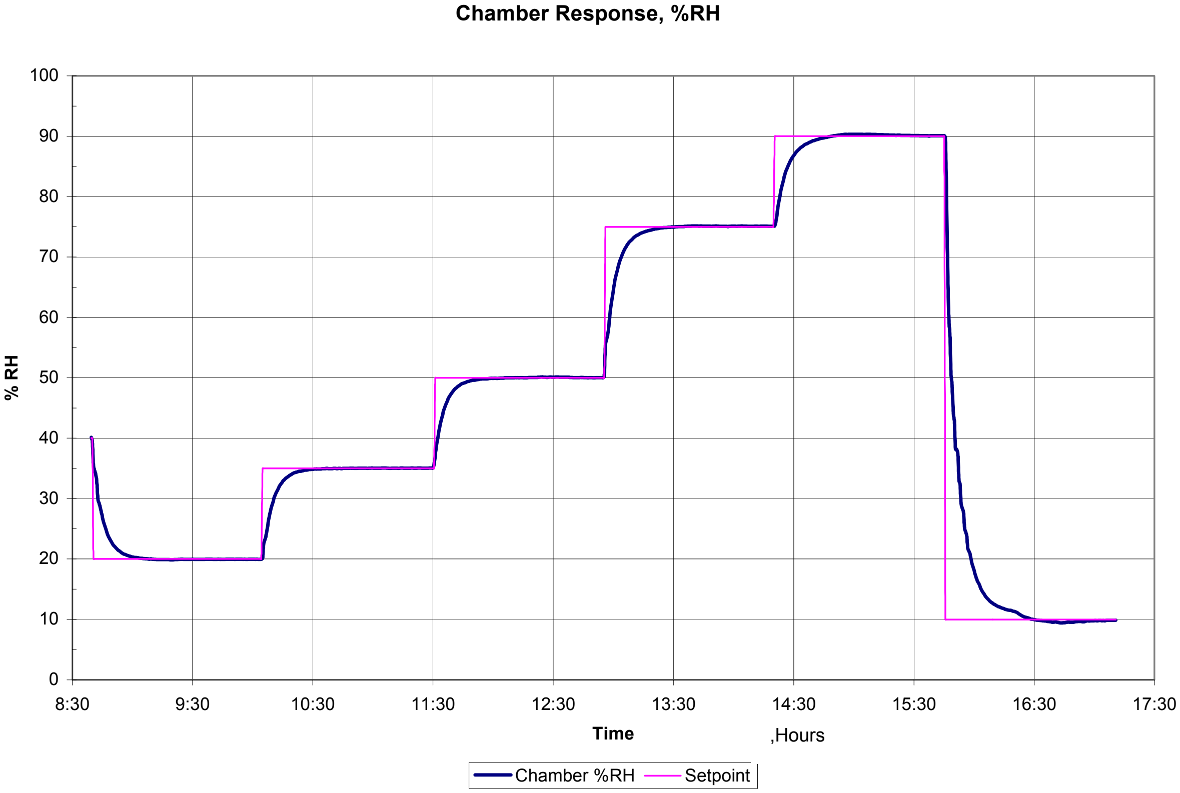


Figure 4 – System Response

### *5.3 System Stability*

After reaching an equilibrium % RH condition, the system controller continues to make incremental adjustments to the 3-way solenoid valve duty cycle in relation to the chilled mirror dew point and dry bulb temperature sensors. The response of the system to these corrections is shown in Figures 6 through 8, which illustrate the % RH stability of the system at several % RH equilibration points. The % RH stability of the system is +/- 0.2% RH through the 10 to 90 %RH range.

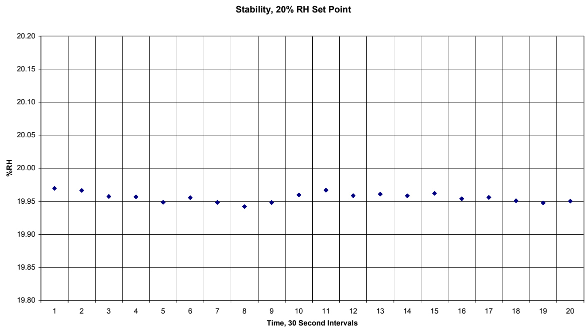


Figure 6- Stability, 20% RH

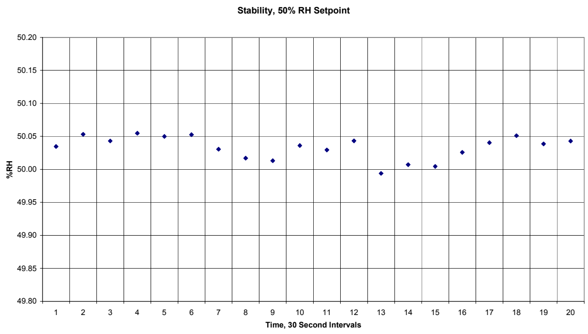


Figure 7- Stability, 50% RH

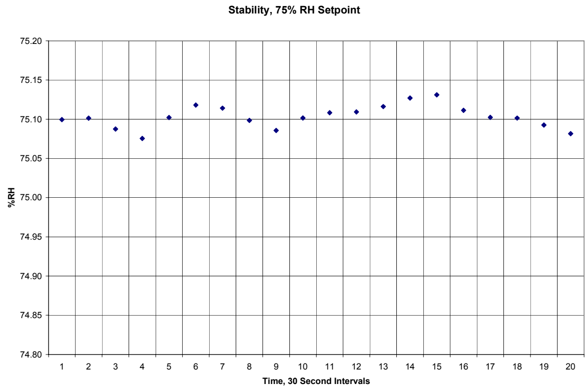


Figure 8- Stability, 75% RH

## 6. Conclusions

The divided-flow humidity generation technology is a proven method for creating precision humidity environments. When integrated with a NIST traceable chilled mirror reference hygrometer and closed-loop system controller, the technology enables the creation of accurate, repeatable, and reliable % RH calibration chambers.

The development of the bench-top humidity generator utilizing this technology is the latest iteration of these concepts, incorporating a variety of enhancements over previous designs. The system provides % RH accuracy of +/-1.5%, %RH stability of +/- 0.2% and system response times of less than 40 minutes to full equilibrium.

## 7. References

A.L. Buck, "New Equations for Computing Vapor Pressure and Enhancement Factor", *J. Applied Meteorology*, Vol. 12, Issue 20, 1527-1532, 1981

Peter R Wiederhold, *"Water Vapor Measurement"*, Marcel Decker, Inc. Boston MA, 1997

Mark E. Brownawell, United States Patent 5,056,547 "Relative Humidity Generation Techniques", Oct. 15, 1991

*GE General Eastern's Bench-top Humidity Calibration Chamber*