Proceedings of the TFAWS 2013 Conference on Thermal and Fluid Analysis
TFAWS2013
July 29-August 2, 2013, Central Florida, USA

NASA LSP VAPOR MIGRATION TEST EQUIPMENT DESIGN

William E. Boggs **Anthony P. Cook**

Robert M. Gore **Kyle D. Harris** **Michael I. Jasiukowicz**

Dr. Marcel Ilie
Department of Mechanical and Aerospace Engineering
University of Central Florida
Orlando, Florida 32816, U.S.A

ABSTRACT

Launch Service's concern of the effects of launch conditions on the evolved expendable launch vehicle (EELV) payloads in the event of environmental control system (ECS) interruption have led to the implementation of analytical models predicting the transient vapor migration from an ambient environment into the payload volume of the launch vehicle. This model has been developed for predicting potential contamination due to condensation. Due to payload sensitivity, understanding the thermal and vapor migration into the payload fairing is crucial for making accurate real time launch/no launch decisions. Experimental test equipment capable of simulating and measuring these conditions is currently being designed and built in an effort to verify the analytical model.

The purpose of this design is to replicate actual scenarios of temperature and relative humidity differences between the ambient environment and the payload section. The design process and component selection was driven by cost, manufacturability, and design constraints which were met using calculations and estimations from various thermo-fluid laws and concepts. The proposed design is to create an inner control volume to replicate scaled ratios of different launch vehicle volume configurations. This test fixture is to be located within the larger external control volume simulating launch site atmospheric conditions. Each volume has independent environmental control systems capable of achieving steady state temperature and humidity conditions over a specified range. The inner control volume will be equipped with a controllable leak area and sensor array in order to induce and track the thermal and vapor migration between the inner control volume and the simulated environmental conditions during testing. A typical test procedure would include achieving steady state conditions in both internal and external control volumes, disabling the environmental control in the inner control volume, opening the vents of the test fixture to the surrounding environment and tracking the vapor and thermal migration with respect to the inner control volume over time with a three dimensional sensor matrix. The data from the matrix of sensors with a calculated uncertainty will be collected in order to validate the original analytical model.

The use of standard parts, simple to manufacture custom parts, and accurate cost effective sensors is essential in achieving the design goals within the proposed budget. The test chamber is being designed to simulate a wide range of environmental conditions with an emphasis on maintaining the feasibility of the design, maximizing cost effectiveness while achieving experimental accuracy.

INTRODUCTION

An untested humidity migration model is currently the basis of predicting humidity migration from the operating environment to the payload fairing (PLF) in the case of environmental control system (ECS) interruption. Due to the sensitivity of evolved expendable launch vehicle (EELV) payloads to moisture contamination, it is of high importance to understand and mitigate the risks these ECS interruptions may cause near launch time. To understand the physical phenomenon of vapor migration concerning the large payload fairings a system of tests has been outlined to verify the response of different environmental conditions on a simulated PLF air volume. To accomplish these tests an environmental chamber is in development to serve as test equipment to simulate environmental launch conditions and measure the transient thermal and vapor migration with respect to a simulated PLF air volume.

For relevant data collection, a system of two independent control volumes has been designed. One control volume to replicate a range of PLF geometries, which is denoted as the inner control volume (ICV) and the other to deliver the desired test environments, designated as the external control volume (ECV). The geometric design of both control volumes is important based on the function of each. The ECV's purpose in design is to replicate environmental conditions, provide a simulated air continuum, and deliver the air quality desired within a reasonable amount of time. The purpose of the ICV is to replicate; with predetermined geometric ratios shown in Figure 1, the air volume and environmental conditions typical in the PLF during a prelaunch timetable.

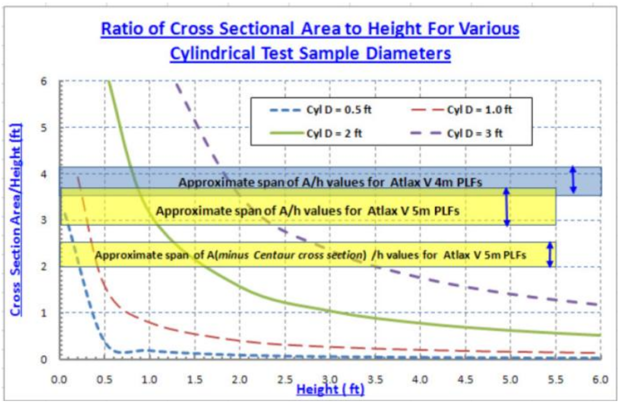

Figure 1. Ratio of cross sectional area to height for various cylindrical test equiptment diameters.

The ICV is designed to simulate the air volumes of multiple EELV PLF arrangements across several launch platforms and equipment. This calls for a cylindrical body that is capable of adjustable internal air volumes. Based on provided payload volume estimates and predetermined PLF air volumes, several test equipment diameters were considered to provide a feasible solution. The variability of the air volume inside the ICV is accomplished through the application of an adjustable piston fitted to the cylindrical volume. This test fixture is paired with an ECS capable of delivering the desired air quality within the ranges shown in Table 1.

Table 1. Temperature and Relative Humidity Ranges

ECV Ranges	ICV Ranges
$T_{ECV} \rightarrow +35°F - +95°F$	$T_{ICV} \rightarrow +40°F - +85°F$
$\emptyset_{ECV} \rightarrow 40 - 95\%$	$\emptyset_{ICV} \rightarrow 30 - 90\%$

During testing, the ECS to the ICV is disabled to simulate real world ECS outage conditions. The ICV functions as a "dead load" in that it does not generate thermal energy, simplifying the measurement of steady state conditions in the ECV during the test. The ECV ECS will continue to function to maintain steady state conditions simulating an expansive real-world environment.

Much like the design of the ICV, the ECV is required to meet geometric constraints and deliver a specified air quality with a known accuracy. To be large enough to provide a continuum of stable air quality in which to fit the ICV, the ECV's air volume has been calculated and determined. Based on the construction materials and the energy loading of the ECV, an array of equipment has been selected to deliver the temperatures and relative humidity levels desired as indicated in Table 1.

The assumption that air and water vapor both behave as an ideal gas allows for application of the ideal gas law, Equation (1), in this design solution.

$$P_w * V = n * R * T \qquad (1)$$

Using psychrometric tables, energy calculations, refrigeration loading standards and material selection the design of both the ICV and the ECV are presented and validated for functionality and accuracy. The design of each component of both systems is presented with the intent of providing a solid foundation upon which to accumulate the desired empirical data.

SYSTEM ARCHITECTURE

The design is divided into two systems separately controlled by ECSs. The combined elements of the ECV; shown in Figure 2, simulate the environmental conditions affecting the EELV.

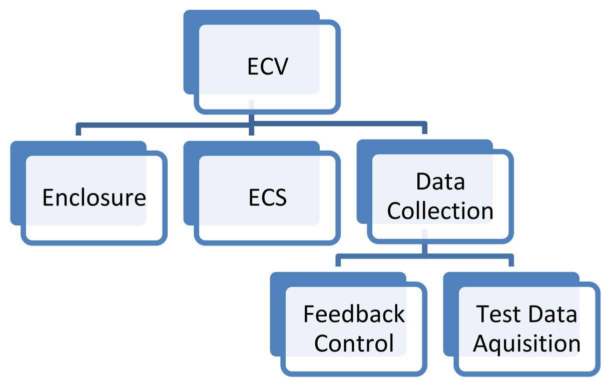

Figure 2. ECV hierarchal design table.

The ICV and its subsystems; shown in Figure 3, simulate the payload air volume of an EELV PLF.

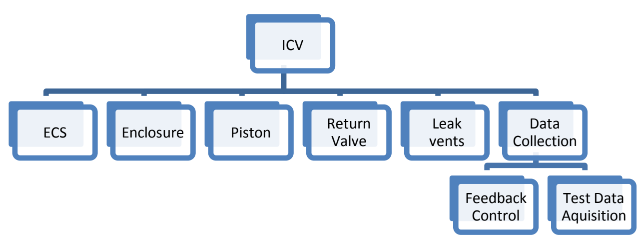

Figure 3. ICV hierarchal design table.

The ECS of the ECV and ICV are designed and function similarly such that the ICV ECS components are selected as scaled down versions of those used in the ECV ECS. The presented ECS differs from other test chamber refrigeration systems encountered in a review of test chamber patents in two notable ways. First, using direct humidity addition to keep constant the level of relative humidity during testing and second, by removing large amounts of humidity during initial cool down as this enables a more rapid change in the temperature of the system.

Separate three dimensional sensor matrices in the ECV and ICV are used for data collection and control of environmental conditions throughout each respective control volume. These sensors function both while pre-test steady state conditions are being achieved and during the transient thermal/vapor migration data collection portion of the test. The sensors utilized in the project will measure temperature, relative humidity, and absolute pressure. The General Electric T9600 sensor is a convenient, ready to install unit that measures both temperature and relative humidity. The sensor is supplied fully calibrated and temperature compensated, and provides a preconditioned output to the data acquisition (DAQ) system. The stated temperature accuracy at 77°F (25°C) is ±1.08°F (±0.6°C) and ±2% for relative humidity when measuring between 20-80%. When measuring relative humidity over 80%, the accuracy is ±3.5%. Pressure sensors are used to monitor ambient pressure and establish pressure gradients. All sensors selected operate in the same voltage range as our DAQ system natively uses.

Each control volume has a computer controlled ECS system allowing for the independent achievement of any permutation of steady state relative humidity and temperature values over the aforementioned ranges. A single control system controls both ECS's and the proposed development path for this utilizes LabVIEW for both data acquisition from our sensor array and provides feedback for control of the ECS components. The data acquisition device selected will be the National Instruments modular CompactDAQ system.

The air volume of each component is used to calculate the energy load of the system enabling determination of the Btu/hr rating of the refrigeration and heating units required to temper the air to the desired ranges. Utilizing the most energy saturated condition of air desired as the heaviest load on the system, the amount of energy removal needed was determined through the use of psychrometric charts, thermal property tables and validated by air volume decomposition, treating the air and water vapor as an ideal gas mixture.

The amount of heat needed to be removed from dry air to obtain this change in temperature can be calculated using Equation (2).

$$\Delta Q = c_p * m * \Delta T \qquad (2)$$

Using the specific heat , mass of water vapor and the maximum change in temperature the amount of heat to be removed from the moisture in the air is determined. These calculations can be verified by Psychrometric charts. These charts enable simple verification of the calculated energy change under the assumption that the overall mixture and each mixture component behave as ideal gasses at the states under consideration. [1]

The following sections breakdown the design and design parameters specific to the ECV, ICV and their subsystems. The values for the calculated thermal loads and the selected components are presented to form a clear picture of these systems.

THE EXTERNAL CONTROL VOLUME

The design of the ECV is based on the design of a modern thermal chamber. Several system requirements provided for the basis of the design, Table 2.

Table 2. ECV Customer Requirements

Temperature Control
Relative Humidity Control
Data Acquisition System
Timely Steady State Conditions

The system is designed to be large enough to encapsulate the desired test article, and is to be capable of reaching steady state within a reasonable amount of time and of sensing the environmental changes both spatially and over time. The ECV includes: an insulated structure capable of simulating of atmospheric conditions, a data collection system comprising relative humidity/temperature sensors and pressure sensors, an ECS comprising a refrigeration system having a compressor, a condenser, and an evaporator, expansion valve, a system for directly adding atomized moisture, a direct heat addition system and a dehumidification system. The organization of these subsystems, based on design requirements, is illustrated in Figure 2.

The ECV components were selected based on analytical prediction of thermal loads and requirements of the test ranges presented in Table 1. A commercial cooler was selected to provide the required insulative properties. The design of the ECV not only considers the insulative qualities but also the experimental envelope required to fit the test fixture. These design considerations have led to calculations centered on accepting that the air surrounding the test fixture can be classified as an infinite continuum with respect to the environmental envelope. According to the design constraints for the ICV as shown in Figure 1 the maximum air volume for the design based on a 2 ft (0.610 m) cross section is 6.28 ft^3 (0.178 m^3). To design a system that can be considered infinite with respect to the ICV's air volume, the assumption must be made such that an air volume of the ICV $\leq 1\%$ of the air volume of the ECV. This provides a reasonable justification for assuming infinite volume with respect to the ICV. This would require, for maximum simulated PLF air volume, an ECV air volume of 777 ft^3 (22.0 m^3) or greater. This has been satisfied through selection of a 12 ft x12 ft x8 ft cooler box with reinforced floor which provides for approximately 1000 ft^3 of tempered air.

To achieve acceptable results, the design required construction similar to industrial environmental chambers. A review of designs concluded these use insulated panels wrapped in thin sheet metal similar to that of commercial refrigeration insulative panel construction. Expanded polystyrene skinned with steel or aluminum sheeting is commonly used in construction of these panels. Typical panel construction provides for interlocking panels and very low thermal conductivity; k ≈ 0.26 Btu/hr*ft*ºF (0.045 W/m* ºC). [2] This enables the isothermal properties desired by the design parameters to be achieved. The interlocking panels of the control volume minimize heat loss at panel joints. After assembly, the joints of the ECV are sealed with environmental silicone to provide confidence in the joint to seal out undesired contaminants. The ECV walls require small modifications to allow for installation of the ECS components and sensory systems. These areas are filled with foam insulation, sealing them prior to testing.

All air is to be cycled through the system and delivered to the ECV through a 24 in. (0.61 m) plenum. The plenum is designed to diffuse and disperse the air allowing for random air flow throughout the ECV. The implementation of the plenum keeps directionally oriented flow from interfering with the testing results. The vent surface area is maximized to reduced localized flow velocities. Figure 4 shows a basic rendering of the ECV and ICV as currently designed.

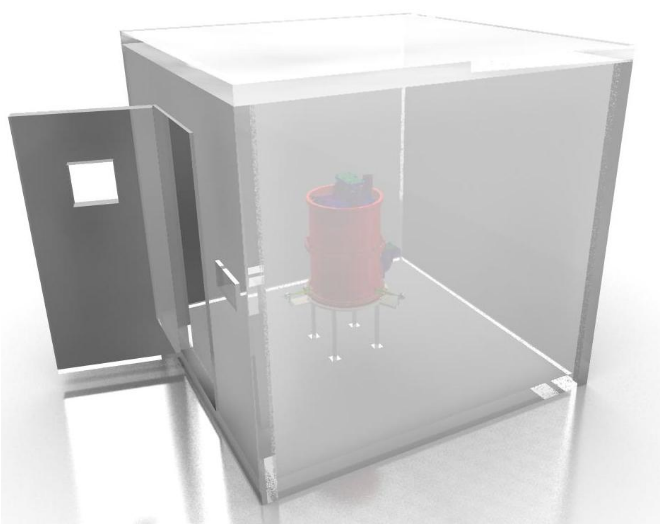

Figure 4. Basic architecture of test chamber.

The design selection of the ECV's ECS components required for the analysis of the system and its performance. There are several factors that go into determining unit sizing for refrigeration units. The design has taken into consideration the moisture content of the air and its effects on the design as well as evaluated the refrigeration load to determine the component level system requirements.

For the ECV, the minimum temperature needed will be 35°F (2°C), and a 25°F (13.9°C) temperature difference between the evaporator's saturated suction temperature and the air volume would provide a relative humidity of around 50-65%. However, this would frequently create icing on the evaporator coils which would inhibit moisture and heat removal rates. Commercial evaporators employ methods that include an air or electric defrost to remove ice buildup on evaporator coils. An electric defrost will add some heat generation to the system, and can remove ice faster than the more common and less expensive air defrost. A 10°F (5.56°C) temperature difference for the evaporator is a commonly available specification, ensuring cost effectiveness. When working in tandem with a dehumidifier capable of bringing the humidity down to 40% at low range temperatures, defrost time can be minimized or eliminated.

When calculating the maximum refrigeration load, an extreme test setup is considered, such as using the equipment in a ventilated warehouse in Florida where mid-summer conditions can get up to 95°F (35°C) and 95% RH. From the test specifications listed in Table 1, the lowest temperature the ECV will reach is 35°F (2°C) giving a ΔT of 60°F (33.3°C).

To select an appropriate refrigeration unit, the refrigeration load was calculated. The refrigeration load is defined as the rate at which energy must be removed from a space to maintain a constant space air temperature [3]. To size the condensing unit and evaporator for the ECV, the total refrigeration load must be calculated. Using methods described in the 2006 ASHRAE Refrigeration Handbook, the total load consists of five major components. The total load consists of a transmission load, product load, internal load, air infiltration load, and an equipment related load [3]. These components will be compiled to ascertain the total refrigeration load in order to size an adequate system.

The transmission load is the sensible heat gain or loss that is determined for each control volume. The convection heat transfer coefficient for air is conservatively assumed to be 4.40 Btu/hr*ft^2*°F (25 W/m^2*ºC) in all calculations.

During a maximum energy difference scenario, the steady-state temperature of the ECV will be 35°F (2°C), and the ICV will be 85°F (29°C). Considering the heat transfer rate between the control volumes, the heat loss from the ICV adds to the cooling load of the ECV. Equation (3) is used to calculate the magnitude of the heat transfer rate between the control volumes in the most extreme case specified. The heat transfer rate for such a case is found to be 159 Btu/hr (46.5 W).

$$\dot{Q}_{ICV} = \frac{\Delta T}{\frac{\ln\left(r_2/r_1\right)}{2\pi k_{SLA} H} + \frac{1}{2\pi r_1 h_f H} + \frac{\ln\left(r_3/r_2\right)}{2\pi k_{TB} H} + \frac{1}{2\pi r_3 h_f H}} \quad \textbf{(3)}$$

The flexible ducting utilized by the ECS associated with the internal control volume also applies a rate of heat transfer with the ECV in the extreme scenario. It is assumed that the ducting acts as a uniform straight cylinder in the calculation. This ducting is a 3 in. (7.62 cm) diameter tube with approximately 1 in. (2.54 cm) of fiberglass batting insulation, and the length of ducting is approximately 30 ft (9.14 m). The heat transfer rate related to the ducting is determined to be 238 Btu/hr (69.6 W).

For the ECV, the walls, ceiling, and floor are all considered to have the same composite cross-section for simplicity, and it consists of 4 in. (0.102 m) of extruded polystyrene between two 0.018 in. (0.457 mm) sheets of galvanized steel. The total area for the simplified composite wall is the average for the interior and exterior surface area, and this value is 611 ft^2 (56.8 m^2). Using Equation (4) with a ΔT of 60°F (33°C) to represent the extreme condition scenario with an ambient temperature of 95°F (35°C) outside the ECV, the heat transfer rate is found to be 1671 Btu/hr (490 W). The total transmission load for the ECV is the summation of the heat transfer rate with the ambient environment and that with the ICV including ducting associated with its ECS. The magnitude of the transmission load is 2068 Btu/hr (606 W).

$$\dot{Q} = UA\Delta T \tag{4}$$

A design parameter applying to the refrigeration system is the ability to reach low temperature conditions of 35°F (2°C) with a RH of 40% with initial conditions of 95°F (35°C) with 95% RH both inside and outside the ECV in a reasonable amount of time. The assumption is made that the dehumidifier is able to remove moisture fast enough to not allow excessive icing or a condensing environment on the inside surfaces of the ECV. The individual contributions of required energy to achieve the ΔT consisting of the interior steel panels, the insulation, dry air, and the vapor are calculated using Equation (2). The robust dehumidification system is considered to be ideally able to remove vapor from the ICV and ECV environment fast enough that the latent heat of condensation is never part of the mechanical refrigeration load. The summation of energy required to drop each substance is then divided by the desired time to bring this total mass to 35°F (2°C), in this case one hour. The product load is found to be 6496 Btu/hr (1902 W).

The internal load was then determined by considering any heat generated by equipment operating inside the refrigerated volume. The calculations assume that the low voltage sensor array contributes a negligible heat transfer rate to the system. The ECV does not have light generating heat during test, and any other mechanical systems would have an insignificant impact to the entire system.

The infiltration load accounts for the rate of heat transfer associated with the ECV door being opened. This may be necessary to make adjustment, repairs or observations. Though this is expected to be infrequent, it is considered to ensure proper sizing according to ASHRAE standards. The methodology for calculating the infiltration load first involves finding the sensible and latent heat exchange for fully established flow in Equation 5 [4]. The infiltration air is considered to be 95°F (35°C) with 95% RH. The refrigerated air is considered to be 35°F (2°C) with a 40% RH to continue the maximum heat transfer rate condition. The densities are found for the mixture using the Ideal Gas Law, Equation (1), and the enthalpy values are found using psychrometric charts. The calculations assume that there are 4 doorway transits lasting 5 seconds each in 24 hours. The door is never left open, and a doorway flow factor is considered to be 0.8 for this ΔT, and the effectiveness is considered to be 0.95 for a conservative estimate of the standard door as seen in the ASHRAE Refrigeration Handbook. After the calculations are made, the total air infiltration load is determined from Equation (6) to be 8601 Btu/hr (2521 W).

$$Q_f = 795.6A(h_i - h_r)\rho_r \sqrt{1 - \frac{\rho_i}{\rho_r}} \sqrt{gH} F_m \tag{5}$$

$$Q = Q_f D_t D_f (1 - E) \tag{6}$$

For the equipment load, heat gain would result from fan motors on the evaporator and from heat associated with the reactivation process in the dehumidification system. There are four 1/20 hp (37.3 W) fan motors attached to the evaporator in the ECV. These motors contribute to the internal load with an approximate total heat transfer rate of 960 Btu/hr (281 W) using ASHRAE data tables [3]. The rotary desiccant dehumidifier contributes approximately ten percent of the reactivation heat necessary to facilitate desorption. For the ECV, this is approximately 1945 Btu/hr (570 W). The total equipment load is determined to be 2905 Btu/hr (851 W).

The refrigeration load is the summation of the transmission, product, internal, and air infiltration load in this application. ASHRAE states that the load is increased by a factor of 10% as a factor of safety [3] to account for any possible variation. Using this methodology, a total refrigeration load for the ECV is found to be approximately 22,070 Btu/hr (6468 W).

Based on these load calculations the equipment selected to be capable of managing these thermal load requirements was determined to consist of a Copeland FFAP-030Z condensing unit and Larkin Compak LCA6215AB unit cooler utilizing R404A refrigerant.

To provide the humidity required to reach the operational ranges outlined in TABLE 1 a unit is used to increase the humidity level through the dispersion of atomized water particles. The selected unit is a centrifugal humidifier which uses a spinning disk to atomize the water. The centrifugal humidifier blows very fine droplets 6 to 8 microns in diameter into the controlled environment using a built-in fan. The droplets are then absorbed into the air and blown into the environment. The selected unit has a high atomization capacity of 14.3 lb /hr and low power consumption of 75W per kg/hr. The mass associated with the atomizer's humidification process is assumed to be relatively small, and therefore the latent heat of vaporization is not factored into load calculations.

The dehumidification unit is affixed to the return air box of the ECV's environmental control cabinet allowing the air to be dehumidified prior to being heated or cooled. The dehumidification of the ECV utilizes an adsorption process with silica gel in a rotary desiccant wheel. The desorption process is enabled through heat addition to the silica gel, and this will consequently add a small amount of heat to the process outlet. This heat addition can reasonably be considered to offset the latent heat of condensation that would be seen in an application where only a vapor-compression refrigeration cycle is used to remove humidity within an enclosed space. A typical 300 cfm (0.142 m^3/s) rotary desiccant dehumidifier to be used in the ECV can collect 5 lb/hr (2.27 kg/hr) at 70°F (21°C) with a relative humidity at 30%. The rotary desiccant dehumidifier working in tandem with a properly sized mechanical refrigeration system allows for a much faster and more efficient process to achieve the desired temperature and relative humidity ranges.

The ECV is fitted with an array of the aforementioned RH and temperature sensors spaced evenly throughout the area to measure and establish a 3-dimensional gradient of those parameters.

A similar method of calculation for the heat addition requirements for each control volume is employed to determine the appropriate size of components. The load is calculated by considering the inverse of the previously used maximum energy difference condition where the ambient temperature outside the ECV and initial temperature inside the ECV is 35°F (2°C) with a desired temperature inside the ECV of 95°F (35°C) to be obtained in a reasonable amount of time. The heating load is calculated as the summation of the values found for the transmission, product, and infiltration loads. These values were found previously when determining the refrigeration load for the ECV. The equipment related load is omitted as a small factor of safety. The heater necessary for the ECV must have the capability to deliver 18,880 Btu/hr (5532 W). The hardware selected for this application is a zero-clearance in-line round duct heater.

Each component of the ECV's subsystem has been selected on its ability to reach and maintaining desired system required environmental conditions within a reasonable amount of time. The system as a whole will provide for stable thermal environments in a timely manner with which to carry out the desired testing.

THE INTERNAL CONTROL VOLUME

The ICV's design is driven by the design requirements and is based on the geometry of EELV payload fairings. The test requirements outline a need to adjust the air volume for different test set ups, provide backflow restriction and controllable leak vents. The ECS of the ICV is similar in types of components and application to that of the ECV with the notable exception of premixing air and moisture prior to being delivered to the ICV due to the much smaller volume.

The size selection for the diameter was driven by the functional height ranges provided in Figure 1. Given a 2 ft (0.610 m) diameter, the ICV volume ranges for multiple PLF applications falls between 0.80 ft^3 (0.244 m^3) and 1.80 ft^3 (0.549 m^3). This cylindrical volume has an aspect ratio of approximately 4:5 at its largest giving it very stable footprint. The geometry of the ICV shell, due to manufacturing constraints, is designed as two components. The two halves are joined through clamping force provided by fasteners around the perimeter. The environmental seal between the two halves is provided by a standard O-ring and gland designed to allow the proper amount of O-ring compression to provide the desired level of sealing. Figure 5 shows an oblique view of the ICV design illustrating the fastener locations and O-ring seal.

Leak valves capable of simulating a wide range of leak cross-sectional areas are installed at the bottom of the lower portion of the ICV housing. The mounting features for the valves are a flat surface with a seal allowing the valve seat to mate to the wall of the ICV. The needles are actuated through application of a non-captive linear actuator giving precise control to the vent area. This range of adjustability in cylinder height and leak valve cross sectional area allows for a wide range of PLF geometries to be simulated

Figure 5. Oblique view of 2-part cylinder with O-ring seal.

The outer shell of the control volume is designed to receive an adjustable top which allows the volume of air inside to meet the operational ranges making a movable piston design ideal for this application.

The piston style top, which is adjustable in vertical travel and held in place by pressure stops, has several unique design features. The adjustable piston head is designed to function not only as the experimental barrier but also as a plenum and backflow restrictor once the test is underway.

The outer flanges of the piston are design to receive upper and lower multilayer adhesive strip weather seals which will center the piston and provide a vapor and thermal barrier. These are replaceable seals if wear or deterioration is found to cause problems.

The back flow check valves designed into the geometry of the piston head will, under ICV ECS operation, allow air to be delivered to the isolated air volume. Upon commencement of a test the ICV's ECS will shut off and the air inlets will close allowing the environment inside the container to be sealed off. This is accomplished through the application of floating ball check valves. The force of the air flow was determined and the mass of the ball selected to allow the ball to be lifted during ICV ECS operation. When the air flow is stopped, the ball falls into place sealing off the body of air. Figure 6 shows the flow diagram of the piston ball valve design.

The flow of air being delivered to the ICV is slowed and disbursed symmetrically through the implementation of the plenum built into the design of the head. This feature also allows excess moisture build up to be collected and then removed from the piston head between testing cycles.

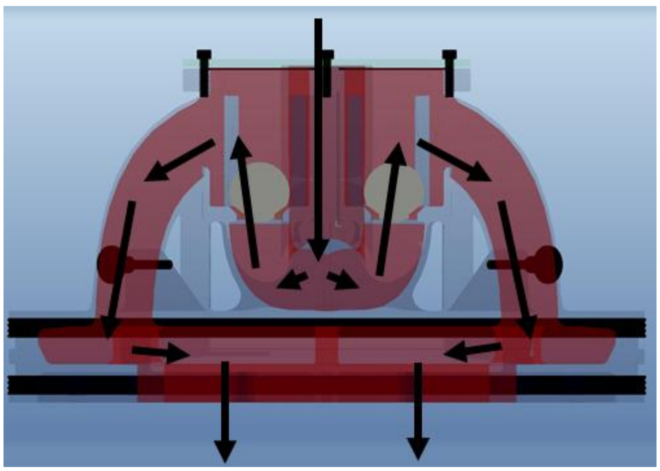

Figure 6. Air flow and force diagram for ball valve design.

Several components of the ICV are fabricated using additive manufacturing techniques. Using stereolithography the ICV is printed, layer upon layer, out of humidity resistant ProtoTherm 12120 enabling a high quality surface finish for sealable mating surfaces. The completed shell of the ICV will weigh approximately 36 Lbs. (160 N) enabling easy handling and setup. A rendering of the ICV as currently proposed is shown in Figure 7.

Figure 7. Oblique view of ICV assembly.

The sensor matrix in the ICV provides temperature and relative humidity data that can be used to construct a dynamic map of the simulated PLF conditions in the ICV. Vertically spaced, axially symmetric planes of sensors allow for a three dimensional map of the thermal and vapor migration to be constructed, ultimately allowing launch officials to precisely determine when an environment conducive to the formation of condensation exists within the PLF. This insight allows for accurate real time launch decisions to be made once an ECS failure has occurred.

Pressure sensors are mounted within the ICV and outside the leak ports on the cylinder to measure a pressure gradient during a test. The sensor selected is the Measurement Specialties US300 which offers ±0.1% best fit straight line accuracy in absolute pressure.

The ECS of the ICV is a closed loop system and is designed to deliver the air quality required for the rather small air volume of the ICV. The system is all external to the ECV and is arranged to mix and evaluate air quality prior to delivery into the test fixture.

In determining the sizing of the ICV ECS components the refrigeration load is calculated in the same manner as for the ECV, although there is no associated product, internal or infiltration load. The ICV is a composite cylindrical wall composed of Somos ProtoTherm 12120 and Thermablock insulation. The 3D printed wall has a thickness of 0.25 in. (6.35mm), and thermal conductivity of 0.127 Btu/hr*ft*°F (0.22 W/m*ºC). The ICV is wrapped in a single 0.39 in. (10mm) thick layer of Thermablok aerogel blanket will provide the highly insulative properties desired due to its thermal conductivity of 0.008 Btu/hr*ft*ºF (0.014 W/m*ºC). At the most extreme thermal energy difference proposed in Table 1, a heat loss of 128 Btu/hr (37.4 W) will occur. The flexible ducting to and from the internal control volume also has heat loss. This ducting consists of 3 inch (7.62 cm) diameter tube with approximately 1 inch (2.54 cm) of fiberglass batting insulation and is approximately 30 ft (9.14 m) in length. The heat loss through the ducting is found to be 238 Btu/hr (69.8 W). Combining the heat loss calculation the transmission load for the ICV is found to be 366 Btu/hr (107 W).

The product load consists of the material that makes up the mixing chamber, the ducting, the 3D-printed components, ThermoBlock, and the moist air mixture. Due to the small amount of mass involved, the product load is considered to be negligible, and is accounted for by the factor of safety. There is no infiltration load for the ICV, and it is neglected in the ICV total refrigeration load. The relevant equipment related load for the ICV is the ten percent of the ICV dehumidifier's reactivation heat, and a 50 cfm (0.024 m^3/s) unit typically would add 324 Btu/hr (95 W) of heat into the system. The single small fan motor on the evaporator is also reasonably accounted for with the generous factor of safety. The summation of the transmission and equipment related load, incorporating a factor of safety of 2, indicates a refrigeration load of the ICV no less than 1380 Btu/hr (405 W). The evaporator is to have a 10°F (5.56°C) temperature difference between the saturated suction temperature and the lowest environmental temperature desired.

Based on these load calculations the equipment selected to be capable of managing the thermal load requirements of the ICV was determined to consist of a Copeland M4FH-0025-IAA-272 condensing unit and Johnstones B92-108 unit cooler utilizing R404A refrigerant.

The heating requirement for the ICV is considered to be the magnitude of the transmission load in the total refrigeration load calculations. A custom in-line duct heater will be constructed to accomplish the heating necessary.

Humidity addition is accomplished by employing a centrifugal atomizer to supply the required humidification. The selected unit will supply the heavy air to an externally positioned mixing chamber. The unit has a high atomization capacity of 2.2 lb /hr and a low power consumption of 75W per kg/hr.

Each component of the ICV's system has been selected on its merits in reaching and maintaining desired system required environmental conditions. Each subsystem working to provide the air quality desired to be replicated.

CONCLUSION

The ECV, ICV, and data acquisition system as presented provide the means to empirically validate current analytical models being used to determine environmental conditions within the EELV PLF during launch site ECS outages. The specified range of air temperature and relative humidity values that the ECV and ICV have been quantitatively shown to be capable of reaching and sustaining necessitate individual ECS components as specified for cooling, heating, dehumidifying and humidifying air for each respective control volume. The development of a feedback control system capable of efficiently operating these components to achieve the desired steady state values, coupled with a precision sensory array and data acquisition system make it possible to generate and collect accurate data. Additionally, the resulting data from vapor migration testing allows for the refinement and optimization of analytical models being used, ultimately enabling launch officials to make real time launch/no launch decisions with increased accuracy during EELV PLF ECS failure scenarios. Geometric constraints provided by LSP along with results from detailed calculations have allowed for design and configuration of the ECV, ICV, ECS components, and sensory systems in a manner which is optimized for accurate and repeatable thermal and vapor migration data generation and collection.

ACKNOWLEDGMENTS

The opportunity to design and present this information was made possible by Dr. Paul Schallhorn, Chief Environments and Launch Approval Branch. The support efforts of Gary O'Neil; Launch Vehicle Thermal Analysis Team Lead, Environments and Launch Approval Branch, as our technical advisor have been a great help in our design process. Sean Hollis, Developmental Mechanical Engineer at Thermablok Thermal and Acoustic Isolation Systems of Tampa Florida has through his company donated their products for use in providing the insulation and thermal isolation required by this design. . Finally this effort would not have been possible without the support of Dr. Lionel Hewavitharana from the University of Central Florida. His dedication to providing us with the tools we need and the guidance required to accomplish our design goals has been unsurpassed during this process.

REFERENCES

[1] M. J. Moran and H. N. Shapiro, Fundamentals of Engineering Thermodynamics, Hoboken: Wiley, 2004.

[2] PermaTherm, *Expanded Polystyrene DataSheet,* 2008.

[3] American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., 2006 ASHRAE Handbook: Refrigertion, Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2006.

[4] W. Gosney and H. Olama, "Heat and Enthalpy Gains Through Cold Room Doorways," *Proceedings of the Institute of Refrigeration* , vol. 72, pp. 31-41, 1975.

CONTACT

William Boggs is a mechanical engineering undergraduate at the University of Central Florida specializing in Energy Systems and can be contacted by e-mail at web18@knights.ucf.edu

Anthony Cook is a mechanical engineering undergraduate student with a back ground in structural steel manufacturing and drafting technologies. He is currently pursuing his bachelor’s degree in Mechanical Systems and can be contacted by phone at 321-321-6998 or by e-mail at cook_anthony@knights.ucf.edu

Robert Gore is a mechanical engineering undergraduate at the University of Central Florida specializing in Materials. He can be reached by email at grtfast@knights.ucf.edu.

Kyle Harris is a mechanical engineering undergraduate researcher at the University of Central Florida specializing in CNC machining and non-oxide ceramics matrix composites manufacturing. He can be contacted by phone at 407-375-0340 or by e-mail at kyle.h@knights.ucf.edu.

Michael Jasiukowicz is a mechanical engineering undergraduate at the University of Central Florida with a concentration in the study of mechanical systems and can be contacted by e-mail at mjasiuko@knights.ucf.edu.

NOMENCLATURE

A	Area	ft^2 (m^2)
c_p	Specific heat	Btu/lbm*°F (kJ/kg*°C)
D_f	Doorway flow factor	
D_t	Doorway open time factor	
d_p	number of doorway transits	
E	Effectiveness of doorway protective device	
F_m	Density factor	
g	Gravitational constant	32.2 ft/s^2 (9.81 m/s^2)
H	Height	ft (m)
h	Enthalpy	Btu/lb_m (kJ/hr)
h_i	Enthalpy of infiltration air	Btu/lb_m (kJ/kg)
h_r	Enthalpy of refrigerated air	Btu/lb_m (kJ/kg)
P	Pressure	psia (kPa)
Q	Energy	Btu (J)
Q_f	Heat exchange for fully developed flow	Btu/hr (W)
$\dot{Q}$	Work	Btu/hr (W)/hr
T	Temperature	°F (°K)
U	Overall heat transfer coefficient	W/m^2*K (Btu/hr*ft^2*°F)
ρ	Density	lb_m/ft^3 (kg/m^3)
ρ_i	Density of infiltration air	lb_m/ft^3 (kg/m^3)
ρ_r	Density of refrigerated air	lb_m/ft^3 (kg/m^3)
ΔT	Change in temperature	