

**MASS**  
**Modular Acoustic Suppression System**

California Polytechnic State University, San Luis Obispo  
2026 Human Lander Challenge Technical Paper



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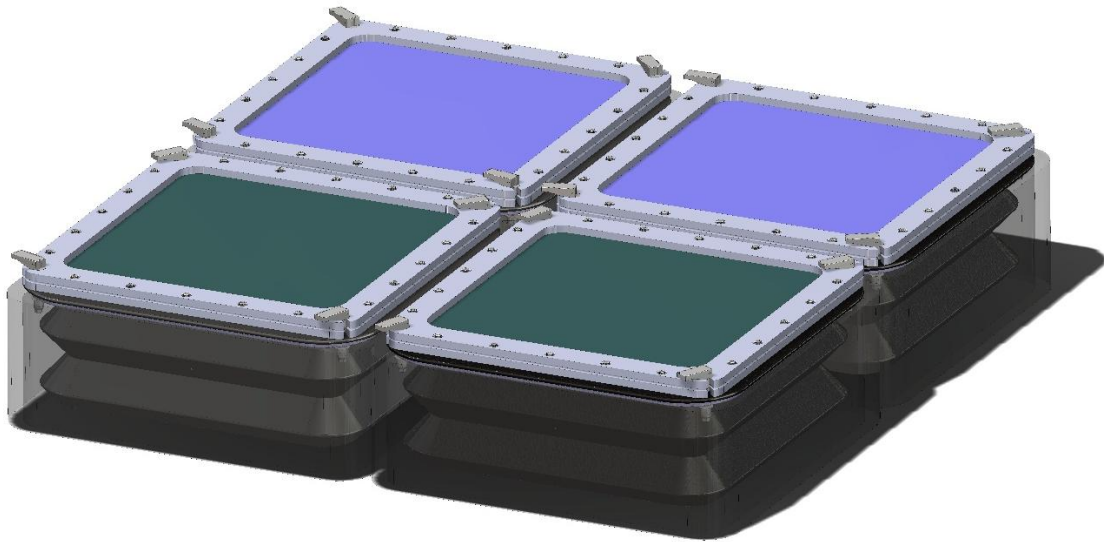
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# Quad Chart



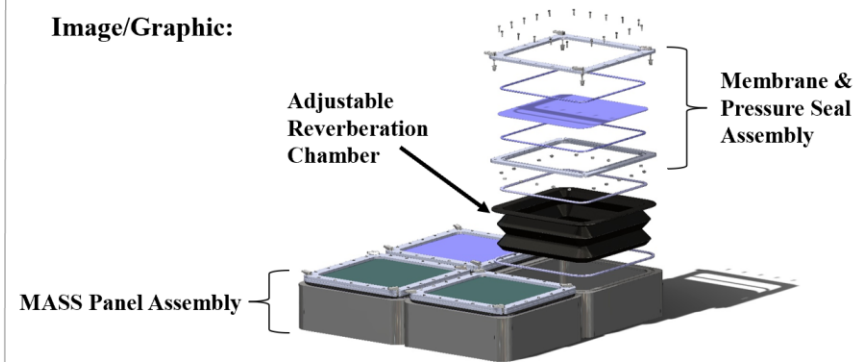
## MASS: Modular Acoustic Suppression System *California Polytechnic State University, San Luis Obispo*



### Theme Category, Major Objectives & Technical Approach:

- **Theme Category:** Noise Suppression and Control
- **Major Objective:** Develop a system that manages pervasive background noise from ECLSS equipment
- **Technical Concept:** Suppress specific targeted frequencies with a tuned membrane absorber
- **Technical Approach:** Evaluate frequency suppression through simulation, analyze material performance

### Image/Graphic:



### Key Design Details and Innovations:

#### Details:

- Tuned membrane absorber reduces frequencies within a range of 60-200 Hz
- Interchangeable flexible membranes act as springs at ambient pressure, effectively reducing sound by achieving resonance with incoming sound waves

#### Innovations:

- Adjustable depth of the chamber allows for a single system to be used in areas of varying frequency
- Acoustic feedback system allows for active adjustment based on environmental input

### Summary of Schedule & Costs:

**Year 1:** Q1: Design Requirements/Constraints, Preliminary CAD; Q2: Preliminary Analysis – Material Selection, FEA/Simulation; Q3: Design Iteration, Finalized Material Selection; Q4: Prototype Fabrication

**Year 2:** Q1: Launch Fatigue, Durability, and Thermal Testing; Q2: Design Iteration Based on Test Results; Q3: Finalize Design; Q4: Final Manufacturing Plan

**Year 3:** Q1: Final Fabrication; Q2: Part Inspection; Q3: Flight Certification; Q4: Final Implementation, MASS available for use in crewed mission

**Lifecycle Development Cost:** \$5.6 M

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## **1. Executive Summary**

MASS (Modular Acoustic Suppression System) is a mass tuned damper system that reverberates a membrane atop a cavity to absorb select frequencies produced by the Environmental Control and Life Support Systems (ECLSS) upon NASA's Artemis Human Lander. The system is adjusted via lead screws to target specific frequencies corresponding to different chamber heights.

Under the classical membrane-absorber model, incident acoustic waves coming into contact with the sealed membrane chamber create small oscillations similar to a spring-mass damper system that act as an acoustic spring to dissipate sonic energy [3]. For a constant cross-sectional area, modulating the chamber depth changes the corresponding tuned frequency according to the relationship derived in Eq. (7), where it was found that lower target frequencies require larger cavity depths. This enables active tuning of the absorber by adjusting the vertical position of the mounting plate.

## **2. Project Description**

### **2.1 Problem Statement**

As humanity progresses further in space exploration, astronauts will need to stay in space for longer durations of time [13, 15]. Currently, the ECLSS in the human lander produces sound outside the specified Noise Criterion levels [1,8] that inhibits astronauts' ability to safely live and communicate aboard the lander, with low frequencies affecting concentration and sleep [2, 8, 11]. NASA needs a way to suppress this noise on the Artemis Human Lander capsule.

### **2.2 Solution**

The California Polytechnic State University (Cal Poly) NASA HuLC team proposes the Modular Acoustic Suppression System (MASS), an adjustable tuned membrane absorber that effectively reduces sound by achieving resonance with incoming sound waves [3]. Using principles of Helmholtz resonance, the system's control scheme takes measured frequency as an input and outputs a required height value to the motor system to change the depth of the acoustic membrane chamber, adjusting to absorb a range of frequency inputs. This design is intended to passively decrease background noise with minimal maintenance, enabling astronauts to more efficiently focus and communicate [8, 9, 11].

### **2.3 Changes from Proposal**

Since the initial proposal, the Cal Poly team has determined that the design could be improved by decreasing the total volume of the design. To implement this design change, denser membrane materials were selected and the targeted frequency range was split into two overlapping ranges based on different membrane densities: 60-170 Hz with a denser surface mass density and 76-200 Hz with a less dense surface mass density. This decreased the required chamber height from 0.305 m to 0.2 m with a smaller minimum depth as well to still meet the required frequency ranges. The addition of another material density led to another resonating unit being added to a panel of devices with membrane densities. Further emphasis was placed on the orifice placed at the bottom of the device that allows for improved pressure equalization as both the environment changes in pressure and the volume shifts through actuation. In addition to these changes in physical design, the budget was updated to reflect the change in design and pinpoint more realistic values.

### **2.4 Innovation**

Current solutions to dampen noise in space and commercial applications are often purely mechanical, bulky, and do not adapt to the environment [8, 10, 14]. MASS is an adjustable system that can adapt to any sound environment within the range of 60-200 Hz throughout the working day and during nighttime [3]. It can, in real time and without user input, adapt to frequency outputs of different research equipment, ECLSS shifts, and vocal levels depending on the time of day. This ability to adjust is also applicable for long duration shifts as life support equipment wears down, which could lead to adjusted frequencies in fans and pumps

as dust and wear accumulates. In addition to this, NASA's current acoustic standards and quiet-fan work put emphasis on higher frequency, tonal fan-noise which operates in acoustic bands above MASS's 60-200 Hz target range [8, 10, 14].

## 2.5 Physical Design Description

The system consists of three main subsystems: a membrane cartridge, a reverberation chamber, and a control system. The final assembly of this acoustic damper system is shown in Figure 1.

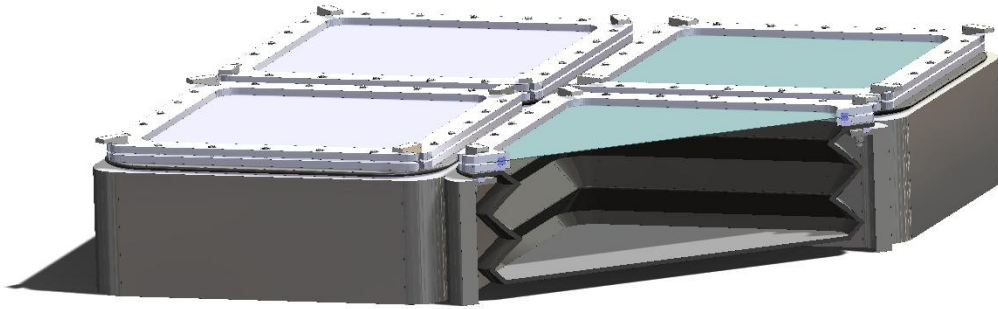


Figure 1. Acoustic Damper CAD Final Assembly.

The membrane cartridge consists of a dense membrane material, a top and bottom plate and gasket, and compression screws. The membrane will act in tandem with sealed air in the cavity to create a spring-like system that will absorb the vibration of emitted sound waves [3]. The reverberation chamber includes another gasket, which completes the pressure seal on the enclosed air cavity, and the accordion pressure cavity. The control system will facilitate vertical movement of the bottom mounting plate, which is attached to the bottom of the accordion pressure cavity, thus enabling the ability to change the distance between the membrane and the bottom of the accordion.

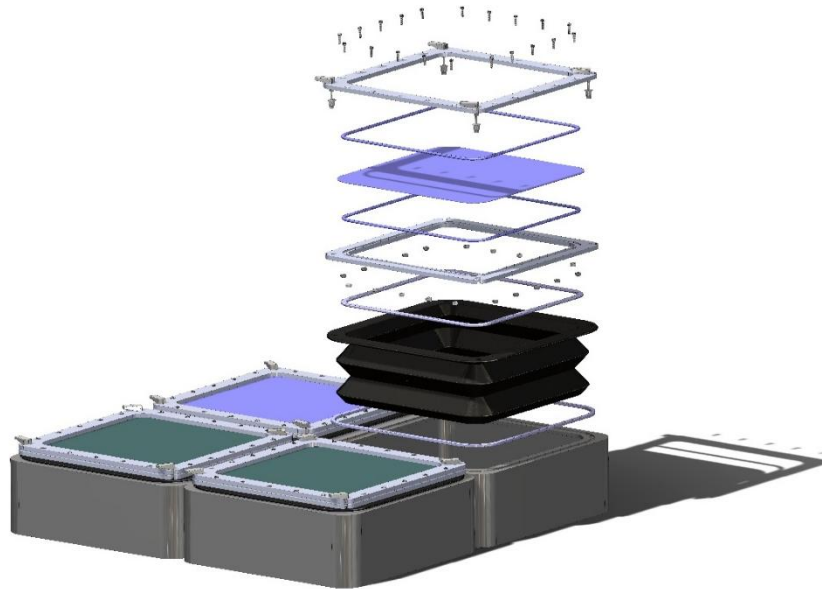


Figure 2. Acoustic Damper Assembly Exploded View.

Based on the depth of the sealed air cavity behind the membrane, the membrane absorber will be able to

suppress a specific select frequency. To actuate the design, first the frequency with the highest decibel rating will be determined by using a frequency sensor in the control subsystem.

A fluorosilicone housing will be used for the reverberation chamber as it is most optimal in terms of balancing weight, cost, and effectiveness. Additionally, since fluorosilicone is inherently flexible, slightly thinner material at the edges of the accordion structure will allow the design to bend in the desired way when the mounting plate is actuated. Fluorosilicone is known for its durability and flexibility in aerospace applications [4], making it the perfect candidate for the housing material [4].

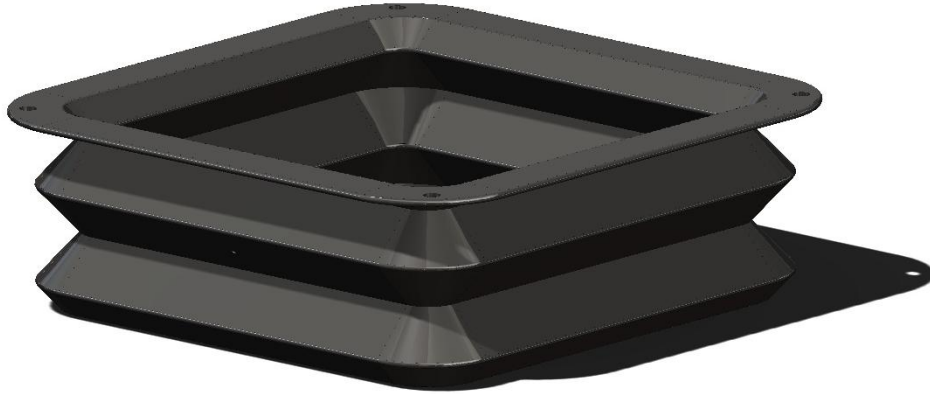


Figure 3. CAD Design of Accordion Housing Structure.

In addition, the gasket sealing the acoustic barrier to the housing will also be manufactured with fluorosilicone, as its flexible nature combined with mechanical fasteners attaching the subsystem components together will be crucial for creating an effective pressure seal. Fluorosilicone also has the desired range in density that is ideal for targeting the desired low frequencies

The MASS control system is based on tuning the chamber's geometry to match the dominant acoustic conditions in the environment. Since the membrane resonant behavior depends on chamber depth, pressure, and membrane properties, the controller uses the measured dominant frequency as the input for determining the desired resonant configuration. The goal is to shift the system's natural response so that it more effectively targets low- frequency noise.

The system can be modeled as a finite state machine due to its operations divided into control states. These include initialization, sampling, frequency identification, target depth calculation, adjustment, hold, and fault handling. This structure helps prevent unstable behavior by ensuring that the system only moves when a meaningful change in acoustic input is detected. For potential future improvement a deadband range can also be used so small frequency fluctuations do not cause continuous actuator motion.

The current control approach is primarily feedforward, meaning the measure of frequency is used to predict the chamber for the depth needed for suppression. A possible improvement would be to add closed loop feedback, where the system compares sound levels before and after adjustment and continues tuning until noise reduction is achieved. Additional improvement could include FFT averaging, adaptative frequency to depth mapping, and position feedback to account for backlash, missed steps, or changing environmental conditions.

## 2.6 Design Constraints for NASA Adoption

The implementation of MASS aboard the Artemis Lander is constrained by strict limitations on volume, mass, power consumption, and reliability. Due to minimal habitable volume inside the Artemis Lander, the protrusion of the design into crew space has been minimized so that it can be stowed efficiently near ECLSS hardware [13]. When dealing with implementation of multiple units in a larger system, compact geometry is very favorable. This directly motivated the reduction of maximum cavity depth from 0.305m to 0.205m. The total system weight has also been constrained to remain below 100 lb and below 1.5 kW of power consumption to remain feasible for long duration spaceflight and implementation.

Another large constraint for NASA adoption is long-term system reliability and maintainability. In alignment with other spacecraft noise reduction efforts, the MASS system must function quietly, efficiently, and reliably for extended durations on future Artemis missions while exposed to thermal cycling, vibration, and continuous operation [12]. To mitigate potential failure modes, such as actuator fatigue, membrane degradation, and loss of cavity seal integrity, the linear actuators are positioned externally to allow ease of replacement and maintenance. Additionally, the membrane and mounting plate displace very minimally during operation, which reduces cyclic fatigue of the system and improves the expected operational lifetime.

The tuned membrane absorber depends on maintaining a sealed air cavity during operation to generate a restoring force necessary for resonance-based noise suppression. Since cavity depth varies during operation, maintaining a constant pressure seal, while also allowing cavity pressure to equalize to ambient pressure during adjustment, are equally critical design challenges. Fluorosilicone was chosen as the membrane and gasket material, as its ability to deform while maintaining a pressure seal makes it well suited for the adjustable reverberation chamber used in the MASS design. The material's ability to resist thermal and environmental degradation also makes it compatible for the harsh conditions of space. To allow cavity pressure to equalize, the base plate contains a small central aperture that is sufficiently small to minimize acoustic leakage during operation while still allowing slow pressure equalization during chamber adjustment.

## 2.7 Analysis

When incident sound waves excite the membrane, it behaves like an oscillating mass. The sealed air cavity provides a restoring force due to compression of the trapped air [3]. Under small oscillations, which can be assumed for sound waves, the pressure change in the cavity follows the relation:

$$\Delta P = -\gamma P_0 \frac{\Delta V}{V} \quad (1)$$

where  $\gamma$  is the ratio of specific heats for air ( $\approx 1.4$ ),  $V$  is the cavity volume,  $P_0$  is the ambient air pressure ( $P_a$ ), and  $\Delta V$  is the change in cavity volume due to membrane displacement. For a membrane moving with displacement  $x$  over area  $A$ :

$$\Delta V = Ax. \quad (2)$$

This results in an effective acoustic spring constant per unit area, where  $d$  is the sealed air cavity depth ( $m$ ):

$$k' = \frac{\gamma P_0}{d}. \quad (3)$$

The membrane surface mass density  $m_s$  ( $\frac{kg}{m^2}$ ) acts as the oscillation mass per unit area. The system therefore behaves as a simple harmonic oscillator with natural frequency:

$$\omega_0 = \sqrt{\frac{k'}{m_s}}. \quad (4)$$

Substituting for  $k'$ , the equation becomes:

$$\omega_0 = \sqrt{\frac{\gamma P_0}{m_s d}}. \quad (5)$$

Converting frequency in Hz, the final equation is:

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{\gamma P_0}{m_s d}}. \quad (6)$$

This expression defines the resonance frequency of the membrane absorber.

Next, the required cavity depth was calculated, and the corresponding material was selected. Rearranging Equation (6) provides a direct design relationship for cavity depth:

$$d = \frac{\gamma P_0}{(2\pi f_0)^2 m_s}. \quad (7)$$

This relationship shows that increasing membrane surface mass, which is the product of material density and thickness, decreases the required cavity depth. Due to the inverse square relationship, it can be ascertained that lower target frequencies require significantly larger cavity depths as depth varies proportionally with  $\frac{1}{f^2}$ . Using the cavity depth expression in Equation 7, the required depth for a suppressible frequency can be calculated within the frequency operating range. This enables active tuning of the absorber by adjusting the mounting plate position.

Next, the acoustic impedance and absorption coefficient were calculated. To estimate the percentage of incident sound energy absorbed, the system can be modeled using the surface acoustic impedance. The surface impedance of the membrane-cavity system is:

$$Z_s(\omega) = R_{loss} + j\omega m_s - j\frac{\gamma P_0}{\omega d} \quad (9)$$

where  $R_{loss}$  is the effective damping term, and  $\omega = 2\pi f$ . The normalized impedance is:

$$z = \frac{Z_s}{Z_0} \quad (10)$$

where  $Z_0 = \rho_0 c$  is the characteristic impedance of air. The reflection coefficient at normal incidence is:

$$R = \frac{z - 1}{z + 1} \quad (11)$$

The absorption coefficient is then:

$$\alpha = 1 - |R|^2 \quad (12)$$

At resonance, the reactive terms cancel, and if the resistive term is tuned near  $Z_0$ , the absorber approaches maximum energy absorption [7].

Since resonance depends on cavity depth  $d$ , adjusting the mounting plate directly changes the tuned frequency according to:

$$f \propto \frac{1}{\sqrt{d}} \quad (13)$$

Thus, by mechanically varying cavity depth, the system can shift its peak absorption across the desired frequency range. The mathematical relationship implemented in the control subsystem allows the actuator to position the mounting plate based on real-time frequency measurements.

This analytical framework confirms that the proposed adjustable membrane absorber can theoretically target and suppress low-frequency bands through controlled geometric reconfiguration. To facilitate this reconfiguration, a linear actuator/linear rail system was implemented that will be attached to the mounting plate. The system will include two linear actuators and four linear rails, with each actuator placed on opposite sides of the mounting plate and two linear rails additionally placed on each of these sides. In an ideal scenario an electromechanical actuator that can reduce ambient noise production by the actuators would be used. A future iteration of the design might include a second membrane or a base plate with a lower effective stiffness. While this does significantly increase weight and system complexity, this would

further reduce the required cavity depth to suppress frequencies in the target range.

### **3. Verification, Validation and Risks**

#### **3.1 Model and Simulation**

Fans on the ISS currently generate a range of frequencies [5, 10, 11, 12]. Many implemented sound suppression solutions engineered to maintain the general NC-50 acoustic levels prioritize higher frequency noise, but low-frequency noise from ECLSS hardware, specifically fans and rotating components, have the potential to propagate into the habitable volumes of the lunar lander and result in a chronic stress factor for crew [6, 8, 9]. That background noise both increases fatigue risk and has the capacity to directly degrade mission performance by masking speech cues, reducing intelligibility, and making comms more error prone. These factors are exacerbated for the small reverberation prone environment of the lunar lander. NASA's crew acoustics guidance ties acceptable continuous background noise to NC limits but explicitly does not allow hearing protection as the method of compliance, which is why passive, built-in noise control presents a compelling solution [8].

Based on the selected membrane material, the required depth to achieve optimal decibel reduction at a specific frequency varies. A denser material requires less depth to reduce a specific frequency, but also suppresses a narrower band of frequencies, while a less dense material requires more depth and suppresses a wider band [3]. The bounds on the adjustability of the design have been set to a minimum of 0.05 meters when the accordion structure is fully compressed, and a maximum of 0.205 meters when the structure is fully outstretched. The maximum depth of the reverberation chamber was reduced from .305 meters to .205 meters since our initial proposal in order to reduce empty cavity space. This smaller required cavity depth additionally decreases the amount the wall mounted panel protrudes into the room, thus saving cabin space and increasing the feasibility of implementation. 3 mm fluorosilicone was chosen as the membrane material for the chamber, with the relationship between cavity depth and suppressible frequency range shown in the plot in Figure 4. According to the simulation, the design can suppress a range of 60-170 Hz at depths ranging from 0.05 m to 0.205 m.

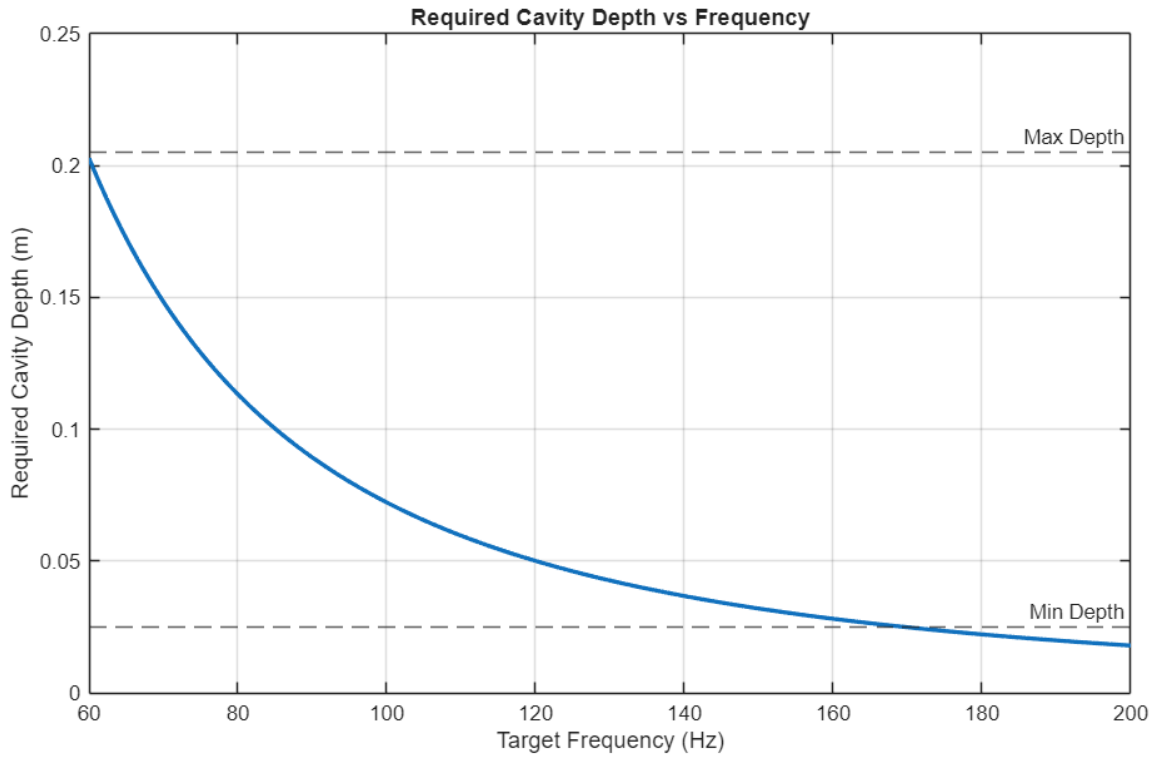


Figure 4. Plot of Required Cavity Depths for 3 mm Fluorosilicone Membrane to Suppress the Target Frequency Range.

Even though fluorosilicone does not cover the entire target range of 60-200 Hz, two different membrane materials or material thicknesses can be used in the larger system of absorbers to encompass the entire range of frequencies. In this case, 2 mm and 3 mm fluorosilicone can be used as the two materials needed to cover the target range. The corresponding range of suppressible frequencies for each material can be seen in Figure 5 below.

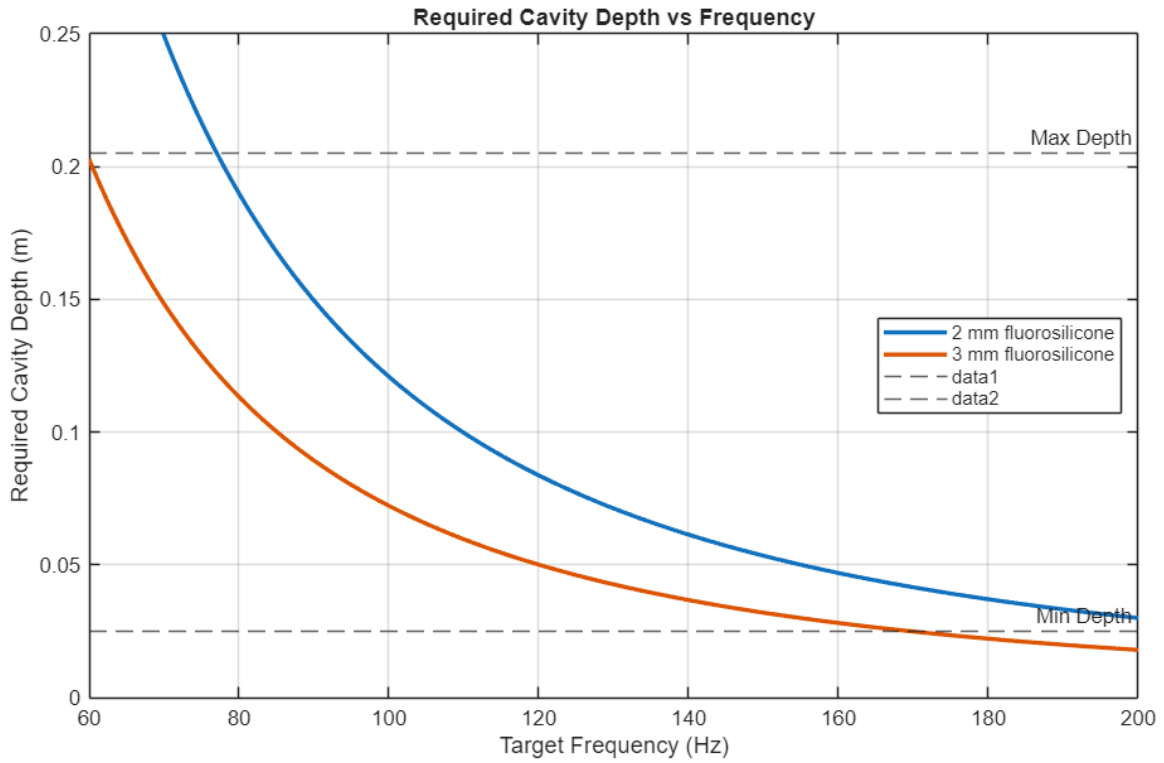


Figure 5. Plot of Required Cavity Depths for 2 mm and 3 mm Fluorosilicone Membrane to Suppress the Target Frequency Range.

3 mm fluorosilicone spans a range of 60-170 Hz, while 2 mm fluorosilicone spans a range of 76-200 Hz. Different materials that cover the target range can be used interchangeably.

### 3.2 Feasibility Prototype Validation

Three tests were conducted on a feasibility prototype of MASS: a linear actuator and electronic verification test, a pressure equalization verification test, and a vibration test. This feasibility prototype was manufactured using slightly different materials, the most significant changes being TPU for the housing and mass loaded vinyl for the membrane, but the tests performed showcase the functionality and feasibility of MASS on a larger scale.



Figure 6. Feasibility Prototype CAD Model.

The linear actuator and electronic verification test performed a sweep for frequencies ranging from 60-250 Hz. This ensured that the actuators adjusted to changing frequencies. The developed control system successfully smoothed frequencies to enable consistent transitions despite highly variable sound conditions and could successfully actuate in both directions for the whole height range. Photos of the testing setup procedure are shown in Appendix A.

The pressure equalization verification test shifted the chamber depth three inches to verify pressure equalization at varying depths throughout five separate tests. The chamber pressure successfully equalized after an average of 4.49 seconds. Chamber pressure equalization was obtained with a slight gap in the seal for the prototype, but for NASA's purposes, the seal would be completely enclosed. In this case, chamber pressure equalization would be obtained through a small orifice at the base of the chamber. Through calculations shown in Appendix B, it was found that a hole diameter of 13 mm would be sufficient.

The vibration test was conducted to verify that the MASS prototype could withstand structural integrity under launches like dynamic vibration conditions. Since launch vehicles subject payload to broadband random vibration, the prototype was mounted to a shake table and subjected to a random vibration profile sweep from 20-2000 Hz. This range was selected to represent the dynamic loading that the system could experience during launch and to identify whether any structural loosening, failure, or instability which could occur during a low, mid, and high frequency excitation. The test was performed in stages, increasing from 25% to 50%, 75%, and finally 100%. After the ramp up sequence, the prototype was held at the full test level for 5 minutes. The final vibration profile reached an RMS acceleration of approximately 10.3 g, and the schedules test completed successfully without premature shutdown.

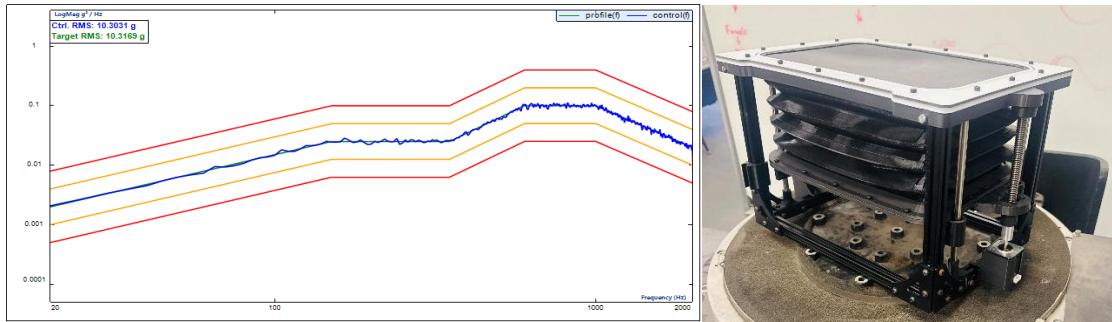


Figure 7. Testing Apparatus and Decibels Versus Frequency Graph for the Vibration Test.

The test was used to evaluate structural stability rather than acoustic performance. The prototype held stable during the duration of the test, with the only issue being loose hex nuts after the vibration test. It was determined that lock-nuts would solve this issue; this change was reflected in our budget. Future testing should repeat the procedure using final flight material and include posttest inspection of loosening, deformation, actuator misalignment, membrane damage, and seal degradation.

### 3.3 Risks

NASA’s upcoming sustained lunar operations include mission durations growing from about 7 days to 30-45 days as well as missions to Mars lasting 1200 days [15]. This long-term stay increases the importance of noise protection for astronauts and the need to ensure all possible risks are mitigated [8, 11]. Possible causes and solutions regarding maintenance and longevity of the design have been considered to enable reliable noise suppression. To analyze any risks posed by integrating MASS into the human lander, a risk management table, which can be seen in Appendix C, and matrix shown in Table 1 were developed. Through this analysis, six critical risks were identified.

Table 1. MASS Risk Matrix.

Probability	5					
	4			8	3	
	3			7		2,4,5
	2			9		1,6
	1					
		1	2	3	4	5
		Consequence				

The greatest risks for this system are related to functionality in terms of acoustic damping, with a high-pressure differential (Risk 3) being the most critical risk based on the matrix. The system’s ability to resonate with target frequencies is highly dependent on maintaining an internal pressure that is equal to the surroundings. If the pressure does not equalize then the device resonant frequency will not match the dominant frequency in the surroundings. To mitigate this, the current design incorporates an orifice at the base of the reverberation chamber. This will allow air to escape and return as the system actuates, adjusting the volume of air inside as the height changes and keeping the pressure stable. Extensive

analysis and testing will be done in future design phases to ensure that the hole size is suitable in terms of recovery time and equalization. Maintenance will be required at regular intervals that includes ensuring the orifice is still opening and is not blocked.

The next risk with a high consequence rating is electrical system failure (Risk 2). Frequent or long-term actuator motion could lead to wire fatigue, loose connections, or actuator motor wear. Additionally, vibration from launch loads could damage these same components. Strategies to mitigate electrical failure include using space-rated wiring and cables, providing strain relief on all moving cables, and mounting the PCB in a way that is damped from vibration. Additionally, current limits should be implemented in the actuators and a fail-safe integrated into the system that allows the device to remain at a default constant chamber depth that corresponds to the dominant, most common ECLSS frequency on the human landers.

There may also be further failures relating to the positioning and accuracy of the identified frequencies. Actuator positioning error (Risk 4) presents a challenge since improper chamber depth would shift the resonant frequency away from the targeted acoustic band. This could result in uneven actuator motion, a drift in calibration accuracy, and inability to suppress the current dominant frequency. To mitigate this risk, MASS will have actuator positioning feedback incorporated into its architecture, periodic calibration checks, and software travel limits to ensure the operation remains within the allowable chamber range of depth. In addition to the mechanical positioning accuracy, the effectiveness of the control system is dependent on reliable acoustic sensing. Acoustic monitoring inaccuracy (Risk 5) could cause the system to tune toward an incorrect dominant frequency which in turn would reduce the suppression performance of the system or cause unnecessary actuator movement. This risk would be mitigated through careful sensor calibration and testing of FFT averaging techniques. The microphone monitoring the surrounding noise would go through extensive validation testing and will be revalidated periodically.

Excessive mass (Risk 7) and excessive volume (Risk 8) are also important risks to consider when implementing MASS into the HLS. Additional payload weight and intrusion into habitable crew space directly impacts mission feasibility so reduction of weight and volume is critical. This design mitigates these risks through lightweight material selection, reduction of unnecessary structural components, and the optimization of cavity depth to minimize protrusion into the space. The maximum cavity depth was reduced by 0.105 m which improved the compactness of the system while still maintaining the target suppression frequency range. Continued design refinement and structural optimization would further reduce system mass and occupied volume before it is implemented.

#### **4. Implementation**

While the MASS design shown in the Quad Chart consists of one panel, the full-scale implementation of this design on NASA's Artemis would utilize multiple panels with separate sensors to reduce frequency decibel levels from separate sources. The first step towards implementation is a budget assessment to ensure that the cost of this project will fall within a reasonable budget.

#### 4.1 Budget

When developing Table 2. which represents a lifecycle budget for developing one flight ready MASS unit using a 10-person engineering team over a three-year period. The largest cost driver is the personnel, with annual labor held constant at \$1.26 million based on ten full-time members at \$126,000 per year. Direct costs were separated from personnel to demonstrate the expected expense for prototype hardware, testing facilities, and final flight hardware. A 30% cost margin is included in account of design changes, added testing, procurement uncertainty, and integration risk. Overall, the projected lifecycle cost is approximately \$5.63 million, which provide a realistic estimate for a finalize MASS from prototype to flight ready.

Table 2. MASS Implementation Budget Summary.

Mission Phase	Year 1: Dev Phase (2026)	Year 2: Test Phase (2027)	Year 3: Fab & Impl. (2028)	Total (\$)
<b>PERSONNEL</b>				
Engineering & Design	900,000	600,000	500,000	2,000,000
Fabrication & Assembly	250,000	160,000	360,000	77,0000
Testing & Inspection	110,000	500,000	400,000	1,010,000
Total Personnel(12)	1,260,000	1,260,000	1,260,000	3,780,000
<b>DIRECT COSTS</b>				
Prototyping Hardware	100,000	50,000	0	150,000
Testing Facilities & Equip.	0	150,000	100,000	250,000
Final Hardware (1 Unit)	0	0	150,000	150,000
Total Direct Cost	100,000	200,000	250,000	550,000
<b>FINAL COST CALC.</b>				
Total Projected Cost	1,360,000	1,460,000	1,510,000	4,330,000
Total Cost Margin(-30%)	408,000	438,000	453,000	1,299,000
<b>Total Project Cost</b>	<b>1,768,000</b>	<b>1,898,000</b>	<b>1,963,000</b>	<b>5,629,000</b>

The annual operating cost was determined using the estimated power input, mission time, system run rate, and the current power market rate. It is assumed that the MASS system would be actuated and run for one minute of every hour of the mission, the power usage would be 1.36 kW, and the current market rate was \$0.12 per kilowatt-hour. Two values were determined: one for a 30-day long mission to the moon, and one for a 1200-day long mission to Mars.

The cost for a 30-day long mission is \$58.75, and the cost for a 1200-day long mission is \$2,350.08. This is a reasonable operating cost for the benefit of astronaut comfort and focus during Artemis Human Lander missions to both the moon and Mars.

## 4.2 Path-to-Flight Project Timeline

A three-year plan is proposed below to successfully implement the MASS system into NASA's ECLSS, with the assumption that a mission will happen in the next 5-8 years. This plan includes the development and testing of the product as well as evaluation of the final system.

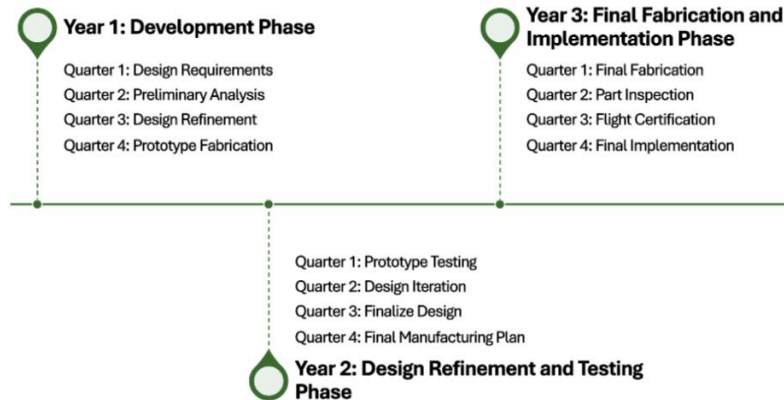


Figure 8. Project Timeline

Year 1, the development phase, focuses on the design requirements of the system. This will be CAD refinement from the Acoustic Artemis design, as well as material testing and final material selection. Year 1 will conclude with prototype fabrication. Year 2, the design refinement and testing phase, focuses on testing of the functional prototype, especially testing focused on launch fatigue aerospace thermal dynamics. Year 2 concludes with a final plan for manufacturing. Year 3, the final fabrication and implementation phase, focuses on the final fabrication of the MASS units being utilized for long-duration space flights. Year 3 concludes with the final implementation of the system.

## 5. Conclusion

The MASS concept shows the feasibility of a tuned membrane absorber for mitigating low frequency noise generated by the ECLSS aboard the Artemis Human Lander. Through simulation modeling, material selection, and prototype validation, the MASS system was shown to theoretically suppress frequencies in the 60-200 Hz range while remaining within NASA volume, mass, and power constraints.

The required cavity depth for MASS was calculated as a function of frequency and membrane surface mass density. Initial analysis resulted in a required cavity depth of up to 0.305 m for a 3 mm neoprene membrane. However, by increasing membrane surface density through a fluorosilicone membrane material selection, and by using two different material thicknesses with overlapping operating ranges to span the entire target range, the required cavity depth was reduced to 0.05 m. This significantly improved compactness and reduced protrusion into habitable cabin space.

Prototype testing verified the critical subsystems of MASS. The actuator and control verification test successfully demonstrated automatic adjustment of the reverberation chamber across the target frequency range. Pressure equalization testing showed that the chamber could stabilize internal pressure after a change in cavity depth with an average equalization time of 4.49 seconds. Vibration testing exposed the prototype to various vibration profiles ranging from 20-2000 Hz with RMS accelerations approaching 10.3 g, which validated structural integrity under simulated spacecraft launch conditions.

This project demonstrates that MASS is a feasible solution for reducing low frequency ECLSS noise onboard future Artemis missions. Future research should focus on experimental acoustic chamber testing to quantify insertion loss across the target frequency range. Additionally, future development to the design should include lightweight structural optimization, refinement of control architecture, and environmental

testing under relevant thermal and vacuum conditions. With further refinement and validation, MASS has potential to improve comfort, communication, and habitability for astronauts during long-duration missions.

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**Appendices:**

**A Testing Appendix: 100-200 Hz Frequency Sweep Reception Test**

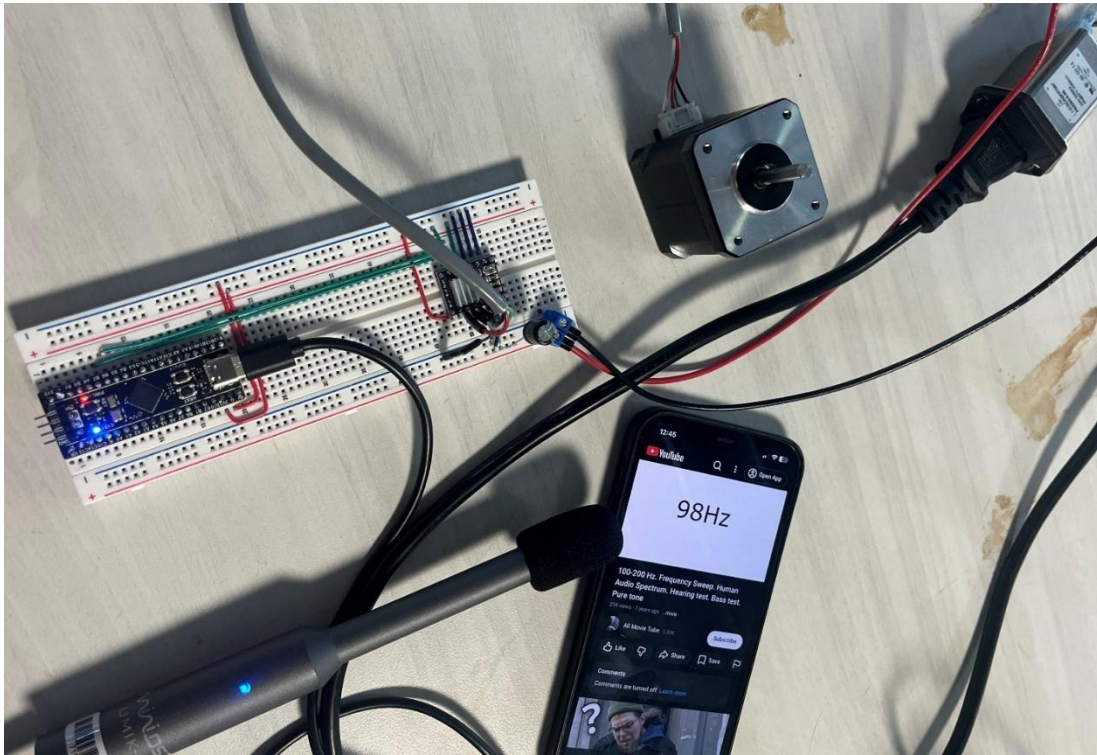


Figure 9. Frequency Sweep with Microphone Connected to STM32 Board to Control Stepper Motor Function

## B Calculations Appendix:

To calculate the diameter of the required hole, the equation for flow rate through an orifice is needed:

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}}$$

where  $Q$  is the volumetric flowrate in  $\text{m}^3/\text{s}$ ,  $C_d$  is the discharge coefficient,  $A$  is the hole area in  $\text{m}^2$ ,  $\Delta P$  is the pressure difference in Pa, and  $\rho$  is the density of air [16]. For a circular hole, the area can be calculated as  $A = \frac{\pi d^2}{4}$ . Solving for the hole diameter,  $d$ , the following equation is obtained:

$$d = \sqrt{\frac{4Q}{\pi C_d \sqrt{2\Delta P/\rho}}}$$

For an assumed pressure difference of 100 Pa, density of  $1.2 \text{ kg}/\text{m}^3$ , discharge coefficient of 0.6, and flowrate of  $0.001 \text{ m}^3/\text{s}$ , a value of 13 mm was obtained for the orifice diameter.

**C Risk Appendix:**

ID	Risk Name	Description	Related Systems	P	C	Plan
1	Membrane Resonation Failure	Membrane fails to resonate at target frequency	Mechanical, HLS	2	5	Complete analysis and testing of membranes before integration.
2	Electrical System Failure	Electrical components stop functioning causing linear actuators to no longer move eliminating noise dampening adjustability	Electrical, Control	3	5	Use electrical and wiring components rated for space, design fail-safe mode where MASS remains fixed at default chamber depth for dominant frequency found in lander during testing
3	High Pressure Differential	Pressure within the internal reverberation chamber is largely different from the surrounding external pressure	Mechanical, Control	4	4	Include a controlled pressure equalization aperture, validate sealing behavior during chamber adjustment, emphasize aperture inspection during maintenance
4	Actuator Positioning Error	Linear actuators move the mounting plate to the incorrect chamber depth	Mechanical, Control	3	5	Add actuator position feed back, do calibration checks, implement software limits to prevent movement outside allowable depth range

5	Acoustic Monitoring Inaccuracy	Acoustic sensors inaccurately identify the dominant frequency within the lander	Electrical, Control	3	5	Implement extensive sensor calibration, frequency filtering, and acoustic measurements before actuator movement; extensive acoustic testing and programming
6	Mounting Failure	Device detaches, loosens, or vibrates from its mounted location during launch or operation	Mechanical, HLS	2	5	Design mounting brackets for launch loads, use vibration resistant fasteners, perform structural and vibrational testing before system integration
7	Excessive Mass	Device mass exceeds critical point where it interferes with launch loads	HLS, Propulsion	3	3	Track device mass throughout design, use lightweight materials, reduce nonessential structural components
8	Excessive Volume	Device volume takes up excessive space within the HLS interfering with astronaut living and working space	HLS	4	3	Minimize protrusion into crew space through optimization of cavity depth and material selection
9	Power Consumption	System requires excessive power for sensors, controls, and actuation	Electrical, HLS	2	3	Limit the duty cycle of the actuators, use low power sensors, ensure it is only operational when frequency retuning is required