International Space Station Acoustic Noise Control - Case Studies

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1. INTRODUCTION

A team is in place to oversee acoustics of International Space Station (ISS) modules, payloads, and Government Furnished Equipment (GFE). The team commits dedicated attention to acoustics in the early design stages, provides technical support, and helps ensure acoustic compliance. This paper outlines several noise abatement projects that the team has participated in. Excessive noise levels from machinery or equipment may affect hearing acuity, speech intelligibility, habitability, safety, productivity, annoyance, and sleep interference, prompting the need for additional acoustic noise control measures. Noise abatement case studies include the quieting of the U.S. Airlock depress pump and heat exchanger, quieting of the Express Rack payload, and support efforts on the Japanese Centrifuge Rotor (CR). Diagnostic noise control measures for fan noise, flow noise and structural borne radiated noise in the payload rack for the European Microgravity Science Glovebox (MSG) are addressed. It is concluded that noise control is most beneficial and cost effective when implemented as early as possible in the design process.

2. U.S. AIRLOCK DEPRESS PUMP AND HEAT EXCHANGER

A. Problem Description and Performance Requirements

The Joint Airlock is a pressurized flight element consisting of two cylindrical chambers attached end-to-end by a connecting bulkhead and hatch. The Airlock is the primary path for International Space Station space walk entry and departure for U.S. spacesuits, which are known as Extravehicular Mobility Units, or EMUs. The Joint Airlock is designed to also support the Russian Orlan spacesuit for EVA activity. A combination of the Russian depress pump and pressure equalization valves located within the hatches accommodate the depressurization/ pressurization capability of the airlock, without major loss of environmental consumables such as air.

The original acoustic requirement for the Russian depress pump for use in the U.S. Airlock was NC-40. NASA verified by testing that this requirement was not met. The A-weighted sound pressure level (SPL) of the airlock pump, measured inside the airlock module, was approximately 100 dBA. The equivalent sound power level (PWL) was 95 dB. The pump featured a muffler on the outlet but not on the inlet. The overall PWL of the unmuffled inlet was measured at 101 dB. The ISS acoustic team started a project to develop noise control measures, which would attenuate the pump generated noise. A proof-of-concept demonstration of the noise abatement procedures and materials was identified as a project requirement. Other requirements included that the redesign should not impact the pump and/or motor assembly design or performance and vehicle mounting interfaces. There should be a capability to perform In-Flight Maintenance (IFM) on the pump and motor assembly. No modifications were permitted to the original hardware, including stowage box, unless they were readily “reversible.” Final designs and materials should be compatible with ISS use in the airlock and with the pump/motor assembly. The overall weight of the noise abatement hardware could not exceed that of the delivered Russian pump and container. The center-of-gravity of the noise abatement kit should be within the envelope of the limits defined by the pump assembly package, or accepted deviations should be worked out.

B. Noise Control Measures

Several approaches were employed to reach the noise abatement design goal. An airlock “quieting kit” was implemented into flight hardware changing the vehicle installation callout to Government Furnished Equipment (GFE). A muffler was designed for the airlock pump inlet. Acoustic covers were applied to water and air lines with Velcro tape to reduce flow noise. Vehicle close out panels, which access the airlock depress pump installation, were treated with acoustic seal and foam applications.

The ISS “GFE Airlock Depress Pump Quieting Kit“ consists of the hardware/configuration that serves the following functions:

- supports the pump structurally and provides a means of attaching the pump to the airlock structure
- isolates the pump from vibrations by the base plate it attaches to
- provides an enclosure made of fabric, foam, and barrier material to attenuate/absorb noise emanating from the pump
- includes a metal box enclosure to support the fabric, foam, and barrier enclosure
- accommodates IFM access; and provides pass-troughs for all fluid and electrical lines that connect to the pump in support of it’s operating functions.

The most important noise reduction feature of the quieting kit was the use of vibration isolators. The isolators must be used with a snubbing washer to ensure that the isolator retains a good structural load capability in tension. Four commercial isolators were used, one each in the outboard locations of the pump mounting pattern. Based upon discussions with the manufacturer each isolator can take over 500 lbs. of load without yield failure when the snubbing washer is used.

The bottom of the metal enclosure box mates to the base plate. Cutouts in the box are minimal and consist of pump installation clearances, and variances in Russian pump hardware and the drawings. The seal between the top and bottom must be acoustically effective. The lower portion of the box provides for mounting of Russian electrical pass-through plates, which must be acoustically sealed.

An acoustic enclosure was constructed (Figure1) with the six sides consisting of acoustic foam and barrier material between nomex liners. The acoustic enclosure fits in the metal box. Its function is to block and absorb pump-radiated noise. All openings in the liner material were acoustically sealed.

C. Test Results

The overall sound power level of the inlet was measured at 101 dB. 24 dB attenuation was achieved by installing the newly designed muffler. The sound power levels of the inlet before and after installation of the muffler are plotted as function of the octave band center frequency in Figure 1. The overall sound power level of the original pump configuration was 95 dB. After installation of the “GFE Airlock Depress Pump Quieting Kit” the overall sound power level was attenuated by 23 dB down to 72 dB. The sound power levels of the original pump and GSE pump configurations are plotted as functions of the octave band center frequency in Figure 1. Finally, the A-weighted sound pressure level measured at the center of the equipment lock was reduced by 27 dB, from the initial 100 dBA down to 73 dBA. Measurements were conducted with inlet and outlet mufflers installed and all closeout panels in place. The 73 dBA is the maximum allowable sound pressure level for a 20 minute intermittent noise during a 24-hour period according to the requirements for ISS pressurized payloads.

3. EXPRESS RACK PAYLOAD

A. Introduction

The EXPRESS (EXpedite the PRocessing of Experiments to Space Station) rack system was developed to accommodate a variety of scientific experiments. Eight EXPRESS racks are being built for use on the International Space Station.

Four EXPRESS payloads were evaluated for compliance with acoustic emission limits. The payloads included the Commercial Generic Bio-Processing Apparatus (CGBA), an experiment used for studying long-duration space flight effects on the fermentation process; the Commercial Refrigerator Incubator Module (CRIM), an incubator used for other experiments; the Plant Generic Bio-Processing Apparatus (PGBA), an experiment used for growing Loblolly pine tree seedlings; and the Protein Growth Chamber – Single Thermal Enclosure (PCG-STES), an experiment used for growing large protein crystals. The evaluation data showed exceedance from 3 dBA to 6 dBA over allowable limits. The goal of the ISS acoustics team was to design and install acoustic mufflers that would substantially attenuate the acoustic emissions of these EXPRESS payloads.

B. Muffler Designs

Individual mufflers were designed for each payload. The EXPRESS racks had protrusion limits of 3.5 to 6 inches from the face of the payload. The payloads themselves had design restrictions due to airflow requirements, pressure drop limits and interference with hoses and cables locations. The exterior shells of all mufflers were constructed of 6061-T6 Aluminum. Velcro was used to attach the mufflers, on orbit, to the face of the payload, while the inlets and outlets were sealed with an elastomer material for a tight gasket interface.

The interior of the mufflers had two basic designs. The first design, for the CRIM and the STES, had the fan noise flowing straight into the muffler after it was re-directed toward one side. The entire cavity was lined with melamine foam, which is an open cell foam with excellent sound absorption properties and good materials outgassing and flammability characteristics. There was a melamine lined baffle in the middle, and the exits were staggered. This forced the sound to impinge on the foam for absorption. Blocking the line of sight and adding absorption seemed to be the most effective and efficient way to reduce payload fan noise. The other two designs, for the CGBA and the PGBA payloads, used a lamination of melamine foam and a thin, perforated aluminum plate to impede both the high and low frequency noise. Increases in transmission loss were observed by adding the aluminum plate, which was in addition to the absorption by the layer of porous absorption material. For the PGBA, a block of foam roughly 14.5” x 3.5” x 2” was placed directly in front of the flow, so that the vent flow was forced through the foam. An angled orifice pattern was cut to block the line of sight and increase sound absorption. This type of design was a spin-off of a Hamilton Standard fan design for an ISS AAA fan. Figure 3 shows the orifice pattern for the PGBA payload muffler. The CGBA had a similarly designed hole pattern. The CGBA muffler also used a polymer for spacers and velcro to isolate vibration and thermal conductance between the inlet and outlet side mufflers (Figures 4 and 5).

Several unique types of prototype mufflers were built and tested before settling on the final design. Figure 6 shows the results of measured sound pressure levels for prototype mufflers designed for the STES payload. The unmuffled STES payload produced a strong tone at 331 Hz (Figure 6). The first muffler was designed to be a Helmholz resonator, tuned to the 331 Hz tone. Muffler 3 was just a simple expansion chamber acting as a reactive muffler. Muffler 6 featured a foam-core shell, lined with melamine foam lined. Muffler performance would vary with frequency and final designs were chosen for their ability to minimize overall A-weighted sound pressure levels.

C. Measurements and results

Testing of the prototype mufflers was conducted in a semi-anechoic chamber. The test apparatus consisted of two 6” x 9” speakers enclosed in a box, constructed of birch wood and medium density fiberboard. The front face was interchangeable to simulate different payload configurations. Three microphones were placed in front of the test fixture. A microphone was placed 2 feet away, directly in front of the mockup payload’s inlet. Data was also acquired at two other microphones and other locations but the results are not reported here. The levels of the test fixture were set by a hand held meter to verify the correct output levels of the model payloads. The first set of tests simulated the payloads without mufflers, with simulated levels based upon acoustic testing of actual payloads. The second test measured the sound pressure levels after the muffler was attached to the front face with Velcro. A computer drawing in Figure 7 shows two of the three CGBA mufflers attached to the model payload. The sound pressure levels were acquired over a 20 second time frame as narrow band, time-averaged data. Through mathematical functions internal to the system. The data was post-processed to yield 1/3 and 1/1 octave bands for further analysis, illustration purposes and comparison with the requirements. An overall A-weighting was also calculated for the acquired acoustic data.

The insertion loss of the mufflers was obtained by subtracting the muffled model payload sound pressure levels from the sound pressure levels measured from the model payload by itself. The attenuation due to the muffler is listed in Table 1 for all four model payloads including the inlet and outlet mufflers for the PGBA. The mufflers performed well for all octave bands from 63 Hz to 8000 Hz, except in the 250 Hz octave band where the mufflers coupled to the structural-acoustic vibration of the test fixture. Increased performance is expected at that frequency for the actual EXPRESS rack configuration. The muffler insertion loss was then applied to the predicted on-orbit levels for each payload. The expected sound pressure levels for the four muffled payloads are compared with the NC-40 requirements in Figure 8. All payloads met the NC-40 requirements except for the STES payload at 500 Hz and the PGBA payload at 250 Hz and 500 Hz. The timely design and testing of the prototype mufflers for the EXPRESS rack payloads showed their effectiveness in attenuate excessively high noise levels.

4. CENTRIFUGE ROTOR

The ISS houses a suite of biological research specimen support equipment that collectively constitute the Gravitational Biology Facility (GBF). Housed within the Centrifuge Accommodation Modules (CAM), the GBF supports research on how the space environment affects a broad range of biological systems. The centerpiece of the GBF is a 2.5 m (8.2 ft) diameter Centrifuge Rotor (CR) that accommodates multiple biological habitats for maintaining a variety of biospecimen types, from cells to rodents to large plants. As the centrifuge rotates, artificial gravitational forces are produced upon the attached habitats that house various biological specimens. Accelerations ranging from 0.01 g to 2.0 g will permit scientists to compare how differing gravity levels affect the biology of organisms housed in habitats under otherwise identical conditions, thus separating the effects of gravity from other factors in the space environment. The centrifuge provides life support resources and electrical power to the habitats as well as data transfer links to ISS systems and to the ground. The hub, or center, around which the centrifuge rotates provides structural support for the rotating part of the centrifuge, and it provides life support to the specimen habitats.

A centrifuge rotor acoustic analysis report and noise control plan were developed by the National Space Development Agency (NASDA) of Japan in the early stages of the project to ensure compliance with the payload acoustic requirements of NC-40, measured at 0.6 m from the CR front surface at the loudest location. Both the analysis and noise control plan are continuously updated to accommodate changes in the design and incorporate the latest development, material and test data. The analysis considers all the noise sources in the CR that contribute to the noise at the compliance location. It also addresses the acoustic absorption, structural damping, and the transmission loss characteristics of the CR shroud. Three particular configurations of the shroud were considered for the acoustic analysis. In the first configuration the shroud was built as a frame structure carrying 3-mm thick aluminum alloy panels. The second case analyzed 2-mm thick aluminum panels with a lining of 40-mm thick acoustic absorption material facing the inside of the CR enclosure. A double wall sandwich concept was considered in the final case consisting of two 1-mm thick aluminum panels with 30-mm thick acoustic absorption material in between. The latter configuration performed best acoustically but is structurally more complicated to build and has a total thickness of 32 mm and without acoustic absorption material on the inside to absorb sound inside the rotor enclosure.

The shroud design, however, is primarily driven by structural and weight considerations. At some point in the design process it was decided that a structural frame with honeycomb panels would save weight while providing the same structural integrity as aluminum alloy panels. It was deemed necessary to accomplish an initial assessment of the acoustic implications of the use of honeycomb material. More detailed analysis and verification was planned for later in the design cycle. While the transmission loss characteristics of aluminum are well documented in the literature, few analytical or test results are available for honeycomb panels. Since a panel made of honeycomb material has a higher stiffness-to-mass ratio than an aluminum panel the fundamental structural resonance will occur at a higher frequency. The bending stiffness B for honeycomb is approximated by a formula given in Reference 1, assuming the core has no flexural rigidity:

$$B = E/(1-\nu^2) * [(t_1^3+t_2^3)/12 + t_1t_2/(t_1+t_2) * (t_1/2+t_2/2+d)^2]$$

where E is the elasticity modulus, ν is Poisson's ratio, d is the core thickness, and t_1 and t_2 are the face plate thicknesses. The fundamental resonance of a simply supported rectangular panel with a surface mass ρ, and dimensions a and b is given by[1]

$$f_r = \pi/2\ (1/a^2+1/b^2)\ \sqrt{(B/\rho)}$$

The critical frequency is given by[1]:

$$f_c = c^2/(2\pi)\ \sqrt{(\rho/B)}$$

where c is the speed of sound. The coincidence frequencies for honeycomb material at all the angles of sound incidence will thus occur at lower frequencies than for aluminum material. The shifts in structural and coincidence frequencies are schematically indicated in Figure 9. The mass law transmission loss for the honeycomb panel is thus lower and occurs over a smaller frequency band. Other resonance frequencies are also important as a degrading in the transmission loss occurs, including dilatational and double wall resonances. Acoustic resonances inside the CR may couple with the CR structural resonances, which might further degrade the transmission loss characteristics of the shroud. Avoiding that highest source excitation frequencies coincide with acoustic or structural resonances might minimize structural-acoustic interaction. Below the panel fundamental resonance frequency, which is controlled by the bending stiffness of the panel, higher transmission loss may be obtained. However, higher transmission loss in this stiffness-controlled frequency region will not be achieved if the structure supporting the honeycomb panel is not as stiff as the honeycomb panel itself. This is illustrated in Figure 10, which shows the transmission loss of a honeycomb panel mounted on an aluminum surface (Reference 2). The inferior stiffness of the aluminum caused the transmission loss to deteriorate rapidly below 125 Hz. Analyses and related verification of these initial assessments should be performed early in the design process.

Sound attenuation measures being considered include visco-elastic damping tape on the honeycomb panels, double wall construction with different resonance frequencies of the individual panels, absorption material between the panels and inside the CR enclosure, and avoidance of resonance interaction. Finite element analysis results and test data from a structural evaluation of an engineering model are anticipated along with the results from acoustic verification tests. These analyses, test results, and other considerations will be at the basis of a new noise control plan and an updated analysis report to ensure compliance with acoustic interface requirements for pressurized payloads.

5. MICROGRAVITY SCIENCE GLOVEBOX

The Microgravity Science Glovebox (MSG) is a joint development project between NASA and the European Space Agency (ESA). Its configuration has been planned around the concept of an experimental workstation where a variety of experiments can be installed and operated in a fashion similar to operation in a ground-based laboratory. The facility provides a large enclosed work volume (WV), power, video, photography, vacuum connections, heat rejection, stowage, filtered air, gaseous nitrogen, lighting, airlock access, physical positioning and hold-down attachments, and computer data acquisition and control capabilities. The MSG Flight Unit rack, shown in Figure 11, serves as a clean working area enclosed and sealed by a large window, designed to contain potential liquid spillage, loose hardware, or gaseous by-products generated as a result of the experiment's performance. Crew access to the WV and operational manipulation of the experiments is through sealed glove ports. The sealed mode of operation and the air circulation provide two levels of containment. The air circulation is provided by the Air Handling Unit (AHU), which has three fans that draw air from the WV through the filter banks (Figure 12). The air is then blown through the heat exchanger, the process control valves and into the air discharge duct where it is redirected towards the WV. The Avionics Air Assembly (AAA) provides cooling for the rack. The AAA as well as the AHU was computer controlled at fan speed settings 1, 3, 4, 5, or 7, with 7 being the highest operating condition.

Initial measurements had indicated that acoustic noise levels of the MSG rack were well above the NC-40 criterion for continuous noise level limits of an integrated rack. This was verified by one-third octave band measurements taken with a Type 1 Sound Level Meter on the Engineering Unit of the MSG in a clean room environment. Although background noise levels were relatively high the data was considered of sufficient quality for diagnostic purposes. The one-third octave band measurements were taken at 0.6 meter from the front of the rack for various operating conditions of the noise sources. Modes included settings 4, 5, and 7 for the AAA fan; and mode 7 for each of the three air-handling unit fans by themselves and in combination. Table 2 shows the highest measured sound pressure levels at locations 0.6 m from front surface of the rack, which was for the condition where all three air-handling units were operating in mode 7. Also tabulated in Table 2 are the NC-40 values and the background noise SPL at the measurement location. Diagnostic narrow band data were also taken with a laptop based real-time analyzer with an omni-directional microphone. Data was linearly averaged with 32 samples over a frequency range from 0-5383 Hz with a 10.7 Hz bandwidth. A Hanning window was applied. Peaks in the frequency spectrum at 118.4/129.2 Hz, 269.2 Hz, 376.8 Hz, and 764.4 Hz were identified as related to the 123.3 Hz rotational frequency and harmonics of the AHU fans 7400 rpm rotational speed. A strong peak at 559.9 Hz existed, but was not identified. It could possibly have been an acoustic resonance in the glovebox cavity.

The one-third octave band sound level meter and narrow band real time analyzer were then used to locate sources of noise inside the rack with the back panels removed (Figure 13). Measurements, with only the AAA operating, were conducted at the inlet of the AAA piping, close to the top bellow, close to the bottom bellow an at the heat exchanger outlet. Measurements were also taken near the left, center and right AHU fans, each of them operating individually and all of them in unison. The AHU fans were unbolted from their mountings to examine structure borne noise. Sound pressure level measurements were also conducted inside the glovebox cavity. Finally, a list of noise sources was compiled which mainly included cavity resonances, and structure borne and air borne noise from the AHU and the AAA, and their heat exchangers. Preliminary results indicated that the AHU fans generated the most noise with distinct peaks at 123.3 Hz and 764.4 Hz, corresponding to the rotational speed and blade passage frequency respectively. However, it should be noted that the measured SPL levels for each of the identical fans were quite different. The noise of the AAA was much lower, especially when it was run at the normal operating speed of mode 4.

Several different noise control measures were applied to the MSG rack to lower the acoustic noise emission. Contoured one-inch thick acoustic foam absorption material was attached to the side, top and back skin panels, while the bottom was covered with five-inch thick pyramidal foam. This application prevents the build-up of acoustic energy inside the rack. Visco-elastic damping materials were suggested as remedy for resonant panel noise radiation. Gaps were closed with tape or silicone rubber materials. Acoustic foam mufflers were constructed and applied to the air handlers and the AAA air inlet duct. Mass-loaded barrier materials were used to wrap hoses and isolate noise sources. New sound pressure level measurements were conducted to verify the effectiveness of the noise control measurements and to record progress.

The AHU units, which were originally made of sheet metal, were replaced by custom-designed and fully milled units. The fin blades received aerodynamic contours and the number of blades were changed from 6/8/6 to 7/8/9. The inlet was smoothed out aerodynamically and size of internal fan axial gaps were minimized. The central rotor axle geometry was optimized to allow a smooth airflow in the narrow central section. The most noise reduction was achieved by different bend geometry of the air outlet housing after the second stage rotor. The new AHU fans were statically and dynamically balanced. The AAA fan configuration and geometries were not changed. The fans were retrofitted with new bearings of a different material to remedy an anomaly that had resulted in a ticking noise during acoustic testing.

The acoustic diagnostic measurements allowed timely implementation of noise control measures. Compliance measurements on the flight unit were conducted in the anechoic EMC (Electromagnetic Compatibility) facility at Astrium GmbH in Bremen, Germany. Based upon what was learned in these sessions. remedial action was taken to implement modifications discussed into the flight hardware. At verification testing for flight, performed at a later date, acoustic sound pressure levels at 0.6 meter from the MSG rack were measured less than the NC-40 requirements, except for a 2.1 dB exceedance in the 500 Hz octave band. The maximum continuous sound pressure levels for the MSG rack are compared with the allowable integrated rack sound pressure levels in Table 3. An exception request for this exceedance was submitted and subsequently approved.

6. CONCLUSIONS

Four case studies were presented, addressing different acoustic challenges. The common goal of these studies was to help ascertain compliance with the acoustic requirements for hardware on International Space Station. Although the approach and proposed solutions were different for each individual case it became apparent that the key to successfully resolve the acoustic issues is to tackle them as early in the design process as possible. This facilitates sufficient time to design, develop and verify noise abatement concepts to procure applicable hardware and materials, and precludes late cost and design impacts. Continuously updated Noise Control Plans and Analysis Reports have proven to be invaluable tools to successfully meet the ISS acoustic requirements.

7. ACKNOWLEDGEMENTS

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8. REFERENCES

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9. TABLES

Table 1. Muffler insertion loss data

	Muffler Insertion Loss [dB]							
	63	125	250	500	1000	2000	4000	8000
CGBA	11	10	4	11	18	14	21	25
CRIM	8	6	1	4	6	10	14	18
PGBA (Inlet)	13	9	4	6	12	17	21	28
PGBA (Outlet)	11	7	1	3	7	18	20	26
STES	7	7	2	4	7	12	18	24

Table 2. Diagnostic SPL measurements on the MSG Engineering Unit rack in the clean room

Octave Band Frequency [Hz]	NC-40 [dB]	Background Noise [dB]	Diagnostic Measurements [dB]
63	64	44.5	53.4
125	56	41.2	54.4
250	50	46.5	49.8
500	45	38.3	62.4
1000	41	38.0	56.3
2000	39	33.3	44.3
4000	38	29.6	39.2
8000	37	23.5	32.2
dBA	49	43.2	61.3

Table 3. Compliance SPL measurements on the MSG Flight Unit rack in the EMC room

Octave Band Frequency [Hz]	NC-40 [dB]	Compliance Measurements [dB]
63	64	45.7
125	56	50.3
250	50	42.7
500	45	47.1
1000	41	39.4
2000	39	36.0
4000	38	35.4
8000	37	30.2
dBA	49	46.9

10 FIGURES

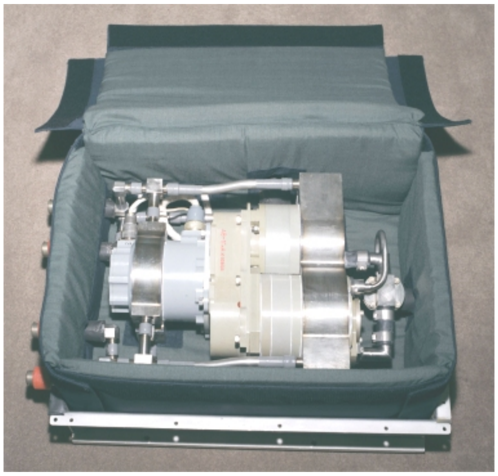

Figure 1: Pump in the acoustic enclosure

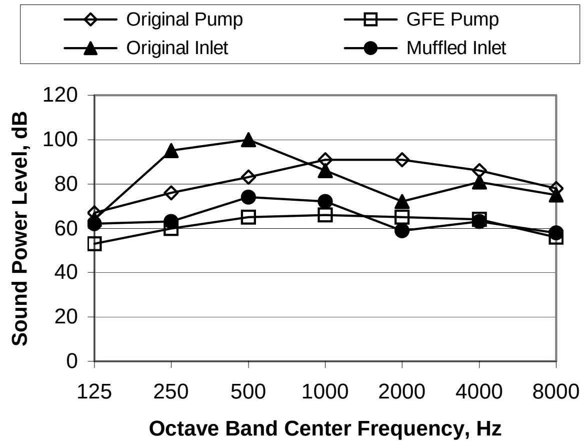

Figure 2: Sound power levels for the original and acoustically treated airlock pump and inlet

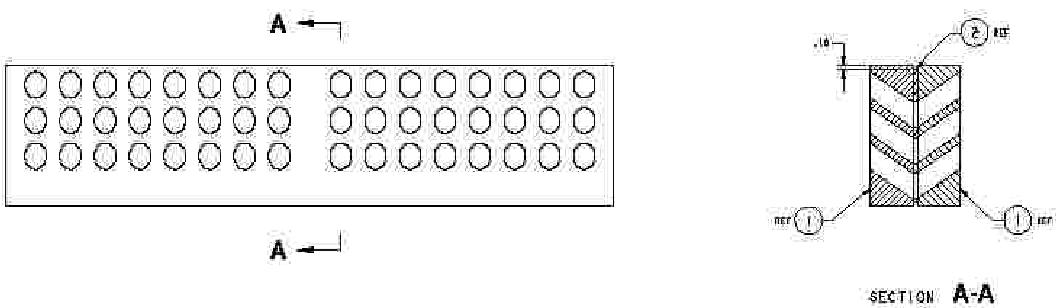

Figure 3: PGBA muffler hole pattern

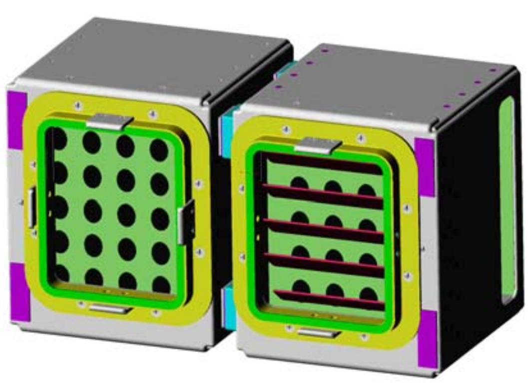

Figure 4: CGBA prototype mufflers

Figure 5: CGBA muffler attached to test fixture

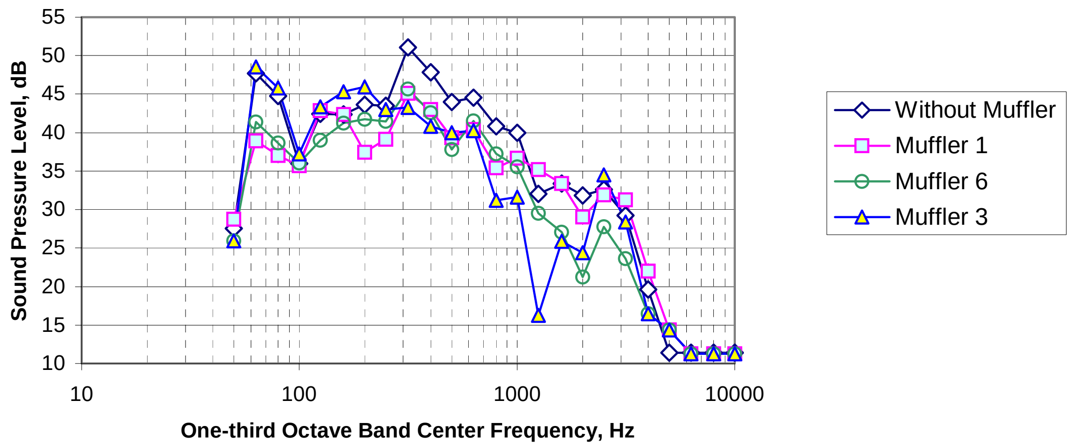

Figure 6. Prototype muffler results

Figure 7. CGBA prototype muffler

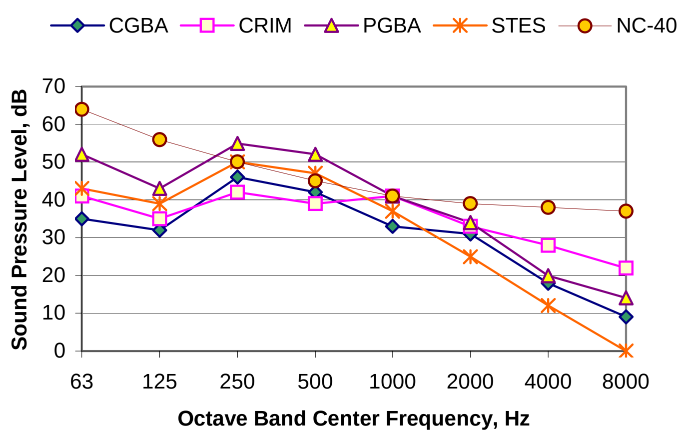

Figure 8. Payloads with mufflers plotted against NC-40 curve

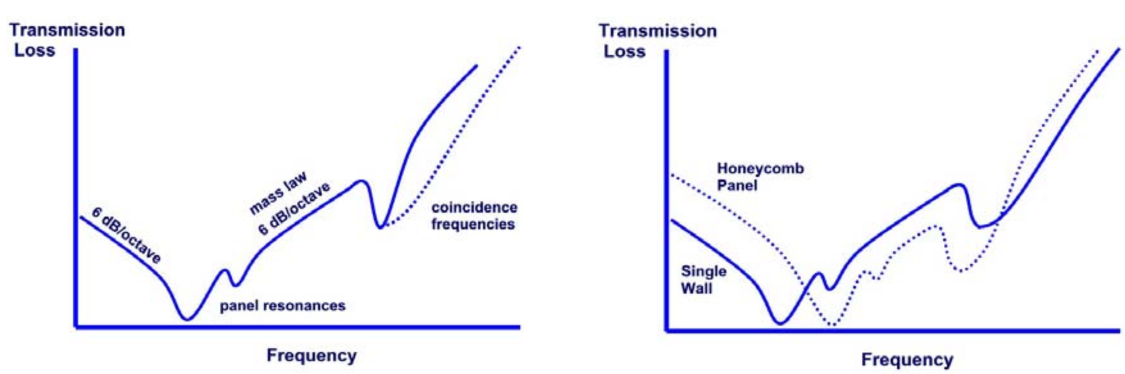

Figure 9: Transmission loss of a homogeneous single wall and a honeycomb panel showing shifts in resonance frequencies

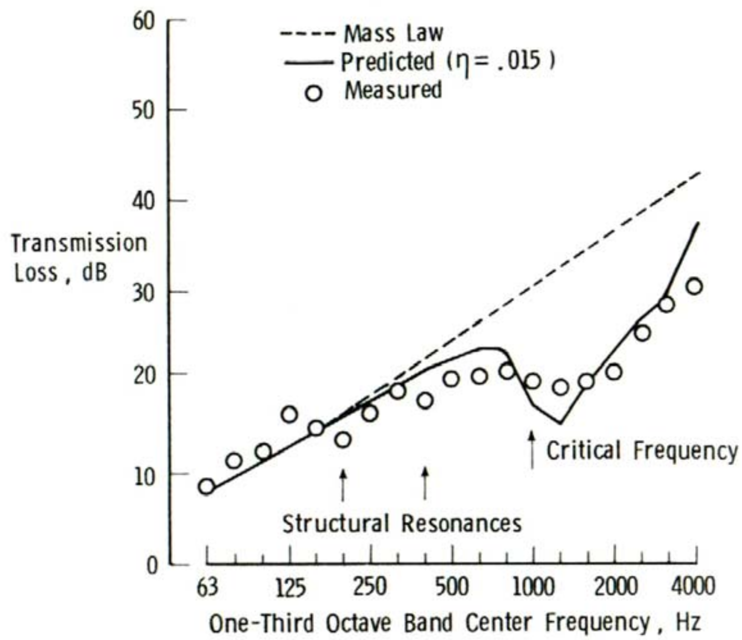

Figure 10: Transmission loss of a honeycomb panel

Figure 11. MSG Flight Unit rack

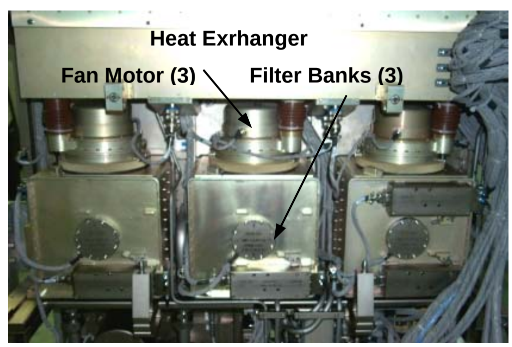

Figure 12. MSG Air Handling Unit

Figure 13. Rear view of the MSG – back panels removed