AEROSPACE INFORMATION REPORT

AIR1826™ REV. A

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Reaffirmed 2024-10

Superseding AIR1826

Acoustical Considerations for Aircraft Environmental Control System Design

RATIONALE

The report required revision to reflect current technology of aircraft ECS noise sources and reduction methods.

AIR1826A has been reaffirmed to comply with the SAE Five-Year Review policy.

FOREWORD

Noise levels in aircraft passenger cabins and flight stations have primarily been caused, in the past, by the engines and the external airflow. The introduction of high by-pass ratio engines of considerably lower noise level, plus improved treatments for aerodynamic noise have resulted in significantly quieter interiors. Against these lower background levels, however, the noise from the environmental control system (ECS) is likely to be more noticeable and may be the dominant contributor to the overall interior noise level. It is important, therefore, for the designer to appreciate the principles governing the generation of ECS noise and to understand the available means for controlling and reducing this noise.

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1. SCOPE

This Aerospace Information Report (AIR) is limited in scope to the general consideration of environmental control system noise and its effect on occupant comfort. Additional information on the control of environmental control system noise may be found in 2.3 and in the documents referenced throughout the text.

This document does not contain sufficient direction and detail to accomplish effective and complete acoustic designs.

1.1 Purpose

The purpose of this AIR is to provide aid for the reduction of environmental control system noise levels and to minimize their effect on passengers and crew members through engineering design. The reader should be aware that the material included in this document is for general guidance purposes only.

2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this document to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of other publications shall be the issue in effect on the date of the purchase order. In the event of conflict between the text of this document and references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

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2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org

AIR1168/3 Aerothermodynamic Systems Engineering and Design

ARP1307 Measurement of Exterior Noise Produced by Aircraft Auxiliary Power Units (APUs) and Associated Aircraft Systems During Ground Operation

ARP1323 Type Measurements of Airplane Interior Sound Pressure Levels During Cruise

ARP4245 Quantities for Description of the Acoustical Environment of the Interior of Aircraft

ARP4721/1 Monitoring Aircraft Noise and Operations in the Vicinity of Airports: System Description, Acquisition, and Operation

2.1.2 ANSI Publications

Copies of these documents are available online at http://webstore.ansi.org.

ANSI S1.6-1984 (R 2006) Preferred Frequencies, Frequency Levels, and Band Numbers for Acoustical Measurements

2.1.3 ASTM Publications

Available from ASTM International, 100 Barr Harbor Drive, P.O. Box C700, West Conshohocken, PA 19428-2959, Tel: 610-832-9585, www.astm.org.

ASTM C522-03 Standard Test Method for Airflow Resistance of Acoustical Materials

ASTM E90-09 Standard Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions and Elements

2.1.4 Code of Federal Regulations (CFR)

Available from the United States Government Printing Office, 732 North Capitol Street, NW, Washington, DC 20401, Tel: 202-512-1800, www.gpo.gov.

14 CFR Part 25 Airworthiness Standards: Transport Category Airplanes

2.1.5 European Aviation Regulations

Available from European Aviation Safety Agency, Postfach 10 12 53, D-50452 Cologne, Germany, Tel: +49-221-8999-000, www.easa.eu.int.

CS-25 Certification Specifications for Large Aeroplanes

2.1.6 U.S. Government Publications

Copies of these documents are available online at http://quicksearch.dla.mil.

MIL-STD-740-2 Structureborne Vibratory Acceleration Measurements and Acceptance Criteria of Shipboard Equipment

MIL-STD-1472 Human Engineering Design Criteria for Military Systems, Equipment and Facilities

MIL-STD-1474 Noise Limits

2.1.7 Applicable References

J. S. Lamancusa, Engineering Noise Control, Pennsylvania State University, 2000

2.2 Related Publications

This list of publications is provided for informational purposes only. These publications are not a required part of this SAE Aerospace Technical Report.

L. L. Beranek, Noise and Vibration Control, Institute of Noise Control Engineering, Arlington, NY, 1989.

C. M. Harris, Handbook of Noise Control, McGraw-Hill, New York, 1979.

C. H. Hansen, Understanding Active Noise Cancellation, Taylor & Francis, 2001

D. Middleton, The Noise of Ejectors, Aeronautical Research Council Reports, UK Ministry of Aviation, 1965

D. M. Howard, J. A. S. Angus, Acoustics and Psychoacoustics, Elsevier Ltd, 2009

ANSI S1.1-1994 (R2004) Acoustical Terminology

ANSI S1.2-1962 (R 2001) Method For The Physical Measurement Of Sound

ASHRAE 2007 HANDBOOK – HVAC Applications, Chapter 47 Sound and Vibration Control

3. FUNDAMENTALS OF ACOUSTICS

3.1 Sound

Pressure alterations or particle displacements propagated in an elastic medium produce sound. In air, sound consists of propagated changes in pressure that alternate above and below the ambient pressure. These changes occur when vibrating objects accelerate the air particles next to them.

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The speed of sound in a particular medium is defined as the product of frequency and wavelength:

$$c = f\,\lambda \qquad \text{(Eq. 1)}$$

where: c is the speed of sound

f is the frequency

λ is the wavelength.

The speed of sound in air, which varies with temperature, is given by the formula:

$$c = 49.03\ R^{1/2}\ \text{ft/s}\ (c = 20.05\ T^{1/2}\ \text{m/s}) \qquad \text{(Eq. 2)}$$

where T is the temperature expressed in Kelvin, and R the temperature expressed in Rankine.

At 68 °F (20 °C) c = 1125 ft/s (343 m/s). The wavelength of sound in air as a function of frequency at 68 °F can be read from Figure 1.

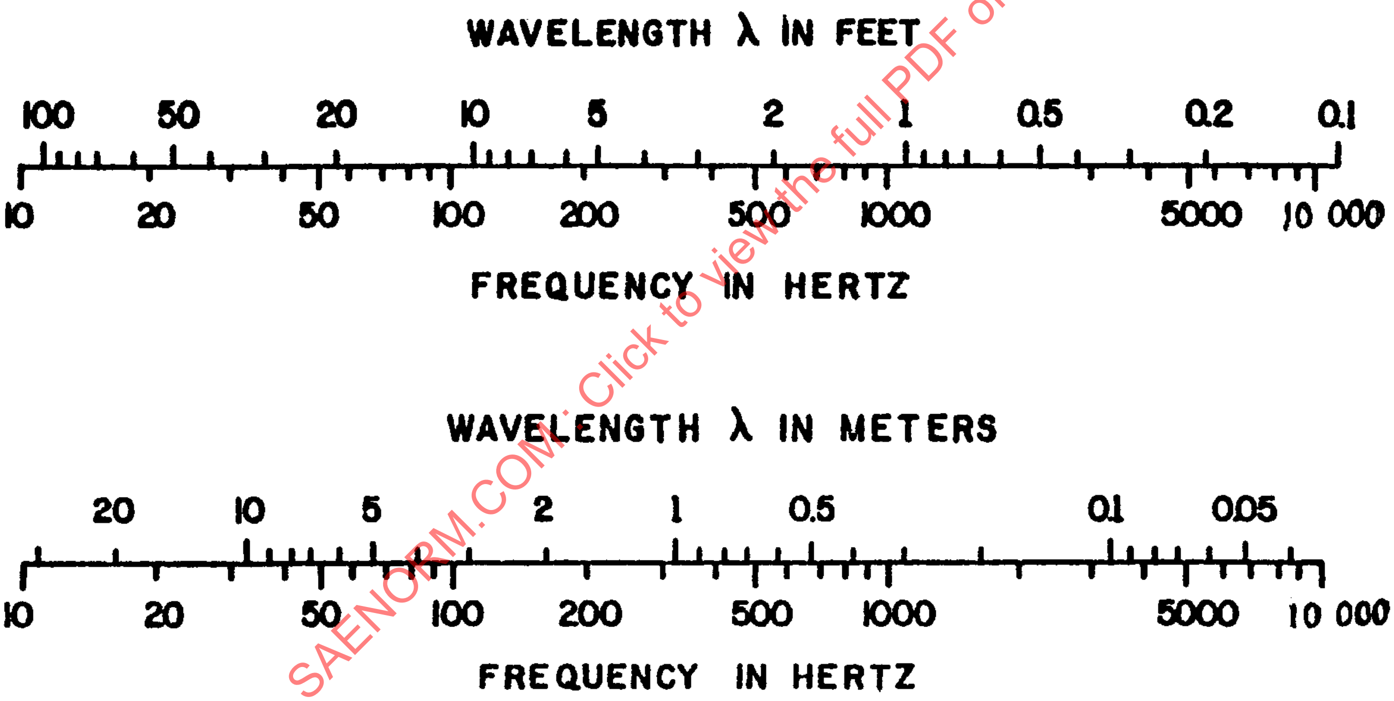

Figure 1 - Wavelength in air versus frequency at 68 °F (20 °C)

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3.2 Noise

Noise is any undesired sound (If ambiguity exists as to the nature of the noise, a phrase such as "acoustic noise" or "electric noise" should be used).

3.3 Frequency Spectrum

The quality of sound is determined primarily by its frequency spectrum. A plot of sound pressure level or sound power level against frequency is called a frequency spectrum.

A "pure tone" is a sound whose level is concentrated in a single frequency (i.e., the frequency spectrum is a single point plot and the sound waveform is a pure sinusoidal curve). As an extension, a "tone" or "tonal noise" is one in which the sound levels spread around a narrow frequency band, while in a "broadband noise" the sound levels are evenly distributed on a large frequency range. "Harmonics" are tonal noises whose characterizing frequency is an integer multiple of a lower frequency known as "fundamental frequency".

The principle of superposition is applicable to waveforms, thus a generic sound can be decomposed in a number of linearly combined waveforms, and will show one or more of the features discussed above (see an example in Figure 2).

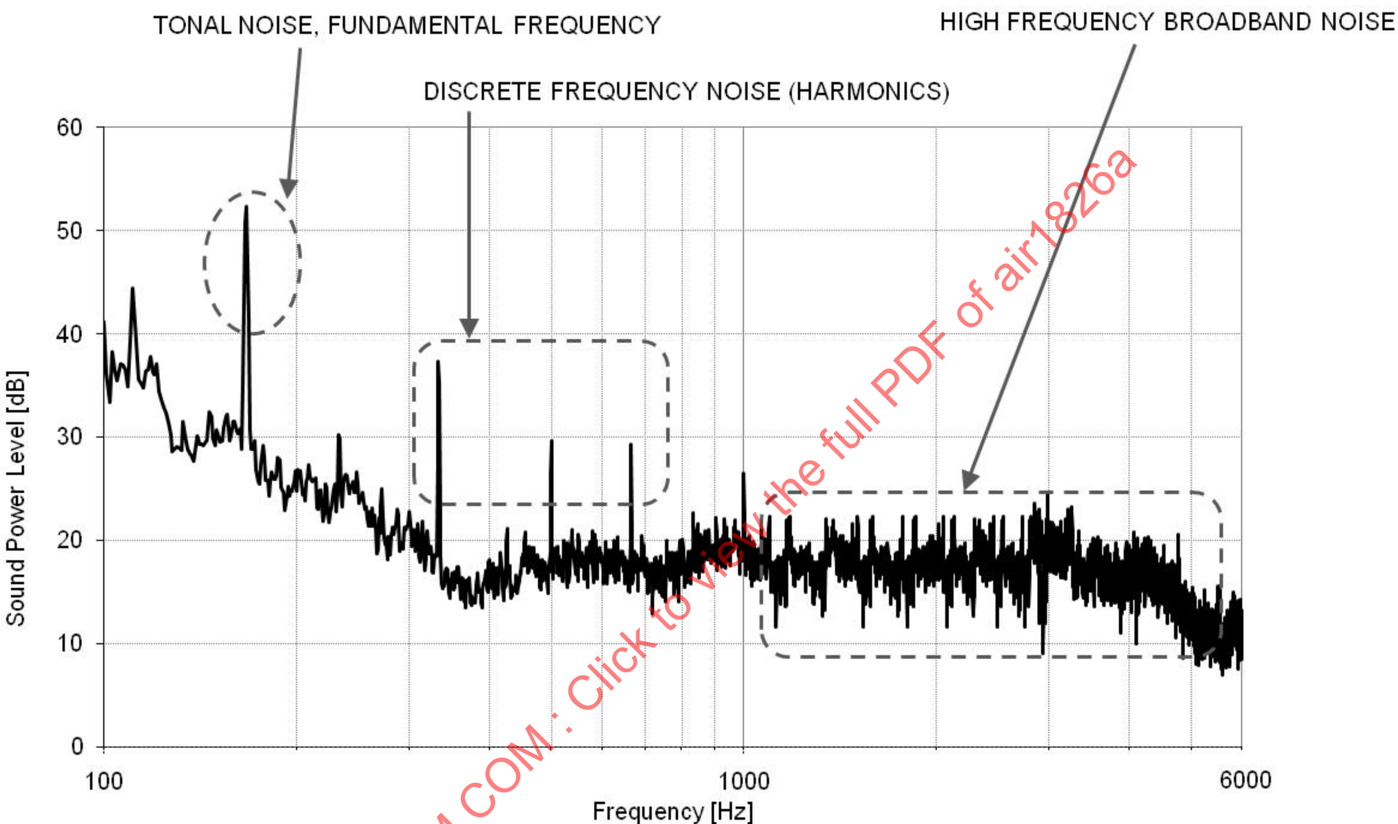

Figure 2 - Example sound spectrum

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3.4 Octave Bands

Part of the audible frequency range, approximately 20 to 10000 Hz is divided into bands, usually one octave wide. The term octave means that the frequency of the upper limit of the pass band is twice that of the lower limit. An octave band sound pressure spectrum is obtained with an octave band analyzer containing electronic filters that pass only those components of the measured sound that have frequencies within the limits of the filter. The first set of octave band limits shown in Table 1 is specified by ANSI S1.6-1984. The second set has been in general use for some time and is given for comparison. Data from the second set can be converted to the first set by graphical interpolation with sufficient accuracy.

Table 1 - Octave pass bands

Frequency Bands	Frequency, Hertz							
Preferred								
Mid-frequency	63	125	250	500	1000	2000	4000	8000
Approximate frequency limits								
Lower	45	90	180	355	710	1400	2800	5600
Upper	90	180	355	710	1400	2800	5600	11200
Previously Used								
Approximate geometric mid-frequency	53	106	212	425	850	1700	3400	6900
Frequency limits								
Lower	37.5	75	150	300	600	1200	2400	4800
Upper	75	150	300	600	1200	2400	4800	9600

3.5 Levels and Decibels

The magnitude of sound is expressed as levels. The level is a measure of the ratio of sound power, or sound pressure to a reference value, and is expressed in decibels. The relationship between sound power and sound pressure corresponds to that between heat and temperature. The ear and instruments respond to sound pressure, but equipment sound ratings are best expressed in terms of sound power output. Sound power cannot be measured directly; it has to be computed from sound pressure measurements. Acoustical energy is proportional to the square of sound pressure.

Sound pressure level Lp (or SPL) in decibels, referenced to the approximate threshold of hearing, i.e., 2×10^{-5} Pascal (N/m^2):

$$L_p = 10\log_{10}\left(\frac{p}{2 \times 10^{-5}}\right)^2 = 20\log_{10}\left(\frac{p}{2 \times 10^{-5}}\right) \qquad \text{(Eq. 1)}$$

where: p = sound pressure in Pa

Sound power level L_w (or PWL) in decibels, referenced to 10^{-12} watt, is defined as:

$$L_W = 10\log_{10}\left(\frac{W}{10^{-12}}\right) \qquad \text{(Eq. 4)}$$

where: W = sound power in watt

Whenever the word level is used as a modifier for sound power or pressure, a reference power or pressure has to be given and the ratio between the measured quantity and the reference quantity is used to calculate the decibel level. The reference for sound power level must be noted since 10^{-13} watt is still often used. If the 10^{-13} watt reference is used, simply subtract 10 dB from the power level to obtain levels referenced to 10^{-12} watt.

Figure 3 is a graphical representation of the pressure and power level formulas. When the pressure or power level changes, its ratio to the referenced value changes accordingly. For a 3 dB increase, the power ratio is doubled while a 6 dB increase in pressure level doubles the pressure ratio. About the smallest change the ear can detect is 1 or 2 dB. While stationary, 3 to 5 dB changes are detected, however, they are not detected walking from one area to another. A 10 dB noise increase would be perceived as twice as loud.

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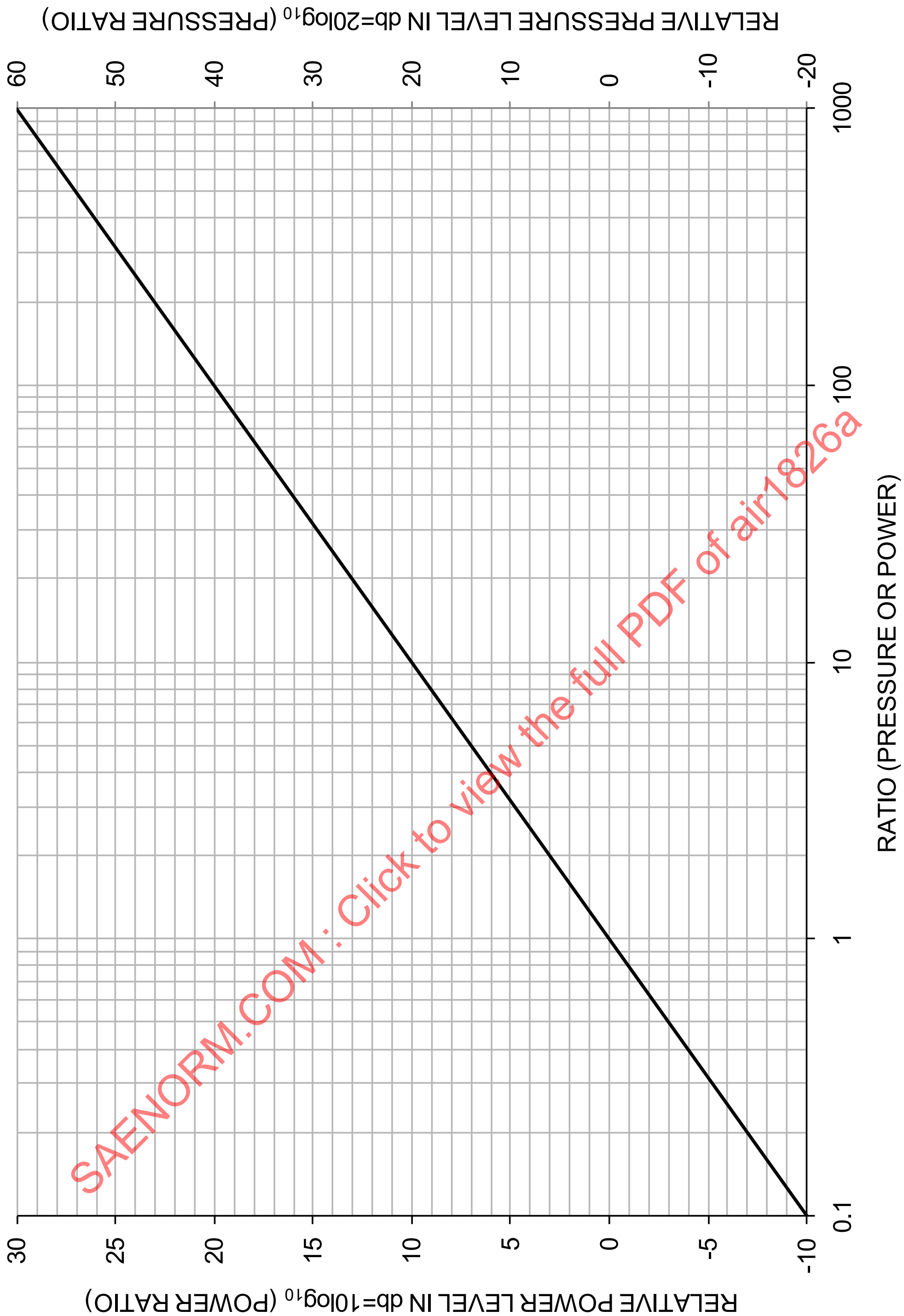

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Figure 3 - Sound pressure and pressure level ratios

3.6 Decibel Calculations

The ratio of the sound power entering an element of a system to the sound power continuing beyond the element, expressed in decibels, is called the attenuation of that element:

$$\text{Attenuation (in db)} = 10\log_{10}\left(\frac{W_e}{W_c}\right) \qquad \text{(Eq. 5)}$$

where: W_e = Sound power entering, watt

W_c = Sound power continuing, watt

Figure 3 may be used to convert the ratio of sound power to decibels and vice versa. If an attenuator absorbs 99.9% of the entering sound power, the power ratio is 100/0.1 = 1000, and the attenuation is 30 dB. Since this is relative, no reference is necessary. If referenced decibel levels are used, the attenuator equation reduces to a simple arithmetic expression:

$$\text{Attenuation (in db)} = L_{We} - L_{Wc} \qquad \text{(Eq. 6)}$$

where: L_{We} = sound power level entering (dB)

L_{Wc} = sound power level continuing (dB)

The attenuation of a group of elements through which the sound passes in succession is the arithmetic sum of all individual attenuation values in decibels.

When the decibel system is used, the levels of two or more sounds cannot be added or subtracted arithmetically. The following equation can be used to combine levels:

$$L = 10\log_{10}\left(\sum_i 10^{\left(\frac{L_i}{10}\right)}\right) \qquad \text{(Eq. 7)}$$

where:

L_i = i^{th} level to be combined

A graphical method can also be used to combine two sound levels at a time. If more than two levels are to be combined, the two highest levels should be combined, and then the resulting value combined with the next highest level.

Rearranging terms from Equation 7 for the case of only two levels, the following formula can be derived:

$$Y = 10\log_{10}\left(1 + 10^{-\left(\frac{X}{10}\right)}\right) \qquad \text{(Eq. 8)}$$

where:

X (in dB) = arithmetic difference between the two sound levels to be combined (larger minus smaller).

Y (in dB) = amount to be arithmetically added to the larger sound level to be combined in order to obtain the sum of the two levels.

Equation 8 can then be plotted to get Figure 4 which correlates Y to X.

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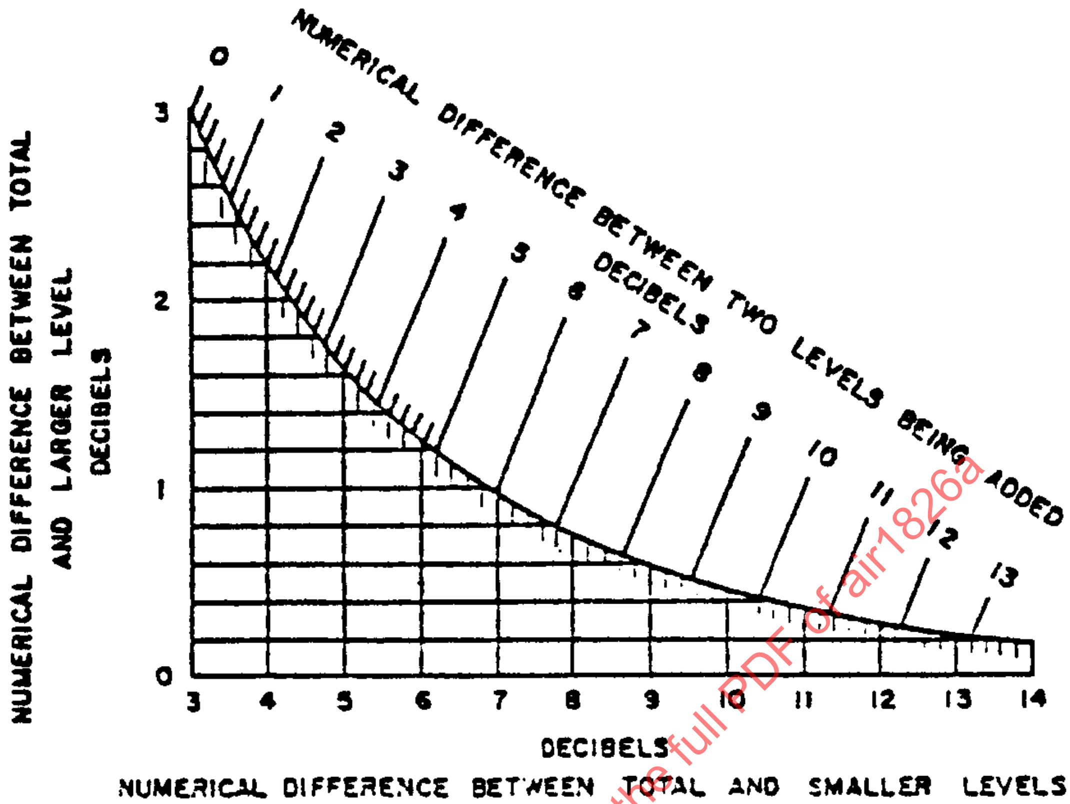

Figure 4 - Chart for combining levels

Example calculations:

Given L_{W1} = 86 dB and L_{W2} = 80 dB, their arithmetical difference is X = 6 dB. From either Equation 8 or Figure 4 it follows that Y = 0.97, so to obtain the sum of the two sound levels (L_{W1} amplified by L_{W2}) we add Y to L_{W1}, or $L_W = L_{W1} + Y$ = 86.97 dB.

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3.7 Comparative Levels of Common Sounds

Table 2 - Comparative levels of common sounds

Sound Pressure Level (dB)	Sound Pressure (Pa)	Sound Pressure (psi)	Common Sounds and Qualitative Perception
160	2×10^{3}	3×10^{-1}	Medium Jet Engine
140	2×10^{2}	3×10^{-2}	Large Propeller Aircraft-Air Raid Siren-Riveting
130	64	9.6×10^{-3}	*[Pain threshold]*
120	20	3×10^{-3}	Discotheque
100	2	3×10^{-4}	*[Very noisy]* Heavy City Traffic Subway
80	2×10^{-1}	3×10^{-5}	Busy Office
70	6.4×10^{-2}	9.6×10^{-6}	*[Noisy]*
60	2×10^{-2}	3×10^{-6}	Normal Speech at 1 m
40	2×10^{-3}	3×10^{-7}	*[Quiet]* Quiet Residential Neighborhood
20	2×10^{-4}	3×10^{-8}	Whisper
0	2×10^{-5}	3×10^{-9}	*[Barely Audible]* Threshold of Hearing for a child

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For the definition of the Sound Pressure Level (L_p) see Equation 3.

4. DESIGN APPROACH

To reduce the ECS noise in the aircraft interior, two approaches are possible, source reductions and transmission path attenuations. In order to reduce noise at the source, it is necessary to identify the components or parts of the system generating the noise and to determine the mechanism of the sound generation process. Modifications may then be designed as appropriate. When all available steps have been taken to reduce noise at the source, any further improvement must come from the addition of insulation, absorption, or other treatments into the path between source and cabin. This requires that the dominant sound transmission path be determined, i.e., either structureborne or airborne.

Excessive noise can be generated by active components such as air cycle machines and by the flow of air through the system. Air noise is generated at valves, in the ducts at positions of rapid area or shape change, and at exits from the ducts. Noise radiated from the surface of ducts may be due to unsteadiness in the internal air flow. Vibration of ducts and/or equipment may be transmitted via the airframe structure and radiated as noise into the cabin.

Information on noise suppression methods is given in Section 5, while Section 6 provides information on their application.

Although the major problems arising from ECS noise concern the passenger cabin and flight station, efforts should also be made to protect ground crews working around the aircraft.

Noise generated by the aircraft while parked on the ground, typically during routine operations between two flights, is referred to as "ramp noise". Unlike noise generated during flight, ramp noise is not regulated by Airworthiness Authorities: the authorities of each airport set their own limitations and/or penalties for ramp noise levels based on a number of factors that may include the proximity of nearby housing or standards agreed upon with a labor union. The more stringent limitations require that all gas turbine auxiliary power units (APU), and consequently environmental control systems (ECS) be shut down 5 minutes after arrival at the gate and started no more than 5 minutes before the estimated time of departure.

Air exhausting from air cycle machine fans or other overboard bleed flows can generate unacceptably high noise levels. Potential noise problems can be minimized in the design stage of equipment (such as fans) by adopting minimum noise generation principles.

Further improvement can come from the treatment of exhaust ducting and by reducing the exhaust velocity of emergent jets to as low a value as possible. A general design rule to be borne in mind is that high frequency sound is more easily attenuated and absorbed than low frequency sound (see Figure 5 for typical four-engine jet interior noise level).

4.1 Design Verification

Verification methods for the ECS interior noise and for ramp noise are largely based on full scale measurements on the final aircraft configuration.

The following SAE documents have been specifically published to support the designer in the verification effort:

ARP1323 "Type Measurements of Airplane Interior Sound Pressure Levels During Cruise" provides guidance for the definition of measurement procedures, for the selection of instrumentation and for data reporting when showing compliance with aircraft interior noise level goals or contractual requirements.

ARP4245 "Quantities for Description of the Acoustical Environment of the Interior of Aircraft" defines quantities that may be used to describe various attributes of the sound field in the interior of aircraft.

ARP1307 "Measurement of Exterior Noise Produced by Aircraft Auxiliary Power Units (APUs) and Associated Equipment During Ground Operation" provides guidance for the definition of measurement procedures, for selection of instrumentation and for data reporting when showing compliance with airport regulations for ramp noise. The document covers not only APU, but also associated equipment like ECS.

ARP4721 "Monitoring Aircraft Noise and Operations in the Vicinity of Airports: System Description, Acquisition, and Operation" provides engineering methods that can be applied to monitor aircraft noise in the vicinity of airports, as well as validation methods for permanent monitoring installations.

5. NOISE SUPPRESSION METHODS

5.1 Location and Orientation

The overall acoustic design for occupied areas in a passenger aircraft should be such that the noise due to the ECS does not contribute to the ambient noise level, i.e., it is not greater than a level about 10 dB below the combined engine and aerodynamic noise in each octave band. If the ECS equipment noise contains discrete frequencies or tones, the restriction becomes more severe and is a special problem. The ECS and its components can produce noise in two ways: (1) direct radiation of airborne sound waves into a space and (2) transmission of vibration or structureborne noise through the mechanical mountings to large radiating surfaces.

Noisy ECS equipment should, whenever practical, be mounted in areas as far away acoustically from occupied compartments as possible, such that the structureborne noise has to travel through long and tortuous mechanical paths, and airborne noise is subject to the transmission loss of one or more walls and partitions.

Noise should be reduced at the source, whenever possible. ECS equipment such as fans, blowers, valves, solenoids and regulators should have maximum permissible noise levels included as a part of the procurement specifications.

The noise sources should be oriented so that maximum sound radiation patterns are directed away from points of possible complaint. Maximum sound radiation usually occurs in the direction that air enters and leaves the duct.

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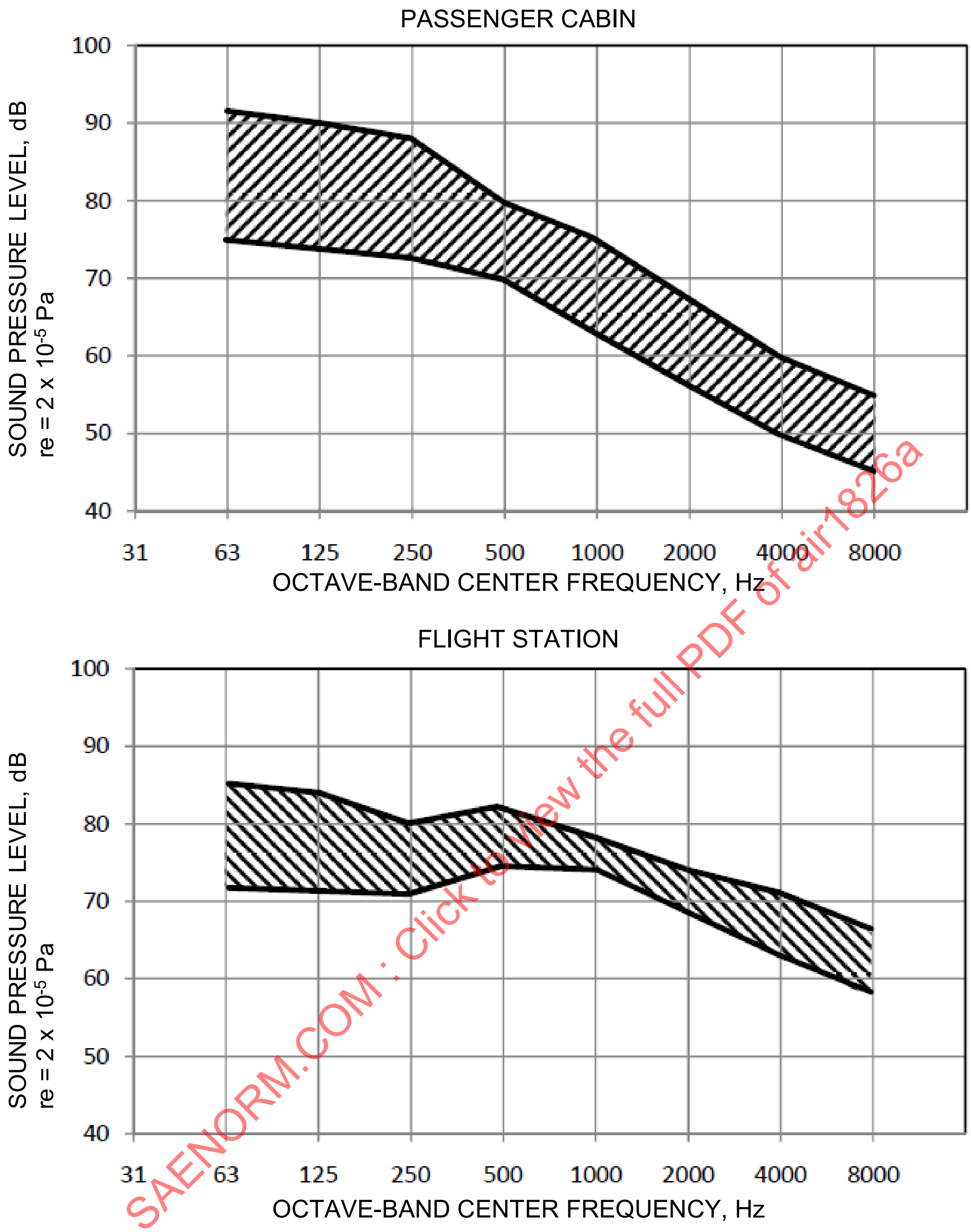

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Figure 5 - Typical four-engine commercial jet transport measured noise levels

NOTE: Existing conditions include:

(1) low bypass ratio turbofan engines, (2) normal long range cruise conditions, (3) measured at head height at aisle and window seats, (4) cabin air conditioning in normal operation, (5) individual passenger “eyeball” ventilators closed, and (6) measured at head positions of pilot and copilot.

5.2 Absorptive Materials

The process of sound absorption involves the conversion of acoustical energy into heat as friction is encountered. This attenuation results in a reduction of reverberation.

The materials most successfully used are either porous or geometric configurations with absorptive characteristics that can be predicted. In the first case, such materials are normally indicated as "(solid) foams" or "cellular engineered materials" and their porosity can be achieved by open, closed or semi-closed cells.

Election of absorptive materials should include a check to ascertain if the candidate material is considered a Material of Concern (MoC). Some regulatory agencies have placed restrictions or have banned the use of certain MoC, and special approvals may be required. Seek guidance from the aircraft manufacturer on the use of MoC.

A comparison between the two classes of materials is illustrated in Table 3.

5.2.1 Solid Foams

Open cells material (also known as "reticulated foams") are characterized by a structure in which the pores are all interconnected, forming a network that is filled by any fluid in which the material is eventually immersed into. For this reason, when exposed to air, such materials are relatively good insulators, but will absorb moisture and airborne contaminants. Also, their inherent structure renders them softer and lighter than the other type of foams. The most common open cell materials are acoustic foam, silicon foam, vinyl foam, and polyurethane foam.

Closed cell materials do not have a network of interconnected cells, and are heavier and more resistant than other foams, showing a higher compressive strength. Resistance to external contaminants and insulation properties can be implemented by filling the foam with gas during the manufacturing process. Even without these additional features, such processes are generally more complex and require more raw materials, which makes them more expensive and increases the manufacturing time when compared to other types of foam. Widely used closed cell material include PVC foam, Polyethylene Foam, Neoprene (polychloroprene foam).

Semi-open cell materials (like some rubber foam) represent a hybrid solution between open and closed cell foams: their behavior is similar to open cells when in normal conditions, but will assume the properties of closed cells foams when subject to compression loads. This generally makes them more appropriate for sealing and similar applications.

5.2.2 Fibers

Layers of fibers can be oriented and layered into specific geometric configuration in order to develop engineered materials showing the required acoustic absorption properties.

Common type of fibers include fiberglass (most used) and rockwool.

5.2.3 Selection Criteria

Absorptive materials can be combined to make different, multilayered packaging concepts for cabin thermo-acoustic insulation.

Consideration should be given to the following prior to selection:

a. Fire protection and Non-toxicity: flame resistance requirements in FAR/CS 23.853(a) and flame propagation requirements in FAR/CS 25.856(a) are applicable for thermo-acoustic insulation used onboard. For aircraft with more than 20 passengers, the additional requirements in FAR/CS 25.856(b) shall also be met, preventing flame penetration from the fuselage belly into the occupied compartments.

b. Moisture absorbing materials should be avoided in most applications.

c. The material should withstand the design temperature range without degradation.

d. The material should withstand those fluids that could contact the surface (as a result of leakage or spills) without disintegration (such as hydraulic oil, coolant, water, or engine oil or vapor).

e. The material should withstand environmental conditions such as mechanical vibration, sunlight, and a specified aging period without disintegration.

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f. The acoustic absorption coefficient should be known. This is the ratio of sound energy absorbed by the surface to the sound energy incident upon the surface. Theoretical total absorption equals 1; conversely zero absorption equals 0. This coefficient is usually established by reduction of laboratory test data (standing wave tube apparatus, reverberation room, etc.).

g. The specific flow resistance (R) should be established (see Equation 9).

$$R = \frac{\Delta P}{v} = \frac{\text{pressure differental across material (Pa)}}{\textit{perpendicular particle velocity}(m/s)} \quad \text{(Eq.9)}$$

$$\mathrm{R} = \left(\frac{\mathrm{Pa}}{\mathrm{m/s}}\right) = \left(\frac{\mathrm{N\,s}}{\mathrm{m}^3}\right) = (\mathrm{rayl}) \text{ (see 5.4.1)} \quad \text{(Eq.10)}$$

where rayl is a common unit for flow resistance, in units of N s/m^3

NOTE: The apparatus and procedure recommended for determining R is given by ASTM C522-03.

The value of R depends on pore size in foams, the finer pores having higher R values.

The value of R can vary from 1 to 10000. In most noise control applications, when the absorber is backed by a relatively rigid surface, the problem is to keep the value of R low enough to reduce surface reflection, but still high enough to dissipate the pressure wave as it travels into the absorber and is reflected back at the rigid surface. Optimum values are dependent on the thickness and impedance capability of the absorbing material. Figure 6 shows a sample relationship between the acoustic absorption coefficient and specific flow resistance (see 5.4 for additional information).

Table 3 - Sound absorptive material property comparison

Property	Foams	Fibers
Flammability	Can be made self-extinguishing	Good fire resistance
Acoustic	Excellent absorption in mid-to-high frequencies	Excellent absorption in mid-to-high frequencies
Environmental Considerations	Nontoxic, vibration resistant, deteriorates at high temperatures, absorbs moisture	Fiber contamination can be hazardous: poor vibration resistance: good high temperature properties, absorbs moisture
Acoustical Degradation	Little susceptibility if faced and edge sealed	Little susceptibility if faced and edge sealed
Applications	Equipment enclosures duct/plenum treatments	Equipment enclosures high temperature applications

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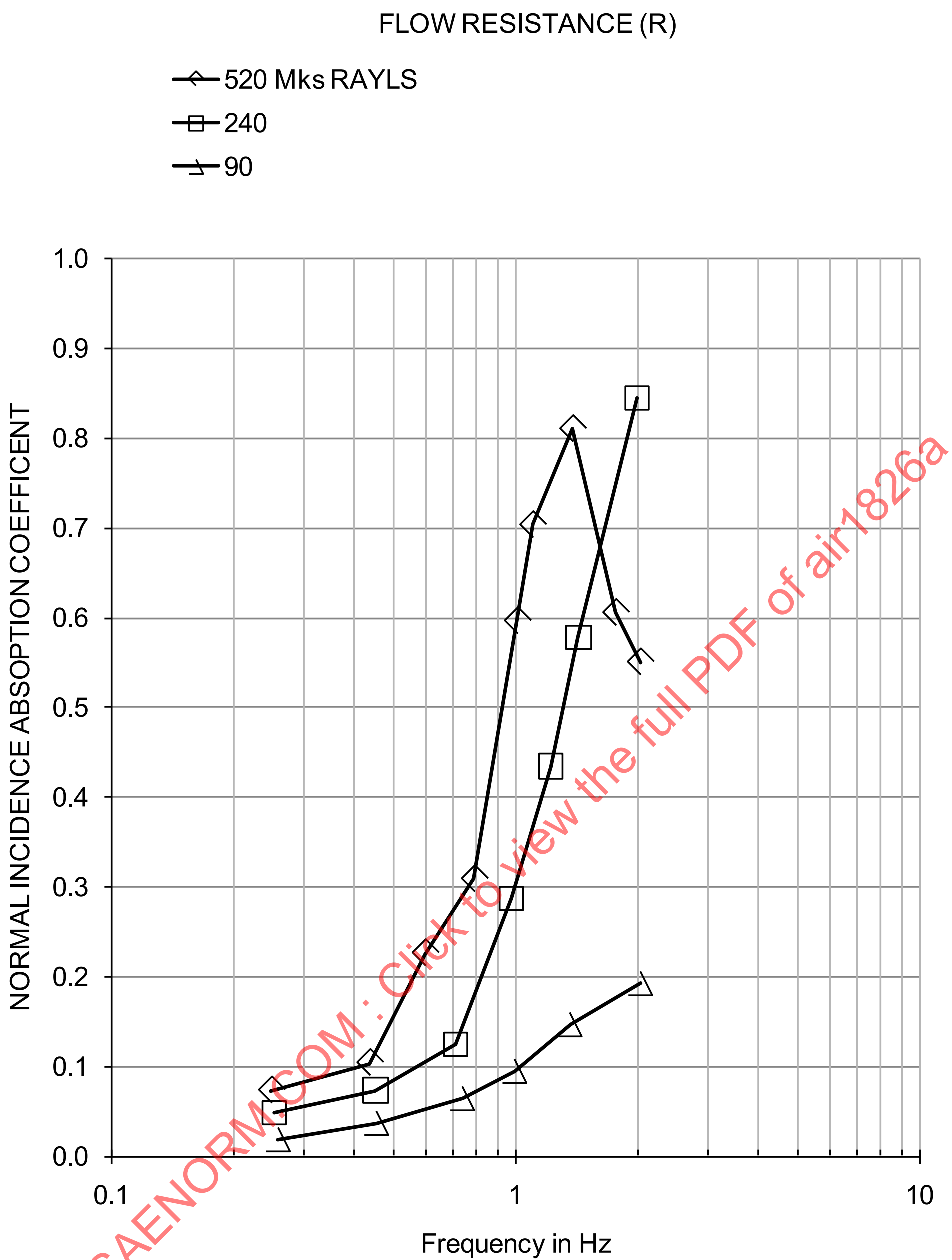

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Figure 6 - Unfaced urethane foam 0.5 inch (0.0127 m) thick density 3.8 lb/ft^{3}m^3 (60.7 kg/)

5.3 Transmission Loss

When sound waves impinge on a partition, the varying sound pressures acting on the surface set it in motion. A portion of the power carried by the sound waves is thus transferred to the partition. If the partition is a homogeneous panel, both sides vibrate in unison, and the same vibrations appear on the other side and radiate sound waves into the adjacent space. In more complex partitions, for example, one with double walls, part of the power carried by the sound waves may be attenuated within the partition itself. On the other hand, if the partition is porous or has holes or cracks through it, the sound waves may travel through the partition by way of the air channels.

The basic physical measure of the sound insulation value of the partition is the ratio of the sound power incident on one side of the partition to the sound power transmitted to the space on the other side. Expressed on a logarithmic scale, this ratio is known as the sound transmission loss (TL) expressed in decibels.

TL is a fixed property of a partition but varies with the frequency of the sound. The TL of all materials and partitions increases with frequency but at some point in the curve, a plateau or dip will occur with most materials where the partition develops traveling waves in sympathy with disturbing sound waves. When this coincidence occurs, the amplitude of vibration in the partition increases and more sound is transmitted.

Stiff partitions exhibit the worst dips in transmission loss while limp materials are the best. Sheet lead, leaded vinyl, or some other heavy material such as barium or silicone rubber are suitable for limp partitions. Materials containing lead are not considered acceptable for inhabited areas.

Figure 7 presents TL curves for limp partitions at various angles for the incident sound power. The ASTM standard for the measurement of TL requires random incidence (see ASTM E90-09).

Where noise must be blocked by a partition or contained in an enclosure, there are three cardinal requirements to be met to get the best performance:

a. The more massive the partition is, the higher the TL it is capable of. Double wall configurations can provide significant TL increases relative to a single wall with the same surface density.

b. The less stiffness the partition possesses, the better the performance that may be expected of it.

c. To achieve good TL, the partition must be free of leaks - either air leaks or structural features that constitute flanking sound paths.

5.4 Mufflers

A muffler may be described as any section of a duct that has been treated with the intention of reducing the transmission of noise, while at the same time allowing the free flow of air. A properly designed muffler should satisfy the following criteria:

Acoustical: Specifies the minimum noise reduction required from the muffler as a function of frequency

Aerodynamic: Specifies the maximum acceptable pressure drop at a given temperature

Geometrical: Specifies the maximum allowable envelope and restrictions on shape

Mechanical: May specify the materials from which it is constructed; is especially important in cases involving high temperature, velocity, and humid airflows.

In general, there are two types of mufflers: absorptive and reactive. In aircraft ECS the former is used, as reactive mufflers do not meet the aerodynamic criterion.

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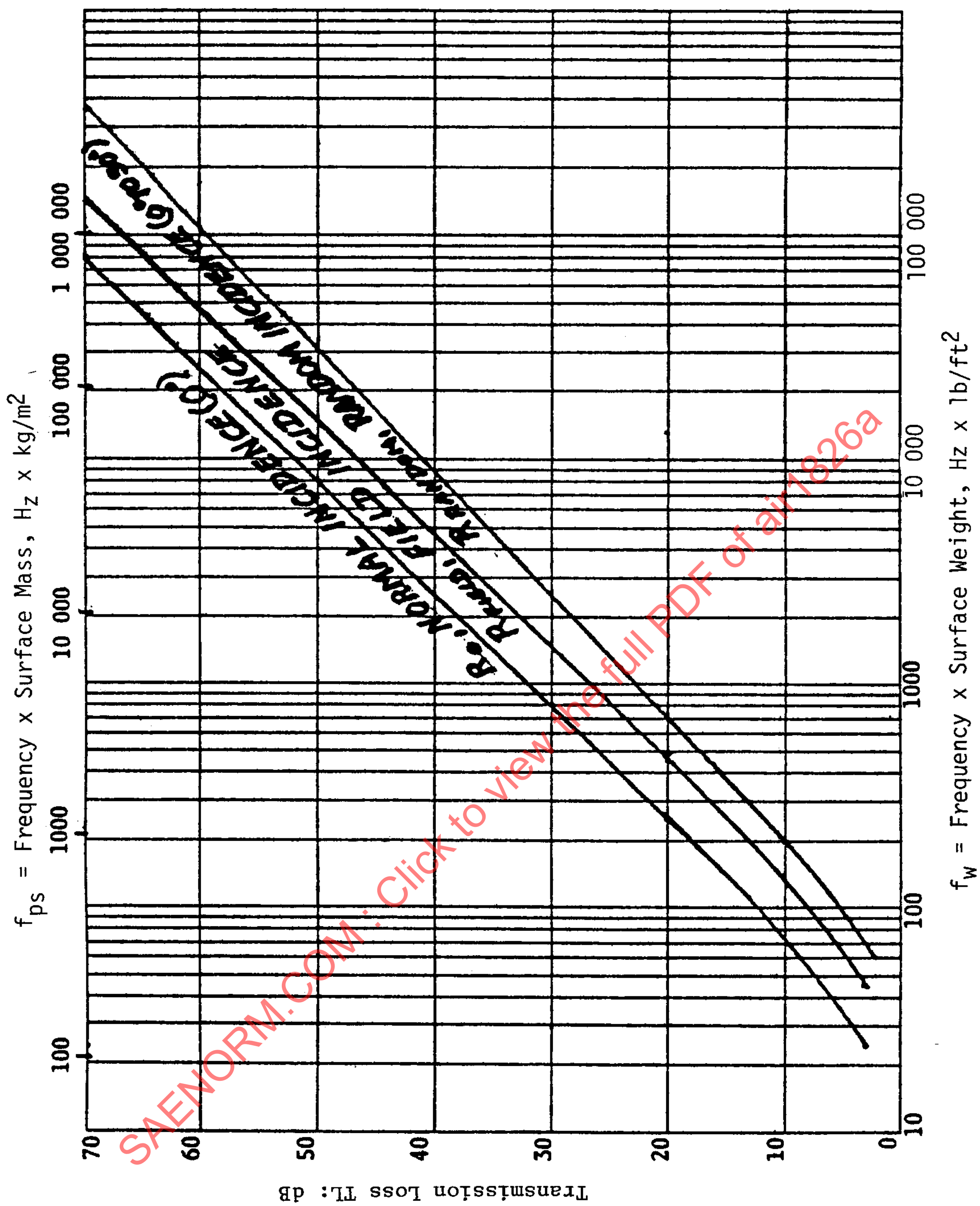

Figure 7 - Theoretical TL of mass-controlled limp panels

NOTE: In Figure 7, field incidence assumes a sound field which allows all angles of incidence up to 78 degrees from normal

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5.4.1 Absorptive Muffler Design

The single resonant panel type is highly frequency selective and inferior to the multiple layer resonant panel type if attenuation is required over broad frequency bands. The multiple layers shown in Figure 8 are essentially equivalent to each other. The required thickness of acoustic panels is dependent on many parameters. For many applications, the thickness should be approximately from 0.1 to 0.2 times the wave length of the lowest frequency to be attenuated. For example, if large attenuations are required at 1000 Hz, at normal ambient temperatures, the wave length would be approximately 1 foot and the panel thickness should be between 1 and 2 inches.

A properly designed lining allows the sound energy to enter the panel without reflections or without returning. This happens only when the motion of air particles in the wave front (by which the wave propagates) is governed at the panel face in an exact, prescribed manner. The governing influences are imposed by the face sheet material and backing geometry. The important characteristics of the lining material are its resistance to air particle motion, called flow resistance, and mass inertia.

These characteristics are related to material sheet thickness and pore size. If a porous or perforated sheet is essentially rigid over the frequency range considered, the acoustic characteristics are independent of the material, but totally dependent on the geometry of the air passages through it. Generally, the air passages should be short, narrow, and crooked. Thin, fine fibrous sheets have the most desirable characteristics. Perforated sheets are acceptable where penalties can be minimized by accurate design criteria (dependent upon environmental conditions) and where other advantages offset remaining penalties. Thickness and hole size in perforated sheets should be minimized. In humid environments drainage should be provided.

Where bulk lining absorbers are applied, a moisture barrier is sometimes necessary to prevent the sound absorber from becoming a liquid absorber. Thin Polyethylene Terephthalate (PET) sheets or bags may be applied for this purpose, but always with some penalties in acoustic performance. Half-mil PET, properly suspended so as to form a flexible diaphragm (not matted against other material), has acceptably low penalties. However, effects of deterioration and rupture of the PET must be considered by the designer.

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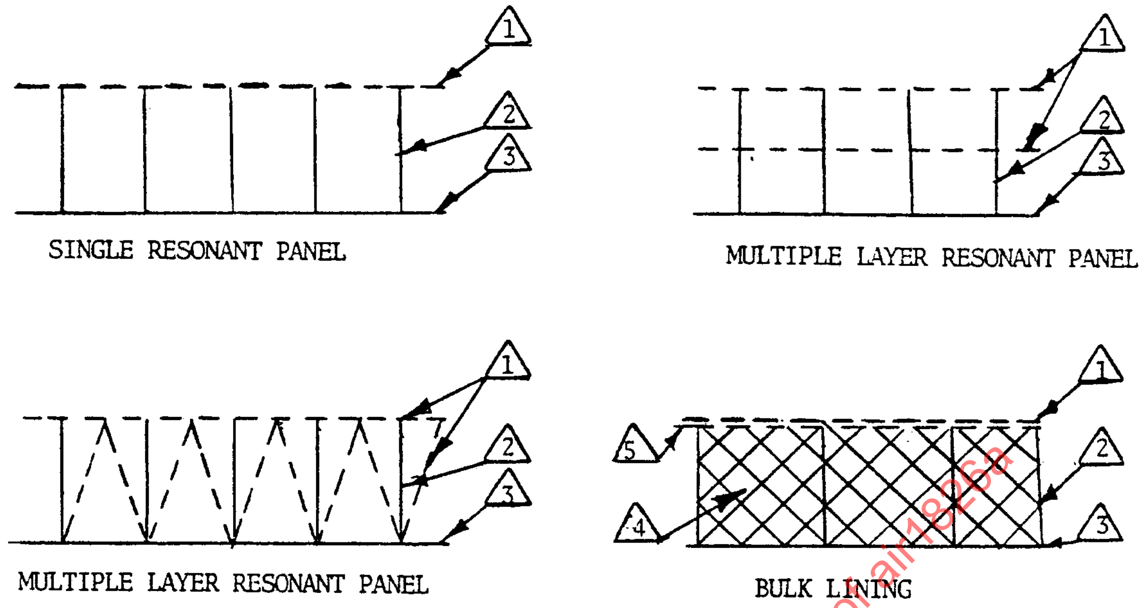

1. Porous sheet of controlled porosity; fibrous or perforated; metallic or nonmetallic.
2. Honeycomb or other partitioning constructions.
3. Impervious backplate (pressure seal for duct walls).
4. Open-pore fibrous material such as fiberglass or open-cell polyurethane foam.
5. Spacer to prevent fibers from plugging pores of face sheet 1 .

Figure 8 - Typical duct lining

NOTE: It is usually possible to choose a lining for a given application from several which would produce equivalent reductions in noise level (Figure 8).

Acoustic panels applied to ducts should be honeycomb structure or divided into partitions. Partition spacing should be approximately equal to the panel depth. If it is necessary to provide drainage paths through the partition, small holes in the partitions may be used.

Acoustic flow resistance is commonly used to determine the type of lining material to be used. Flow resistance (R) is defined as the ratio of pressure differential between the two sides of a plane surface to the average air velocity flowing through that surface (not the velocity through a single pore). Refer to Equation 9.

Acoustic materials are nonlinear, that is ΔP and u do not vary linearly, and resistance changes as these parameters vary. The resistance can be calculated by the equation:

$$R = R_o + R_1 \cdot u \qquad \text{(Eq.11)}$$

Ro and R1 are constants characteristic of the material and may be measured for each specimen in the laboratory.

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Flow resistance and tolerances have not been standardized. Some materials are specified using nominal flow resistance while others use a nonlinearity factor. This results in only two values of R being given (u = 0.2 m/s and 2.0 m/s). A more practical design method for a given application is to specify the highest tolerable flow resistance at the highest possible flow condition and the lowest tolerable resistance at the lowest flow. The supplier of the materials would be required to furnish materials that meet these requirements.

As a guide, an absorptive muffler should have a length to width ratio of 4:1 or more. Such a muffler can be expected to reduce the noise in excess of 20 dB in the frequency region of the maximum effectiveness of the lining material, provided the material is optimally chosen and the entire duct perimeter is treated.

An equation to approximate the noise attenuation in acoustically lined ducts is:

$$Attenuation = 12.6\frac{P}{A}\alpha^{1.4}, \frac{dB}{ft} = \left(9.75\frac{\mathrm{P}}{\mathrm{A}}\alpha^{1.4}, \frac{dB}{m}\right) \quad \text{(Eq. 12)}$$

Where: P = duct perimeter [in (m)]

A = cross-sectional area [in^2 (m^2)]

α = absorption coefficient of lining material, 0 to 1.0.

The limitations of this equation are:

a. Smallest duct dimension should not exceed 18 inches (0.46 m), and not less than 6 inches (0.15 m).

b. Ratio of duct width to duct height should not exceed 2:1.

c. The absorption coefficient should be representative of the entire octave band.

d. Air velocities should not exceed 33 ft/s (10 m/s).

e. The equation does not allow for line of sight propagation of sound which limits high frequency attenuation.

5.5 Vibration Isolation

Vibration isolation is a means of decreasing transmission of vibratory motions of forces from one structure to another. Isolation means the interposition of a relatively flexible element between the two structures. The vibration amplitude of the driven structure is often largely controlled by its inertia. If the isolating element is flexible enough, it will transmit little force to the second structure, except at frequencies in the vicinity of resonance.

The effectiveness of an isolator is measured by its transmissibility, of which there are two types. Force transmissibility is defined as the ratio of the force transmitted to the existing force applied to a mass on it. Displacement (or motion) transmissibility is defined as the ratio of the displacement transmitted through the isolator to the existing displacement applied to it.

Due to interactions between the machine and its foundation, one must contend with their mobilities. Mobility is a complex quantity that is the ratio of velocity and the applied force. An isolator does no good unless it is more flexible (that is, it has greater mobility) than the sum of the mobilities of the machine and foundation. It is rare to get more than 20 dB attenuation at acoustic frequencies with isolators of reasonable stiffness, and it is not uncommon to get no attenuation at all. For this reason, very soft mounts (with natural frequencies of 5 to 6 Hz) are used, and special constructions, such as pneumatic mounts, are now marketed. In any case, if a mount is effective at all, a softer one will be more effective.

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Typical Vibration Isolation Results: An example of progressively more effective combinations of vibration isolation and rigid enclosure is shown in Figure 9. The enclosure is added to illustrate the effect of direct radiation. Enclosures are further discussed in 5.8.

Curve A in Figure 10 shows the octave-band sound pressure level spectrum for a hard-mounted piece of equipment that exhibits a large amount of high frequency energy.

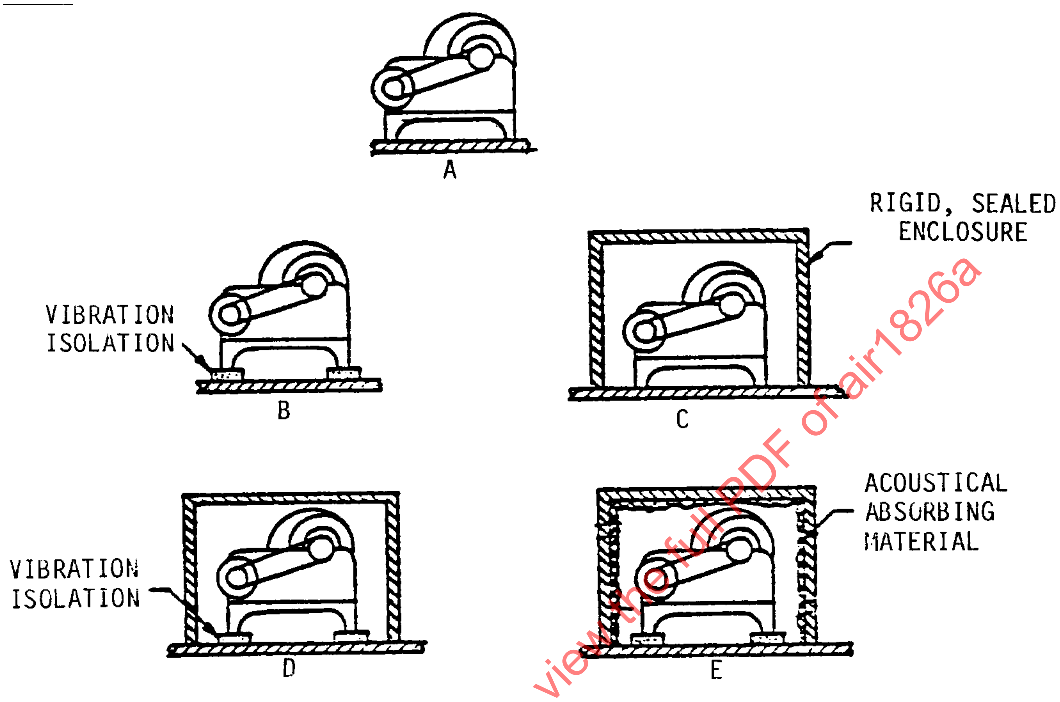

Figure 9 - Examples of noise control methods

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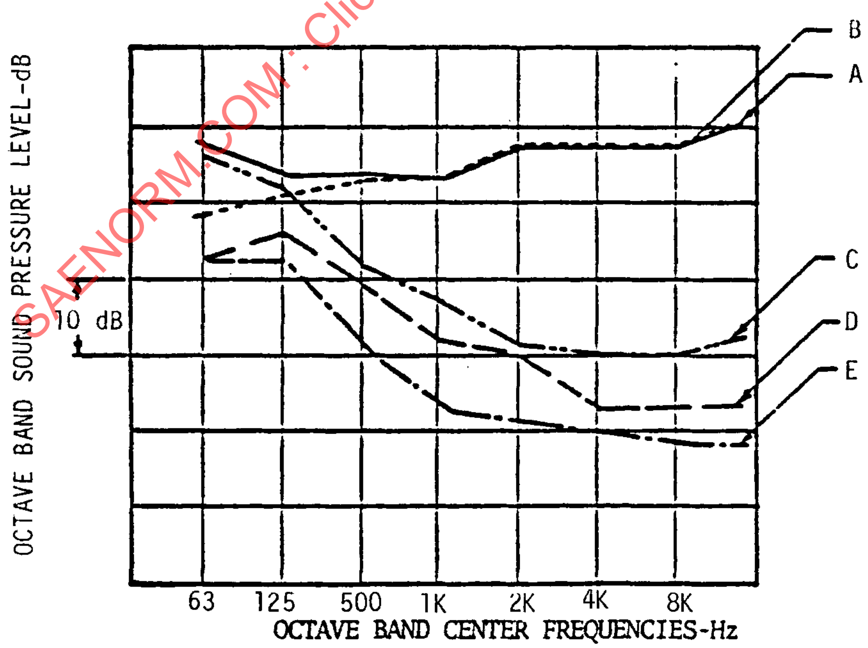

Figure 10 - Typical noise control results

Curve B in Figure 10 shows the octave-band sound pressure level spectrum for vibration isolated equipment. Vibration isolators are important, and the noise reduction is usually at low frequencies. This is due to small equipment radiating high frequency noise directly, the low frequency noise resulting from solid-borne vibration and secondary radiation from walls and partitions.

Curve C in Figure 10 shows the octave-band sound pressure level spectrum for nonisolated equipment mounted in a well sealed, rigid enclosure with the noise level measured outside. Noise reduction is at the middle and high frequency range. Effectiveness is limited at low frequencies due to vibration and solid-borne sound transmitted through supporting structure to the enclosure, causing the enclosure to act as a sounding board. The effectiveness is also limited due to the walls of the enclosure absorbing only a small percentage of the incident sound energy.

Curve D in Figure 10 shows the octave-band sound pressure level spectrum for equipment in a well sealed, rigid enclosure. The equipment is mounted on vibration isolation mounts. The noise level is measured outside.

Curve E in Figure 10 is the same as Curve D with acoustically absorbing material inside the enclosure, reducing internal and consequently external noise levels. The noise level is measured outside.

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5.6 Damping

Damping is the mechanism by which the mechanical vibrational energy of solids is converted into heat. The noise reduction capacity of damping derives from the fact that, since the mechanical energy is dissipated, it cannot be radiated in the form of airborne noise.

Vibration damping materials, by virtue of their mechanical hysteresis, resist motion in any direction and during every portion of the cycle of vibration. Since they provide definite energy losses during each cycle of vibration, their effectiveness is generally proportional to frequency. Hence, a vibration damping treatment of given effectiveness in limiting vibration amplitude at a resonance of one frequency will generally be equally effective at neighboring resonances and more effective at higher harmonics.

Large radiating surfaces such as panels, air flow ducts, plenums, and enclosures benefit most from damping treatment. Large numbers of resonances are present in such structures that are excited by noise, impacts, vibration, or airflow impingement.

Damping treatments usually consist of viscoelastic materials that are applied either in free layers or constrained.

A viscoelastic material is characterized by possessing both viscous and elastic behavior. This is best illustrated in Figure 11, which shows how various types of materials behave in the time domain. For a slab of material with a cross-sectional area, A, and a thickness, T, subject to cyclic loading, F(t), the corresponding response is given by the displacement function, x(t). The cyclic stress on the sample material is found by dividing the input load by the cross-sectional area, and the resulting cyclic strain on the material is found by dividing the displacement by the thickness.

A purely elastic material (Figure 11a) is one in which all the energy stored in the sample during loading is returned when the load is removed. As a result, the stress and strain curves for elastic materials move completely in phase. For elastic materials, Hooke's Law applies, where the stress is proportional to the strain, and the modulus is defined at the ratio of stress to strain.

A complete opposite to an elastic material is a purely viscous material, also shown in Figure 11b. This type of material does not return any of the energy stored during stress at a loading frequency of g is out-of-phase with the strain by some angle ϕ, (where $0 < \phi < \pi/2$, defined in Figure 11c). The angle ϕ is a measure of the materials damping level; the larger the angle the greater the damping.

For a viscoelastic material (Figure 11c), the modulus is represented by a complex quantity. The real part of this complex term (storage modulus, E1) relates to the elastic behavior of the material, and defines the stiffness. The imaginary component (loss modulus, E2) relates to the material's viscous behavior, and defines the energy dissipative ability of the material. Using Hooke's Law to define the modulus for complex values, we can define the complex modulus, E* as:

$$E^* = E_1 + E_2 i = \frac{\sigma_0}{\varepsilon_0} e^{i\phi}$$

(Eq. 2)

The loss factor η is defined as the ratio between the loss and the storage modulus.

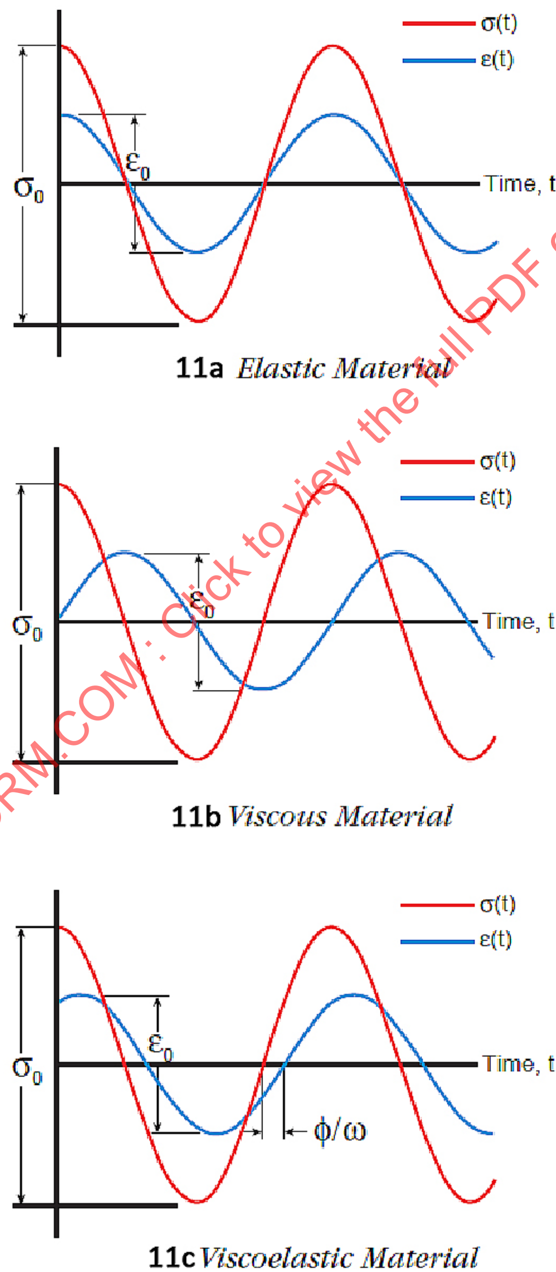

Figure 11 - Cyclic stress and strain curves vs. time for various materials

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These materials exist in various unique states or “phases” over the broad temperature and frequency ranges in which they are used. These regions are typically referred to as the Glassy, Transition, Rubbery, and Flow Regions, see Figure 12. Viscoelastic materials behave differently based on which region they exist in for a specific application.

In the glassy region the polymer chains are rigidly ordered and crystalline in nature, possessing glass-like behavior. Stiffness, E1, is at its highest for the material in this region, and damping levels are typically low. The glass transition temperature, Tg, of a material refers to the elbow of the storage modulus curve at the edge of the glassy region as it enters into the transition region. Tg also defines the peak of the loss modulus, E2, curve.

The transition region is so named because the material is transitioning from the glassy to the rubbery region. It is in this area that the viscoelastic material goes through its most rapid rate of change in stiffness and possesses its highest level of damping performance.

The reference temperature of a material, To, is used to define the peak of the loss factor curve. In this region, the long molecular chains of the polymer are in a semi-rigid and semiflow state, and are able to rub against adjacent chains. These frictional effects result in the mechanical damping characteristic of viscoelastic materials.

In the rubbery region, the material reaches a lower plateau in stiffness.

Damping is at a lower, but reasonable level. A material selected to exist in this region is ideally suited for such devices as isolators or tuned mass dampers because the modulus varies only slightly with changes in temperature and frequency.

In the flow region, the material undergoes both local and global displacements and permanent deformations occur, also defined as "creep".

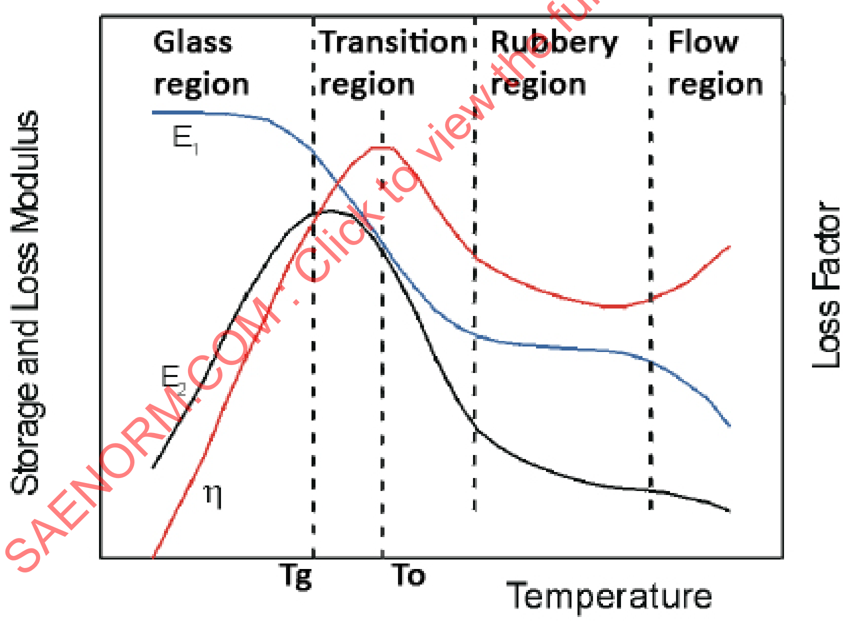

Figure 12 - Variation of modulus with temperature for a typical viscoelastic material

In free layer configurations, strain energy dissipation in the viscoelastic material occurs by extension, and it is the greatest at the antinodes and least at the nodes of the resonant vibrations. Therefore, one may expect only small reductions in structural damping if one removes some of the damping material near the nodes. However, one must keep the uninterrupted stretches of damping material long enough so that they will be made to extend significantly as the composite is flexed.

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Viscoelastic material may also be constrained, such as in damping tapes, where dead soft aluminum foil is bonded by viscoelastic adhesive. In constrained layer configurations the viscoelastic material dissipates energy in shear. Shear in the viscoelastic layer is produced by relative shearing displacements of the elastic layers, and these are greatest at the nodes. Thus, one may expect to lower the effectiveness of the treatment relatively little by removing it (that is, the viscoelastic layer, the constraining layer, or both) near the antinodes. Interrupting the constraining layer may result in increased shear at the points of interruption, thus increasing damping. Continuity of the viscoelastic layer under a continuous constraining layer is unimportant.

5.7 Wrappings

Aircraft air distribution system components are usually wrapped with fiberglass blankets for thermal insulation. The wrappings can provide a significant amount of damping depending on the thickness and the attachment of the insulation.

Wrappings are usually made either from a layer of porous material alone, or from one or more layers of porous material covered and separated by impermeable layers. Wrappings are typically applied to noisy ducts, valves, or plenums of the ECS. Thermal insulation applied to fresh air ducts can be designed to act as a noise reducing wrapping, thus serving double duty.

A porous material, when used as a sound attenuating layer, attenuates a sound wave partly by acting as a reflecting surface - as does a solid wall - and partly by conversion of the acoustic energy of the sound that penetrates the material to heat by viscous losses in the interstices. Some materials, such as fine fiberglass or mineral-wool blankets are so effective at high frequencies, that a sound wave at 1000 Hz is attenuated 60 dB in traveling 1 foot (0.3 m). As an example, a 2 inch (0.05 m) thick fiberglass blanket with 0.6 lb/ft^3 (10 kg/m^3) density and 1 μm fiber diameter wrapped around a duct would reduce the duct noise by 10 dB at 1000 Hz and by 26 dB at 8000 Hz, but it would be ineffective under 500 Hz.

The noise reduction of a porous blanket may be improved by covering it with a limp impervious membrane such as mass loaded leaded vinyl. If the blanket in the above example would be covered by a 0.5 lb/ft^2 (2.5 kg/m^2) lead vinyl, the noise reduction would be 35 dB at 1000 Hz and 70 dB at 8000 Hz; however, it would go to 0 at 180 Hz.

5.8 Enclosures

Enclosures are rigid, usually airtight boxes surrounding noise sources. The transmission loss of the enclosure wall has to be considerably higher than the required noise reduction for the enclosure. The smaller the volume of the enclosure, the higher the noise inside. Thus, when a machine is placed inside an enclosure, the sound level inside the enclosure is much higher (sometimes by 10 or 20 dB) than the original sound level at the same distance from the unenclosed source radiating noise in the open. This occurs because the enclosure attempts to capture the total power radiated from the machine and the internal absorption is not usually sufficient to simulate free field conditions (see Figures 9 and 10).

It cannot be emphasized strongly enough that a well sealed and highly absorbent enclosure is necessary for acceptable effectiveness. To further illustrate the performance of a partial or unsealed enclosure, the following table is presented:

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Table 4 - Maximum achievable enclosure noise reduction

Sound Energy Enclosed and Absorbed Percent	Maximum Achievable Noise Reduction dB
50	3
75	6
90	10
95	13
98	17
99	20

5.9 Active Noise Control

All noise suppression methods previously described in this chapter are defined as "passive" because they achieve sound pressure reduction (soundproofing) with passive means, by either blocking the sound path (noise reduction) or by damping the impinging waves (noise absorption).

Active Noise Control System (ANCS) is based on the principle of destructive interference in waves (also known as “noise cancellation”): since sound is a pressure wave, it can be cancelled in any given point of space by another sound having the same amplitude and an opposite phase. The term "active" is used because an ANCS is able to determine the amplitude and frequency of the unwanted sound (noise) to be cancelled and to consequently generate the “anti-noise” to suppress it.

Active methods are preferred for attenuation of low frequency noise (below 500 Hz) since the characteristic thickness required by passive barriers to be effective at those wavelengths would be impractical (about 3.4 meters at 100 Hz). On the other hand, complexity, cost and power consumption suggest the use of passive suppression methods at higher frequencies.

Current aircraft applications of ANCS have been successfully carried out for turboprop aircraft, in which the interior noise is dominated by harmonic tones of the propeller blade passage frequency (BPF): ANCS are designed to suppress these tones, achieving an attenuation that can be quantified in 6 to 20 dB, achieving a significant improvement of the cabin acoustical environment. However, active noise control is still considered impractical for global reduction of broadband or high frequency tonal noise in aircraft cabins.

An ANCS is made by the following components: one or more input sensors (microphones) to detect the unwanted noise (primary source) and measure the achieved cancellation level (error), a process unit that generates the required signal (controller) and one or more output transducers (control sources) to emit the cancelling sound.

Effective noise attenuation depends not only on the accuracy of the detected amplitude and phase, but also on manufacturing defects in the “anti-noise” emitting devices and on physical limitations: to achieve an effective suppression, the control sources shall be adequately large and located so that they can significantly affect the noise radiated from the primary source. Where a relevant discrepancy exists between required and achieved conditions, the result may either be a general increase in the perceived noise (resonance) or a cancellation achieved only locally, with primary noise being amplified in the remaining part of the field of interest.

ANCS can follow either feedforward or feedback control approaches, as shown in Figures 13 and 14.

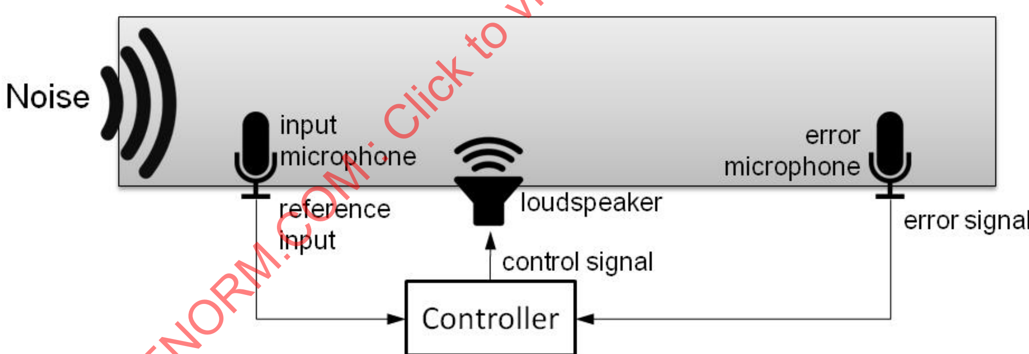

Figure 13 - ANCS, feedforward control approach

In the feedforward system of Figure 13 an input microphone samples the incoming noise and feeds the controller which, in turn, processes the input according to a predetermined transfer function, generating the driving signal for the control source. The error microphone monitors the effectiveness of the process, providing a correction signal to the controller which adjusts the control signal to minimize the detected error.

In most applications primary noise can also be predicted as a function of a measurable quantity (e.g., rotational speed in a fan), so the input microphone can be replaced by another input device (a tachometer in the case of fan noise). This technical solution helps preventing acoustic coupling between the input and the control signal, which leads to instabilities.

Many factors contribute in altering the characteristics of the primary source transfer function (e.g., noise randomness component, environmental conditions, aging components) thus a feedforward system must be self-tuning, continuously adapting itself to the changes occurring in the system being controlled.

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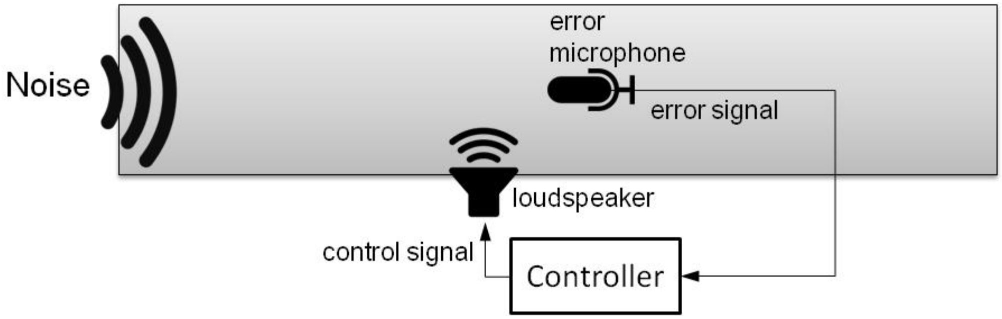

Figure 14 - ANCS, feedback control approach

In the feedback system of Figure 14 the control signal attenuates the incoming noise with the objective of minimizing the residual signal detected by the error microphone. The controller only processes the error signal.

The bandwidth over which a feedback system is effective is inversely proportional to the delay existing between the controller input from the error sensor and the controller output to the loudspeaker. Non-adaptive feedback systems are commonly adopted for noise cancellation earphones.

Feedforward systems are inherently more stable and are more effective at reducing steady state noise, while feedback systems are better at reducing transients.

6. APPLICATION OF NOISE SUPPRESSION METHODS

6.1 Cooling Units

6.1.1 Air Cycle Machine

An Air Cycle Machine (ACM) cooling unit is the assembly of a compressor, a high speed turbine and a fan that may generate airborne noise at its air outlet, propagates structureborne noise through its mounts and ducts, and radiates noise into the surrounding space from its casing.

The air outlet of the cooling turbine is ducted, through a water separator, into the passenger cabin, and, since the components provide no noise attenuation, a muffler is usually required. The turbine outlet noise has a rising spectrum toward the high frequencies that may contain tones at the blade passage frequencies. The noise may be aggravated by high velocity turbulent air flow in the ducting. The muffler design (see 5.4) should incorporate a humidity barrier as the airflow (even past the water separator) is at 100% humidity. If a bulk absorber is used in the muffler, the foam or the fiberglass may be bagged with a foil, such as Polyethylene Terephthalate, but the foil thickness cannot exceed 0.013 mm (0.0005 inch), and it may not be in contact with any protective screens.

Structureborne noise may be a problem if the rotating assembly is not well balanced to 1 g or better. The unbalance noise that is a pure tone at the shaft rotational frequency of the ACM may be heard in the cabin if the ACM is not located in a sufficiently remote location. Vibration isolation is beneficial (see 5.5) but may not provide a solution to the unbalance problem. It is difficult to design ACM vibration isolators to be soft enough to provide isolation and to adequately resist the high thrust loads experienced in an aircraft. In addition, ACM ducts can transmit a lot of vibration, especially if the ACM is soft mounted. Vibration isolating duct couplings are ineffective because soft couplings have to be restrained to take the working pressures, this restraining can render them ineffective. For example, with properly isolated ACMs the maximum cabin tone level can be expected to be about 65 dB in the 300 to 600 Hz frequency range for a 1 g machine. This level may still be acceptable in a passenger aircraft but the noise increases at the rate of 20 log a (acceleration), meaning that a 10 g machine would generate an 85 dB tone - clearly far too excessive. In cases where the ACM must be located near the occupants, that is, separated from the cabin by a bulkhead or a pressure deck, one solution is to use vibration free machines that maintain their balance even with typical service degradation such as nozzle erosion. The use of air bearings will help to ensure a good balance.

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The case radiation of an ACM is usually not a problem due to the adequate transmission loss provided by structure. In cases where the case radiation noise, rich in the high frequencies, has to be attenuated, an enclosure has to be provided (see 5.8).

6.1.2 Vapor Cycle Machines

The Vapor Cycle Machines (VCM) serves the same purpose as an ACM but utilizes a refrigerant compressor in a refrigeration cycle to cool the cabin airflow supplied by the evaporator fan. (A supplemental VCM may also be used to cool equipment separate from the cabin.) The noise source is then the compressor through structureborne paths and in the form of case radiation.

Electrically-driven compressors should be vibration isolated (see 5.5). During procurement, the structureborne noise level should be specified. A helpful document is MIL-STD-740-2 that defines structureborne noise level and its determination. The structureborne noise level of the compressor measured in vibratory rms acceleration should not exceed 120 dB in any one-third octave band re: 10^{-5} m/s^2. This is equivalent to 1 g rms acceleration - the upper limit that was recommended in 6.1.1.

If compressor case radiation is a problem, an enclosure may be necessary (see 5.8).

6.1.3 Ram Air Inlets

The large amount of outside air required for cooling units is ducted via ram air scoops in flight. On the ground this inlet usually serves the ground cooling fan. During flight, noise problems can be caused by these scoops due to high velocity airflow excitation of the structure and/or resonant excitation of the cavity served by the inlet.

Once the structure is excited to a point where the noise is objectionable in the cabin, not much can be done about it. The only solution seems to be the proper control of the airflow through the system. The amount of the airflow is controlled by the inlet, the outlet, or both. The preferred solution is an inlet door to prevent a high level of ram air pressure build-up with attendant spurious noise problems. If the outlet has to be regulated, provision for a minimum amount of airflow has to be made in order to prevent an unstable condition inside the duct and a plenum chamber downstream. It is very important to duct the air through the system smoothly with proper diffusion. Incorrect design of this duct results in loud roaring and whistling noises in the cabin.

6.2 Bleed Air High Pressure Pneumatic Equipment

6.2.1 Air Turbine Motor

An Air Turbine Motor (ATM) is a high speed turbine that drives a hydraulic pump or other accessories. Airborne noise is generated at the air discharge and structureborne noise is propagated through its mounts. The ATM hydraulic pump can be another noise source that causes high structureborne noise levels transmitted through the ATM mounts and the hydraulic lines.

The air exhaust can affect ramp noise and it may require a muffler (see 5.4) that has to be effective through a wide frequency range up to 11000 Hz since the blade passage frequencies of the turbine dominate the broadband spectrum.

Careful balancing of the rotating assembly is important. Unbalance should not exceed 1 g, otherwise cabin noise may become a problem if the ATM is mounted on the fuselage. In such installations, the hydraulic pump may require a vibration damper or some other pulse cancellation device.

6.2.2 Emergency Power Units

Emergency power units include hydraulic or electric power units driven by gas generators or ram air.

Noise control measures are usually not warranted for these sources due to their truly emergency operation when noise is not objectionable. An exception may be in the case of a really startling noise that could frighten passengers.

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6.2.3 Control Valves

Two major noise producing mechanisms are associated with flow/temperature control valving. The first is the turbulent mixing process downstream from the valve and the second is the shock noise, both are associated with choked flow. Choked flow occurs when the flow speed equals the speed of sound in the gas. For valves with pressure ratios less than 3 (ratio of upstream pressure to downstream pressure) the noise is associated with both mechanisms. At pressure ratios greater than 3, the shock noise predominates. Reduction in flow speed (larger flow areas) will reduce sound power levels. Most shutoff valves used to control ECS flow are of low ΔP design. These valves, which have aerodynamically smooth internal surfaces, are usually not critical noise generators. However, noise components may occur under certain conditions, (e.g., excitation of edge tones). Once such an excitation mechanism is identified, the generation of discrete tones can usually be controlled by de-tuning (rounding off sharp edges or corners) or cutting feedback paths by treating reflecting surfaces with a sound absorbent layer. Excessive freeplay in moving parts should be avoided.

6.2.4 Bleed Air Discharges

It has been the practice on military aircraft to limit air velocity in bleed air distribution systems to Mach 0.25 max. This reduces the effect of air velocity on structural fatigue and noise. This critical value generally applies to ducting that is isolated and insulated from the crew and passenger compartments. When discharge velocity is below 100 m/s (330 ft/s), the major portion of aerodynamically generated noise is usually attributed to interaction between flow and obstacles in the ducting. Typical obstructions are changes in cross section, struts, stringers, or guide vanes. Reducing the pressure drop of such obstructions by rounding off sharp corners, cutting feedback paths by treating reflecting surfaces with a sound absorbent layer, use of airfoils rather than grids or oddly shaped bodies will reduce the discharge noise level.

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6.2.5 Jet Pumps

In some commercial and military transport aircraft, jet pumps are used in the ECS to heat cargo compartment floors from beneath and to exhaust air from galley and lavatories. In some business aircraft, jet pumps are also used in cabin air distribution systems to improve air circulation and in emergency pressurization systems to provide an alternate inflow source.

Jet pumps are very simple in design with no moving parts and the high pressure air necessary to operate the pump is readily available as bleed air from the aircraft power plants. A sketch of a typical jet pump is shown in Figure 15.

Jet pump noise is the result of flow noise generated during the turbulent mixing process in the shear layer as the expanding air leaves the nozzle. Since there are no obstacles in the ensuing jet, the resulting forces can occur only in opposing pairs because no counterforce from an obstacle is present. These fluctuating pairs are called quadrupoles, and the sound power level associated with a quadrupole source is proportional to the eighth power of the velocity of the jet (u^8). The sources of noise are distributed over a considerable length downstream from the nozzle. Assuming that the nozzle diameter is small compared to the duct diameter, the jet may be considered a free jet. Then, the frequency spectrum and directivity of the air jet may also be determined. The frequency spectrum of a free jet exhibits broadband character with a shallow peak occurring at a frequency that is dependent on nozzle diameter and exit velocity. The location of this frequency may be determined from the nondimensional empirical relation:

$$S = \frac{f_D D_N}{u} = 0.2 \quad \text{(Eq. 14)}$$

Where: S = Strouhal number

f_D = Peak frequency, Hz

D_N = Nozzle diameter, m (ft)

u = Exit velocity, m/s (ft/s)

Standard practice is to measure the noise level at a distance of 1 meter from the jet pump. The generated noise level commonly exceeds 85 dB and depends on the pressure and flow ratios of primary to secondary ports.

The simplest method for reducing noise generated by jet pumps, especially for frequencies greater than 500 Hz, is to insert an oversize duct with an internal lining of noise absorptive material downstream or upstream of the nozzle. The lining material should be selected so that the highest attenuation will occur near the peak in the jet noise spectrum. For example, a high density open cell foam with low porosity will tend to absorb more of the low frequency noise and reflect more of the higher frequency noise. The material selected should have a reasonable service life and durability that is needed to withstand moderate temperatures and constant use with exposure to aircraft cabin contaminants such as oil, tar, nicotine, and hydraulic fluids. Because of the severe environment, the design should incorporate means to readily replace the absorbent material.

As an example, the noise downstream of a jet pump in a bare duct was effectively reduced by the addition of a noise absorptive material lining in the duct. The noise reduction was as high as 40 dB for the higher frequencies as shown by Figure 16, where the mid frequency (150 to 500 Hz) resonance is due to the cavity of the oversized duct.

A comparable level of attenuation can also be achieved by adding a muffler downstream the jet pump. To be effective, the muffler should have a winding internal airflow passage, in which airflow is first reversed and then brought back to the original direction. This solution works as a low pressure loss, low attenuation reactive muffler (see 5.4).

Jet pump nozzle design should also be taken into consideration. In most jet pumps, the flow is choked at the nozzle exit. Since noise absorptive material is generally more effective in attenuating higher frequency noise, it is best to increase the peak frequency (as defined above) of the source. Therefore, if a change in nozzle size would result in an increase in the peak frequency, then the change would be desirable.

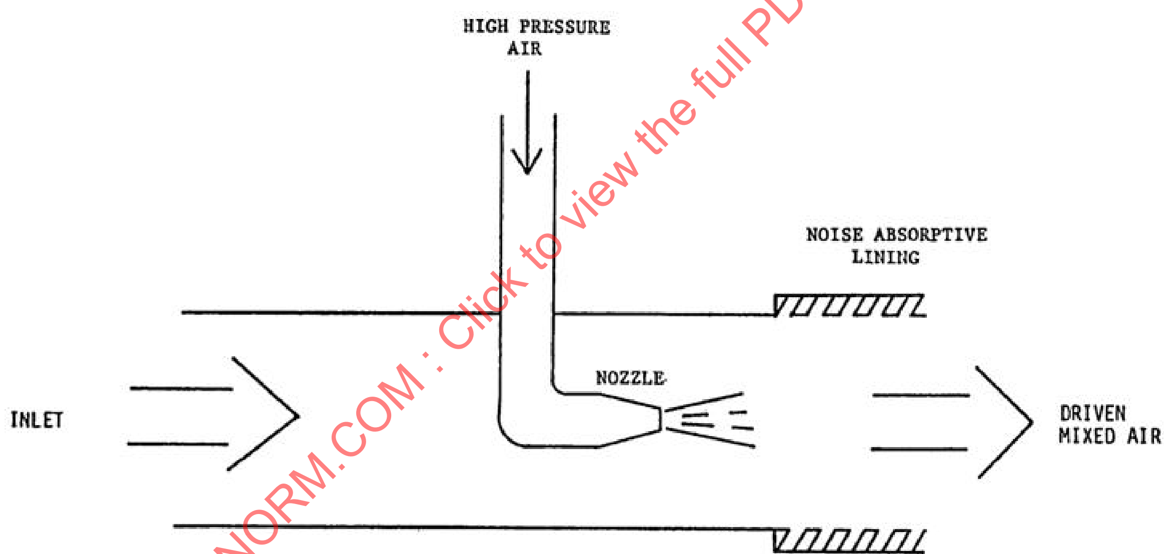

Figure 15 - Generalized schematic of jet pump and noise absorptive lining

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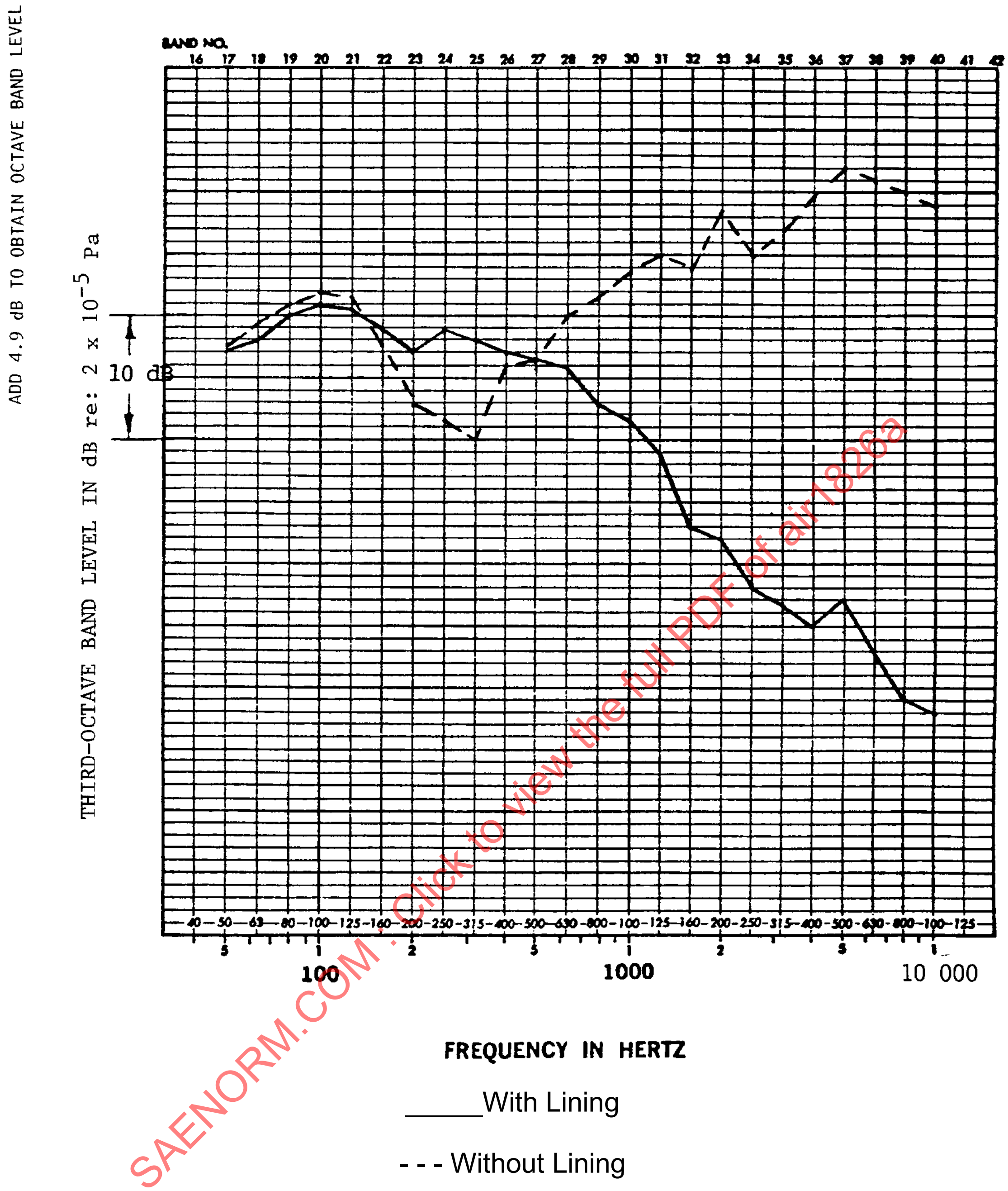

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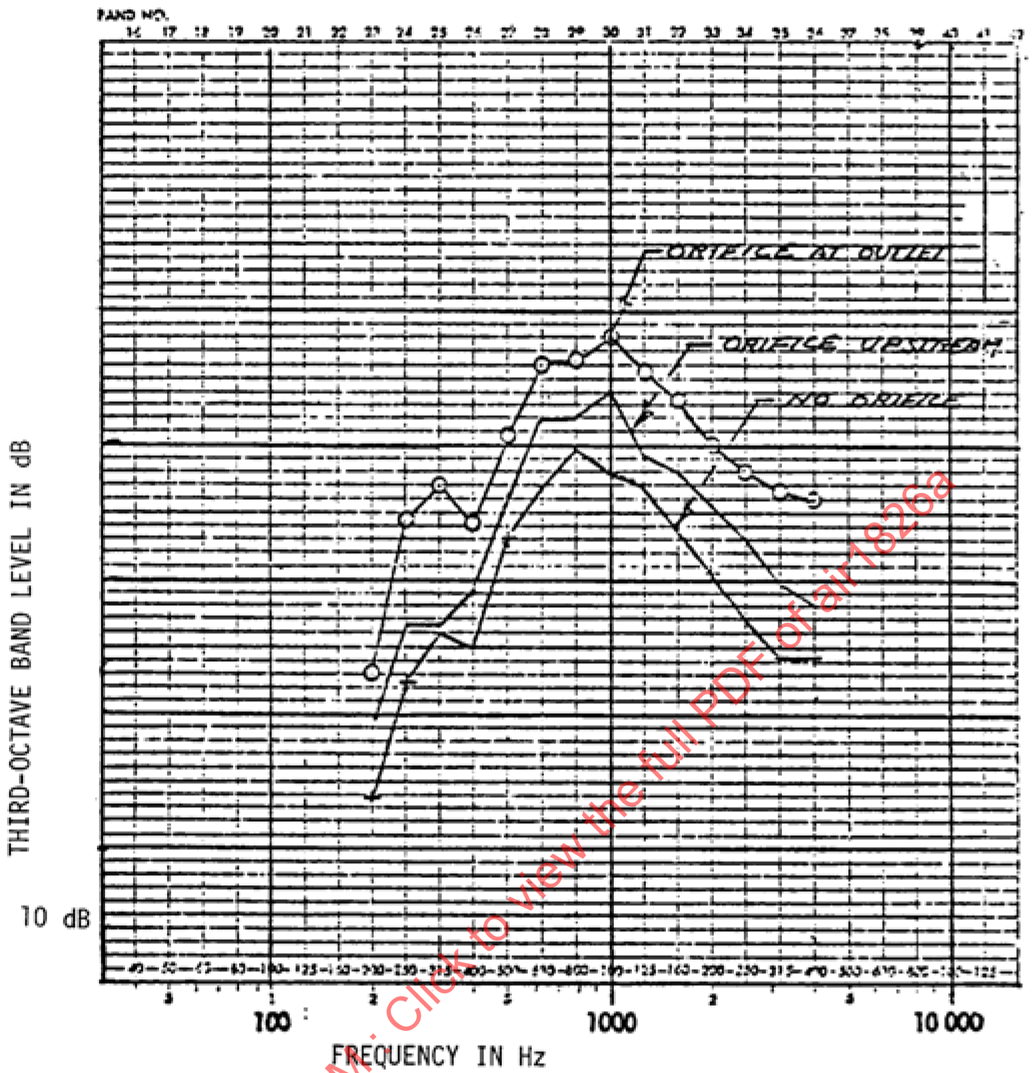
Figure 16 - Comparison of noise downstream of a jet pump used to pull air from lavatories and galleys

6.3 Air Distribution Systems

6.3.1 Ducting

Internal duct noise generated by the velocity of air flow, which induces aerodynamic vibration in the duct walls, is reradiated as sound. Duct air velocity is usually the product of sizing based primarily on cooling flow requirements, installation space limitations and weight considerations. If noise is not evaluated in the initial design, then the resulting air velocity is most likely to be too high for ideal noise levels. Reduction of air velocity by increasing duct size may not always be feasible as a noise problem solution. Plenum design, insulation, and distribution of air must be considered. Location of flow balance orifices as used in ducting upstream of compartment outlets should be located at least 8 diameters of duct diameter upstream of the outlet for significant noise reduction. An example of noise comparison by test is shown in Figure 17.



DUCT DIA. = 0.032 m (1.26 in)
ORIFICE DIA. = 0.017 m (0.67 in)
AIR VELOCITY WITH NO ORIFICE = 36 m/s (118 ft/s)
ORIFICE JET VELOCITY = 109 m/s (357 ft/s)

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Figure 17 - Outlet noise test

Design Recommendations

a. The air velocity in the system should be kept to a minimum and preferably not to exceed 50 ft/s (15 m/s).

b. The air velocity in the distribution system should be kept as close to being the same as possible. (See Figure 18.)

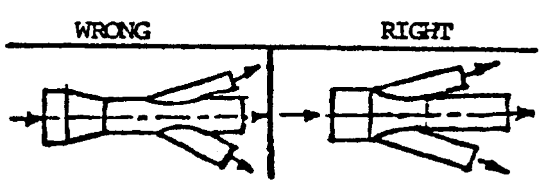

Figure 18 - Duct branching

c. There should be no elements producing a discrete disturbance within 8 diameters of an outlet. (See Figure 19.)

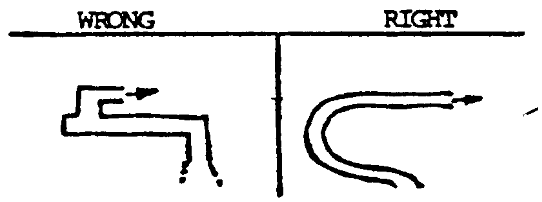

Figure 19 - Duct routing

d. There should be a minimum number of elements producing a discrete disturbance in a distribution system. (See Figure 20.)

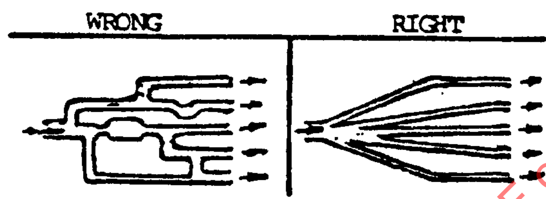

Figure 20 - Duct networking

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6.3.2 Plenums

A plenum is defined as an enclosed space in which the air pressure is greater than that of the outside atmosphere. By this definition, most of the ducting in an environmental control system are plenums. However, when plenum chamber design for performance as a noise reduction element is desired the following recommendations can be made:

a. The interior surfaces should be covered with sound-absorbing material with consideration given to the local flow velocities (see Figure 21).

b. The inlet and the outlet should not be located directly opposite to each other.

When the absorption coefficient of the lining is large, the transmission loss of the plenum is limited by the direct transmission between inlet and the outlet. The transmission loss of a special multiple-chambered plenum (see Figure 21) is more effective than the ordinary single-chambered plenum. For further information see 5.4 on mufflers.

6.3.3 Outlets

Excessive duct outlet noise is probably the most conspicuous irritant to crew and passengers. In the design of compartment outlets, consideration should be given to outlet air velocity into the distribution plenum (less than Mach 0.1 if possible) and the velocity of air as it passes crew or passengers should not exceed 1.6 ft/s (0.5 m/s) (see MIL-STD-1472) and should, for optimum comfort, be between 20 and 40 fpm (0.1 and 0.2 m/s) (see AIR1168/3). When air flow at the diffuser is below 330 ft/s s (100 m/), jet related noise is not significant. However, grid noise as a result of an interaction between flow and rigid body such as guide vanes and grilles can constitute a sound generating mechanism. See 6.3.4 for further discussion of grille noise.

Figure 22 is an example of a test crew air outlet that was designed for the low noise levels shown. The design includes the application of a sound absorption material and a double diffuser screen.