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A PRELIMINARY DESIGN STUDY ON AN ACOUSTIC MUFFLER FOR THE LAMINAR FLOW TRANSITION RESEARCH APPARATUS

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1.0 INTRODUCTION

Reducing aircraft drag is one of the present principal foci of aeronautics research within NASA Langley Research Center. An important element within this program is study at a basic level of laminar flow and the mechanisms of transition.

This report concerns one aspect of the design of a new research tool for studying laminar flow and the mechanisms of transition, the Laminar Flow and Transition Research Apparatus (LFTRA).

Since the presence of acoustic pressure fluctuations is known to affect transition[1-3], low background noise levels in the test section of the LFTRA are mandatory. In fact, it is anticipated that some experiments to be conducted in the LFTRA will concern the influence of acoustics on transition. While, for some years it has been recognized that turbulent intensity may be increased by the presence of sound[1-3] it has also recently been demonstrated that sound can suppress turbulence[4,5].

In designing a special purpose apparatus for the study of laminar flow and transition, the obvious principal requirement is the removal of all those factors which are known to cause premature transition in existing wind tunnels. High on the list of importance among these factors is acoustic noise. Present wind tunnels have usually been designed to maximize efficiency in order to obtain the maximum possible flow velocity for a given test section. Acoustic considerations have been of secondary importance resulting in rudimentary muffler arrangements or no muffler at all. The consequence of this historical fact is that existing wind tunnel test sections experience noise levels which are totally unrepresentative of the aircraft flight environment. An additional consequence is that the study of laminar flow and transition in most existing facilities is severely limited.

The proposed LFTRA is a continuous flow non-recirculating wind tunnel. Air is sucked in through an intake containing honeycomb flow straighteners and screens for ensuring good quality low turbulence flow (Figure 1). The flow is contracted into the test channel (of section .7m x 1.2m) and then ideally through a short low turbulence muffler/diffuser to a centrifugal blower.

The goal of this study was the preliminary design of a muffler for noise generated by the centrifugal fan of the proposed LFTRA. Since the muffler has to be integrated with the fan and the test section, a subsidiary goal was to investigate possible trade-offs in accomplishing this system integration.

A basic goal of the LFTRA is a noise level in the test section no greater than 60 dB in any octave band. This goal was set on the basis that under all circumstances this noise level would be about 10 dB below the flow self noise level in the test section and thus would be insignificant in initiating transition.

However, since an octave band spectrum of 60 dB is also representative of typical room noise level, this requirement is extremely stringent. This report describes the initial approach to the design of a muffler for the LFTRA, the difficulties which were encountered in integrating the muffler with the remainder of the system and outline a possible solution to these problems.

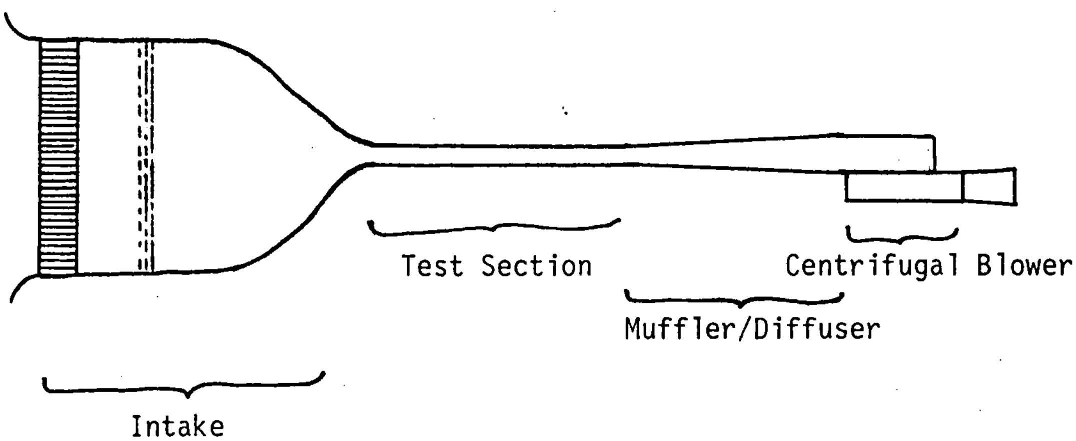


Figure 1. Idealiz ed Layout of LFTRA

2.0 MUFFLER DESIGN BACKGROUND

2.1 Modeling Considerations

Mathematically, the linearized, inviscid, duct acoustics problem is usually formulated as a boundary value problem in quantities representing small perturbations of the basic conservation of momentum, energy and continuity equations of fluid dynamics. Two distinct sets of equations result from the perturbation process and these are considered to represent two distinct phenomena within the physical system:

- The unperturbed mean aerodynamic flow field, which is itself unaffected by the presence of acoustic energy propagating through it, and,
- The acoustic field whose total energy content is unaffected by the mean flow but whose local energy propagation characteristics are affected (sometimes dramatically) by it.

Modeling of the muffler for the LFTRA is a complex task due to the three dimensional nature of the geometry, acoustics and the mean flow. Initially it was anticipated that mathematical modeling for muffler design would be accomplished using the ADAM System[6]. Preliminary modeling and optimization was in fact performed using this system with a two dimensional spatial discretization in the X-Z plane as shown in Figure 2. However, use of the ADAM System proved expensive in terms of computer time for conducting parametric and optimization studies for a large number of geometries and frequencies. Rather than restrict the scope of these studies and also because the refinements for which the ADAM System is designed were not appropriate to the gross nature of the analyses required for this study, a simpler and more cost effective modeling tool was sought.

Omission of mean flow and variable geometry effects result in a great simplification of the equations. In the case of the LFTRA muffler, maximum flow velocities are likely to be in the range Mach .1 to .2. At these flow velocities, omission of the effects of mean flow is unlikely to cause serious error. Also, for reasons of maintaining good flow quality, duct geometry is required to vary slowly. Thus, successive analyses of several uniform duct sections provides a good approximation to the continuous problem.

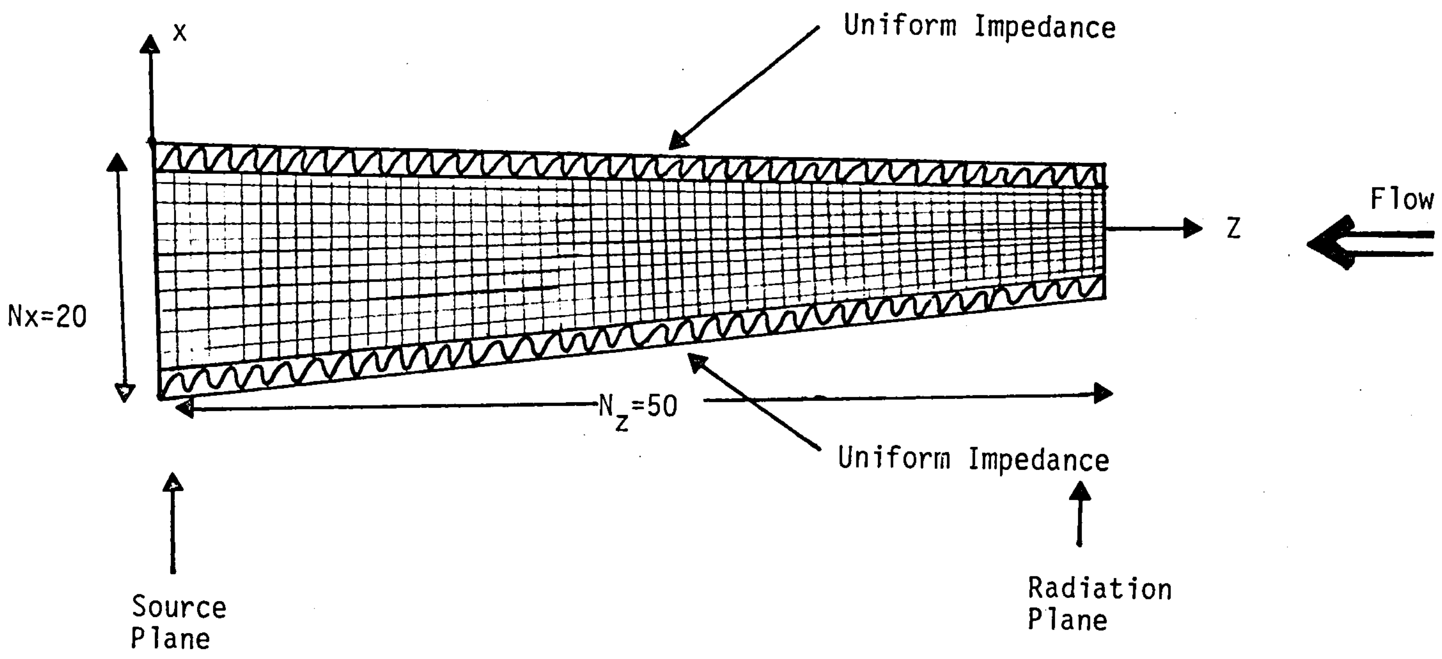


Figure 2. Discretization for Initial Muffler Parametric Study and Optimization Using ADAM System

Accordingly, an alternate simplified model based on a modification of the analysis in References 7 through 9 was developed.

The analysis of Reference 7 is based on a no flow solution of the acoustic equations in terms of a set of acoustic modes propagating in the Z-direction. The spatial dependence of the modes in the X-Y plane as well as the modal propagation constants are found in terms of the eigenfunctions and eigenvalues of a variational finite element formulation resulting from a discretization of the duct in the X-Y plane (Figure 3).

Duct geometry is assumed to be independent of Z. In Reference 7, duct wall admittances are allowed to vary with distance along the walls, that is $\beta_1 = \beta_1(x)$, $\beta_2 = \beta_2(x)$, $\beta_3 = \beta_3(y)$, $\beta_4 = \beta_4(y)$. Since this sophistication was considered unnecessary in the present case, the solutions separate and the two dimensional solution can be reconstructed from separate one dimensional solutions in x and y (see Appendix 1). A highly efficient algorithm suggested by Watson for this simplified case comprises two one dimensional finite element eigenvalue solutions in the x and y with subsequent reconstruction of the two dimensional solution.

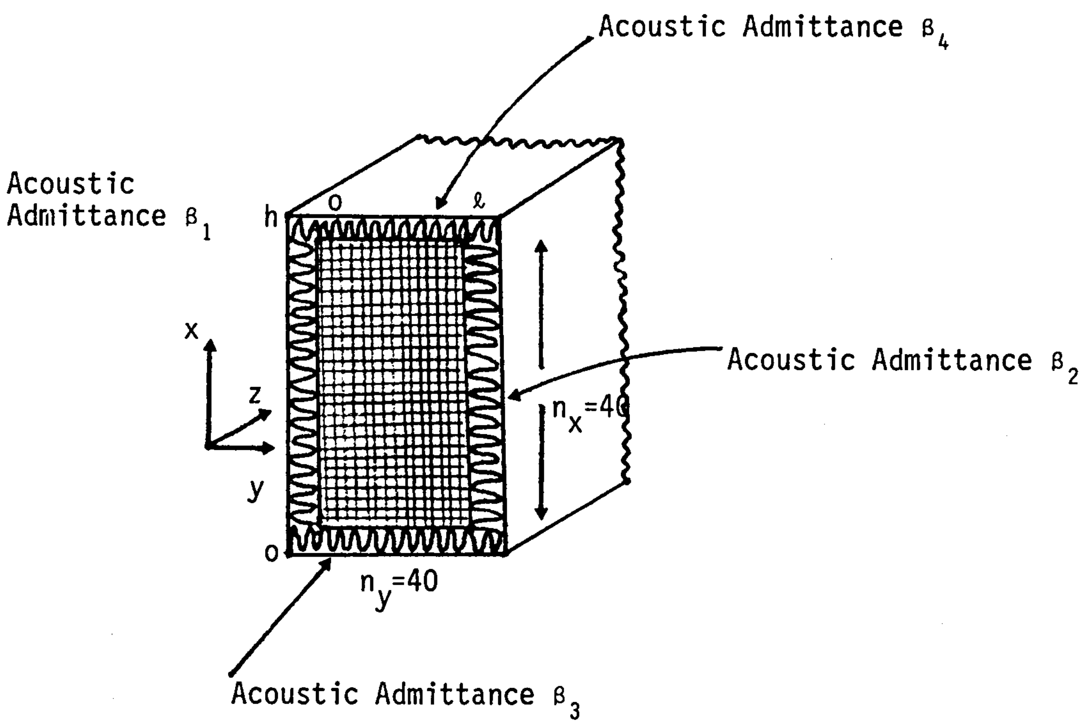


Figure 3. Simplified No-Flow Model Discretization

Argueably for the present case of the LFTRA muffler which is of rectangular section, this model in spite of its simplifications is more suitable than the ADAM System which is designed for axisymmetric ducts. Thus the effects of all four duct walls can be included with the simplified model, rather than just the two closest walls as in the annular duct ADAM approximation.

2.2 Preliminary Design Goals

The design goals for the purposes of this study were established by NASA LaRC. These goals are represented by the requirement for ambient noise levels, in the test section of the LFTRA during operation at Mach .3, not to exceed 60 dB in octave band spectrum level.

To evaluate the feasibility of these design goals in the context of semi-standard existing equipment from recognized manufacturers, data from two companies Howden Fans and TRANSCO are presented below.

Preliminary data from the fan manufacturer, "Howden Fans" on a 120,000 CFM centrifugal blower operating at a pressure differential of 36 inches of water give the following octave-band sound pressure levels measured upstream of the blower:

Frequency (Hz)	31	63	125	250	500	1000	2000	4000
SPL (dB)	130	132	131	133	124	120	115	110

Connecting a "TRANSCO" commercial muffler with a pressure drop of four inches of water to the upstream side of the blower, reduces this noise spectrum to the following:

Frequency (Hz)	31	63	125	250	500	1000	2000	4000
SPL (dB)	106	98	86	68	57	65	65	59

The sound pressure levels given here represent those which Howden and TRANSCO would be prepared to guarantee. Expected sound pressure levels are 3 to 5 dB lower.

Clearly then, the problem area still to be dealt with is the low frequency region from say 20 Hz to 200 Hz where an additional attenuation ranging from 10 to 40 dB may be required.

Additional design goals for the muffler are a low pressure drop and minimal self noise from sound radiated by turbulent flow at side walls.

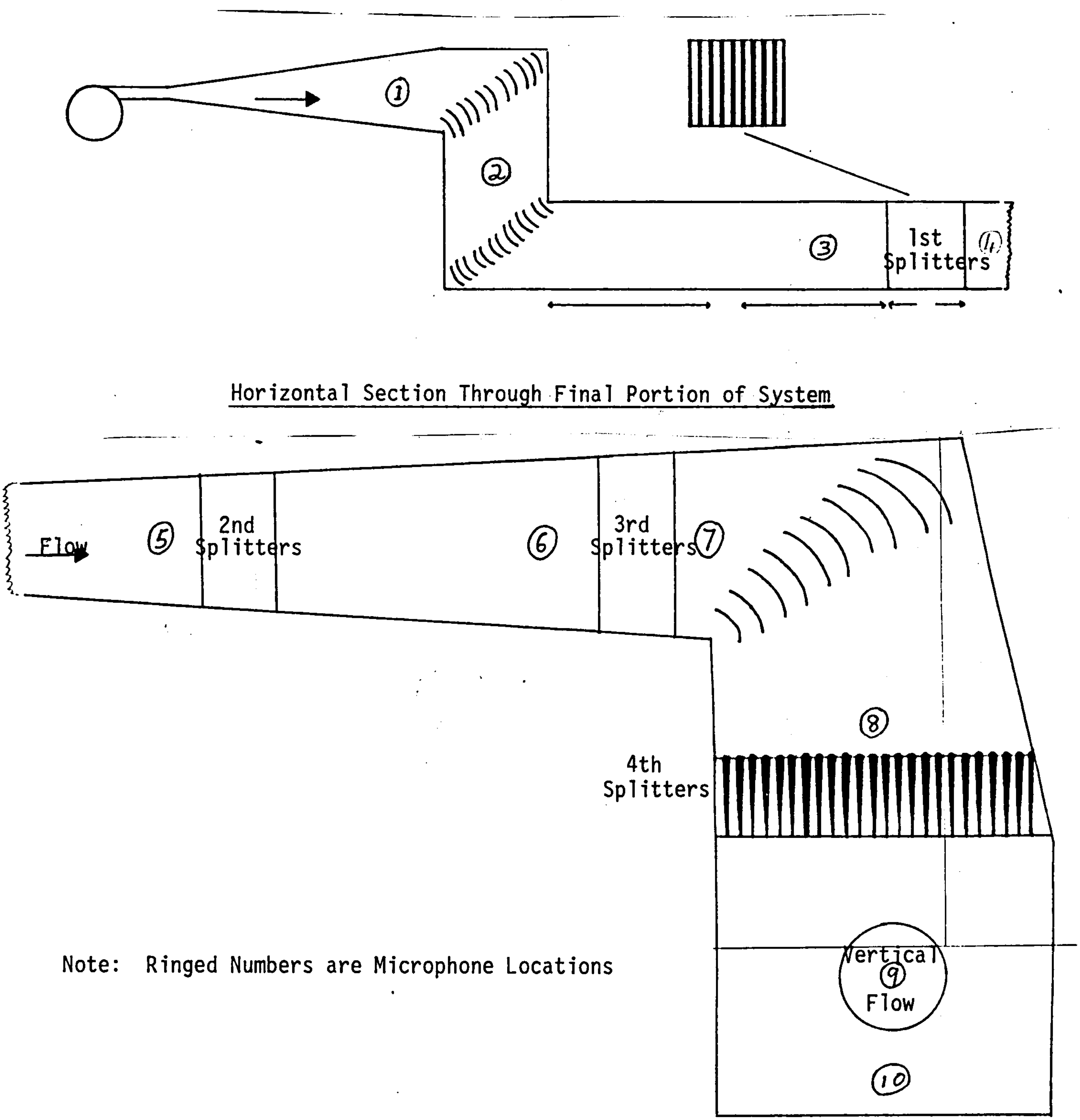
2.3 Measurements in the NASA Aircraft Nosie Reduction Laboratory Low Pressure Air System Muffler

It is informative to evaluate the design goals for the LFTRA muffler together with the manufacturers data presented in the previous section, in the perspective of an existing NASA facility. This facility, the Aircraft Noise Reduction Laboratory (ANRL) Low Pressure air system (Figure 4) is in many respects similar to the blower-duct-muffler system for the proposed LFTRA.

Measurements which were taken on this system specifically for providing background information on LFTRA design, are described in this section and tend to support the conclusion that special attention is required for muffling low frequencies.

Vertical Section Through Initial Portion of System

Horizontal Section Through Final Portion of System



Note: Ringed Numbers are Microphone Locations

Figure 4. ANRL Low Pressure Air System

The ANRL system is powered by a 250 H.P. induction motor running at 1180 RPM through a variable speed drive to a centrifugal blower. Internal diameter of the blower wheel is 3 feet and exterior diameter is 5½ feet. This is about half the size of the proposed LFTRA blower. Connected to the blower exhaust is a diffuser followed by two sets of turning vanes which direct the flow into an underground duct. This duct further diffuses the flow and incorporates four sets of sound attenuating acoustic splitters as shown in Figure 4. Narrow band acoustic spectra measured with the blower operating at 970 RPM are shown in Figure 5. Computed fan blade passage frequency under these conditions is 194 Hz. A measured peak in the acoustic spectra in Figure 5 may be observed at this frequency. Estimated volume flow was in the range of 80,000 CFM.

The design of this system was performed in the early 1970's and represented a state-of-the-art attempt to produce a quiet flow for conduction of aeroacoustic research. In order to accomplish the design goals for sound attenuation without the generation of a significant amount of turbulent flow self noise, the flow is diffused into a large cross-section (8' x 9') before passing the first group of acoustic splitters. This and all subsequent sections of splitters employ bulk acoustic absorbent material behind a perforated face sheet. Splitter width averages six inches and is approximately equal to the splitter separation. Flow velocity approaching the first group of splitter is about 20 feet/sec. Flow velocity between the splitters is about double this value. The duct continues to diverge so that the velocity continues to decrease until by the fourth set of splitters it has decreased to 1/3 of the above values.

Figure 6 shows the measured performance of the muffler with each succeeding stage. Four plots of attenuation versus frequency are given representing the difference between microphone three located before the first set of splitters and four additional microphones following each successive set of acoustic splitters. Detailed interpretation of these plots is not obvious and demands a more thorough effort in data reduction and analysis than was possible in this study. In the absence of flow one would expect four curves equally spaced at each frequency representing approximately similar attenuation from each set of identical splitters. However, due to the presence of flow, sound is created by turbulence and unsteady aerodynamics throughout the length of the duct. Thus the splitters, turning vanes and flow straighteners in the plenum area (microphone 10) act as noise sources.

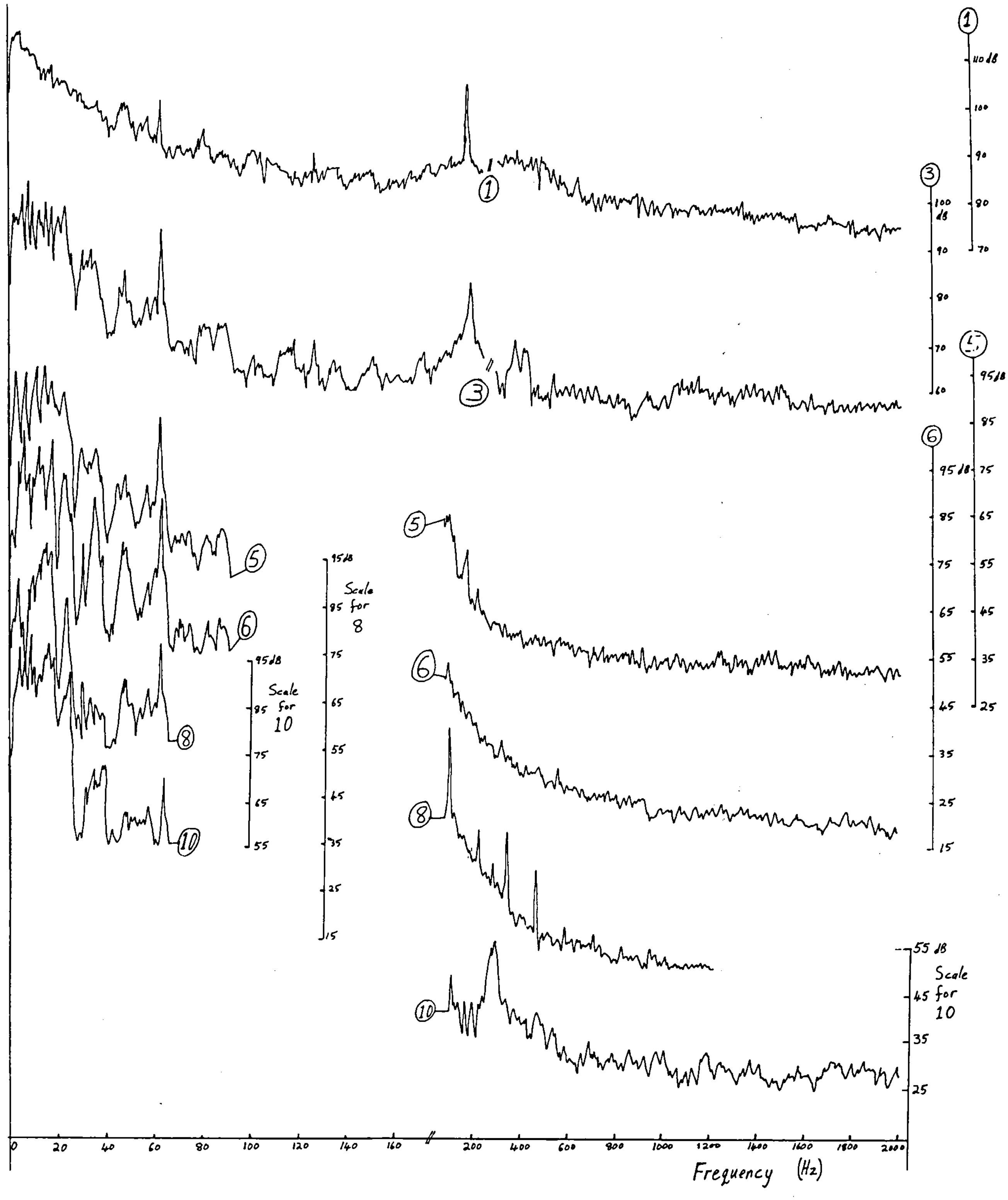


Figure 5. Acoustic Spectra Measured in the ANRL Low Pressure Air System

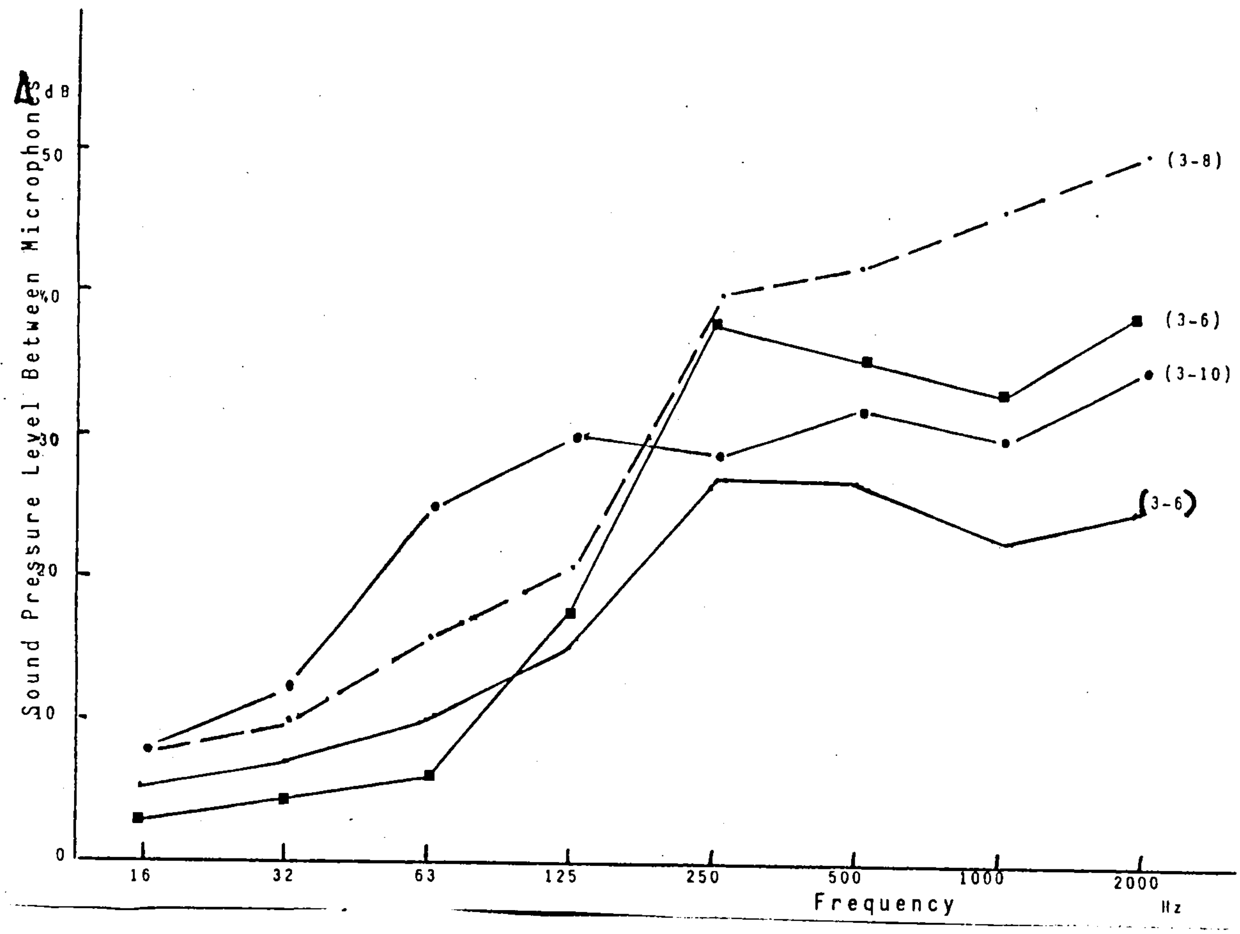


Figure 6. Octave Band Levels Between Microphones in ANRL Low Pressure Air System at Approximately 80,000 CFM

Other difficulties in interpretation of the measurements include limited dynamic range of measurement equipment (which is the reason for the split plots in Figure 5) and high levels of low frequency noise which propagate unattenuated throughout the system causing standing waves down the length of the duct.

In general, however, the following observations appear well founded:

- The splitters are highly effective in the range 250-2000 Hz giving approximately 20-25 dB attenuation per set of splitters throughout this frequency range.
- Below 30 Hz they are almost completely ineffective and noise levels in the plenum are within 10 dB of those at the beginning of the muffler.
- Between 30 Hz and 250 Hz attenuation of each splitter set increases as some function of frequency. In this frequency range, flow noise and the presence of strong reflections make assessment of attenuation difficult.
- Between 250 and 2000 Hz flow noise in the plenum chamber (microphone 10) has increased substantially due to the presence of the nozzle, resulting in a negative attenuation of 10 to 15 dB across the final splitter set.

2.4 Duct Liners

Mathematical modeling techniques such as those described in References 6 through 8 and in Section 2.1 of this report enable the calculation of optimum acoustic duct liner impedance for maximum sound attenuation. However, the process of converting these theoretical values into a practical design is limited by the physical limitations of available materials and fabrication techniques.

In general, all duct liners fall into two categories:

(a) Resonant cavities faced with perforated sheet metal
(b) Bulk absorbant material such as fiberglass, Kevlar, or Feltmetal possibly also faced with a perforated metal sheet.

The impedance of resonant cavities is generally modeled by the relation

$$Z = \theta + i \operatorname{Cot} (kd)$$

where, Z = Acoustic impedance

θ = Flow resistance of the porous face sheet

d = Cavity depth

k = Wavenumber = $\frac{2\pi f}{c}$, f = frequency, c = speed of sound

while numerous models for the point impedance of bulk liners exist (e.g. reference 10, 11).

Some investigators[12,13] have argued that bulk liners should not be modeled as a point impedance and that axial sound propagation within the liner should be accounted for. Modeling of this effect is considerably more complicated however.

With modern fabrication techniques of drilling many closely spaced small holes, almost any face sheet porosity is attainable. Also cavity depth may readily be varied so that at a given frequency, a resonant cavity liner may be made to assume practically any theoretical value.

However, the response of this liner at other frequencies is governed by the above relation which is strongly frequency dependent. This has the result that resonant cavity liners are usually effective over only a very narrow frequency band.

In contrast, with bulk liners it is seldom possible to achieve the theoretically optimum impedance for a given frequency. Thus their attenuation per unit lingth at that frequency is usually inferior to resonant cavity liners. However, bulk liners are extremely effective over broad frequency ranges since in this case they should be compared with not one, but a series of cavity liner sections each tuned to a different frequency.

More discussion on the relative merits of bulk on resonant cavity liners follows in Section 3.0.

2.5 Duct Geometry for Maximum Sound Attenuation

In designing a muffler there are many subtleties and complicated effects that may be used to produce sound attenuation. One parameter, however, dominates all others in determining the maximum attenuation which may be obtained in a lined duct. This parameter is the "kb" value, where $k = \frac{2\pi}{\lambda}$ (λ = acoustic wavelength) and b = distance between the closest opposing duct walls in a rectangular duct (or b = diameter in a circular duct). Figure 7 shows a plot of optimum acoustic transmission loss versus kb taken from reference 14 for a circular duct. Below kb values of about 10 substantial attenuations are possible, while above this value attenuations are substantially reduced.

From Figure 7 we may thus deduce that for good attenuation a high frequencies, wall separation (b) should be small. The usual way to obtain small wall separation together with the low speed diffused flow necessary for low flow self noise is by installing splitters (Figure 8) lined with material of appropriate acoustic impedance for maximum attenuation.

2.6 Flow Noise

The generation of sound by turbulent flow in a duct is a field too complicated and diverse to be addressed in detail in this report. Not only is the field at present incompletely understood, but no readily available information has been found in the literature to enable any more than order of magnitude estimation of the effects.

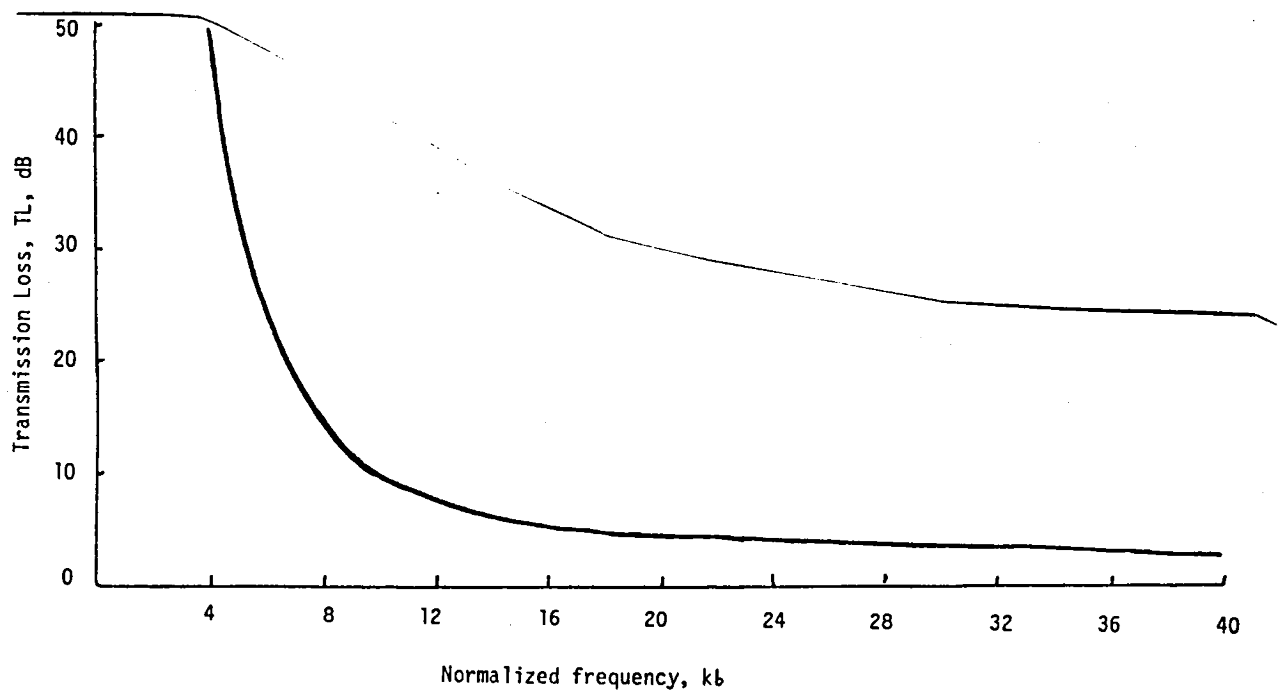


Figure 7. Optimal Acoustic Transmission Loss of a Uniform Circular Liner for a Plane Wave Source as a Function of Normalized Frequency

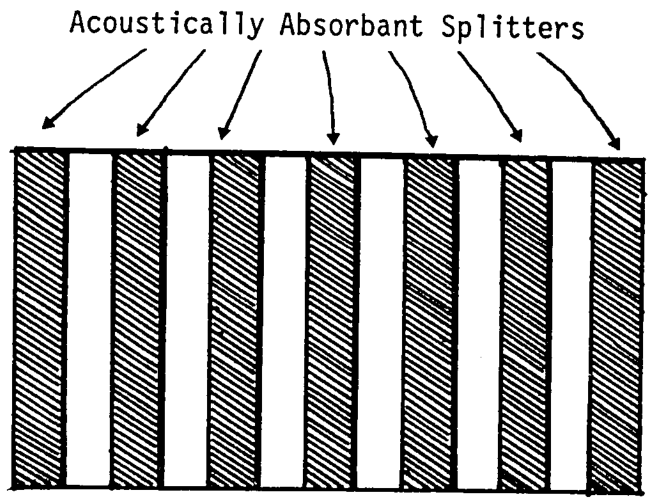


Figure 8. Duct With Splitters

Clearly, however, minimization of flow noise (or self noise) is important to satisfactory muffler design. That this objective is difficult to achieve is illustrated by the acoustic measurements,presented in Section 2.3, of the ANRL low pressure air system.

One of the earliest workers to discuss aerodynamic noise radiated from turbulent boundary layers was Powell[15,16] who demonstrated an enhancement in quadrupole radiation of sound from a turbulent boundary layer due to the presence of a rigid wall. Ffowcs Williams[17] showed that if the wall was allowed to be flexible no more inherently efficient sound radiation mechanism was introduced and that the predominant effect of the wall on the sound field was that of a reflector.

More recently, Tsui and Flander[18] measured sound generation by flow over a perforated liner plate while Howe[19] considered a similar problem theoretically in the presence of suction through the liner.

Nelson and Morfey[20] measured sound generation from a flat place placed perpendicular to the free stream in ducted flow to simulate noise from a splitter. They also established scaling laws for the sound radiated by the resulting separated flow. Their results indicate that prior to the cut-on of higher order duct modes, the sound power generated scales according to the fourth power of flow velocity. After the cut-on of higher order duct modes this scaling transitions abruptly to the sixth power of the flow velocity. Directly associated with radiated sound pressure is the "psuedo-sound" of the pressure fluctuations on the boundary wall. Numerous sets of measurements of wall pressure fluctuations have been made, notably by Willmarth[21,22,23] and Maestrello[24,25].

A problem incidental to this topic is the measurement of sound radiated by flow inside a duct. Insertion of a microphone with a nose cone into the flow creates its own noise due to flow around the microphone which is difficult to distinguish from the noise already present. For example, Figure 9 gives a set of curves for precisely this situation in quiet flow taken from the B&K Microphone Handbook. On the other hand, if the microphone is placed flush with the wall, then the psuedo-sound pressures from the turbulent boundary layer unavoidably contaminate the radiated sound. Only when the boundary layer is laminar are accurate acoustic measurements attainable.

An example of this confusion may be shown by applying Willmarth's empirical relation[21]

$$\frac{p'}{q_o} = 0.006$$

where, p' = overall r.m.s. boundary layer pressure fluctuation

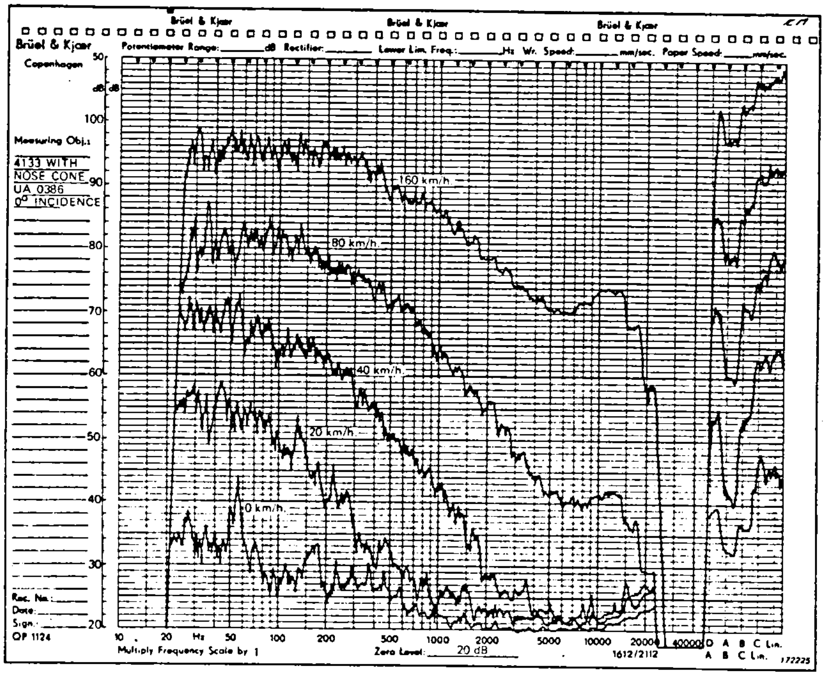
q_o = free stream dynamic pressure,

to the flow velocities given in Figure 9 with the result shown in Table 1.

Table 1

Flow Velocity kM/Hr.	RMS Wall Pressure Fluctuation from Willmarth's Relation (dB)	Measured RMS Sound Pressure Levels With a Nose Cone (dB)	ΔdB $=dB_W-dB_{F.S.}$
20	75	63	12
40	87	78	9
80	99	92	7
160	111	107	4

From the results presented in Table 1, it may be seen that psuedo-sound levels are substantially higher than free stream levels at low flow velocities where the sound generation mechanism is inefficient.



Induced noise levels as a function of windspeed and frequency. 0° incidence microphone 4133 with Nose Cone UA 0386

Figure 9. Induced Noise Levels as a Function of Windspeed and Frequency from a Microphone with a Nose-Cone

3.0 MUFFLER DESIGN

A set of octave band spectrum levels from a possible anticipated fan noise source is given in Section 2.3. It may be seen from perusal of these values that broad band sound attenuations will be required from 31 Hz through 4000 Hz.

Based on the discussion of duct liner performance (Section 2.4), the obvious choice for broad band attenuation is a bulk liner. Cummings[12] gives a set of theoretically based design charts for mufflers using bulk liners. These design charts which have also been verified experimentally, are reproduced here in Figures 10 and 11. In these plots "b" is the wall separation, "σ" is the flow resistance of the material in MKS Rayles and the "Space Factor" is the ratio of (airway cross-sectional area/total duct cross-sectional area).

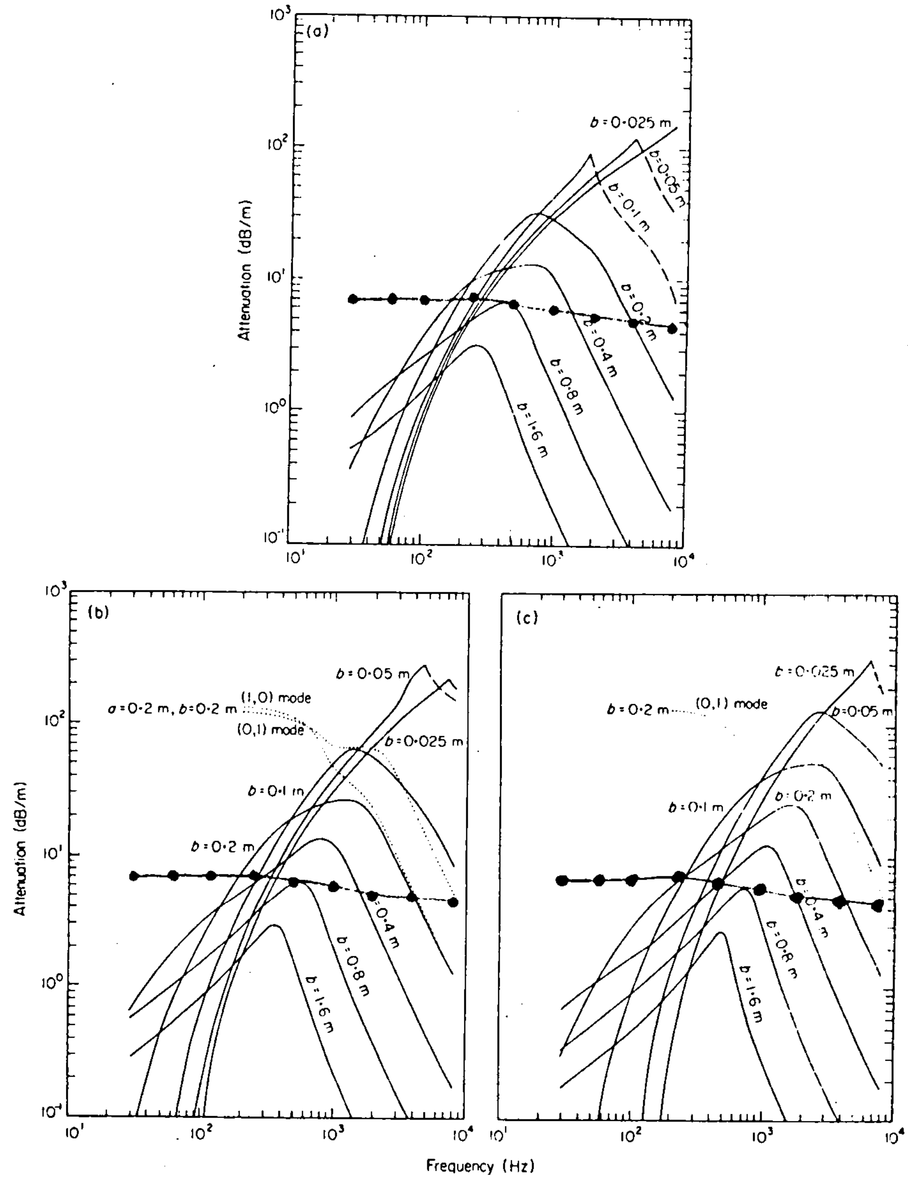
It is convenient to use these design charts of Cummings to evaluate in approximate terms what characteristics a LFTRA muffler using bulk absorbers would possess. Suppose we proceed in the following manner by selecting a maximum muffler length of say 10 meters. Then the required attenuation in dB/m to obtain a 60 dB noise level follows:

Frequency (Hz)	31	63	125	250	500	1000	2000	4000	8000
Fan Noise Levels (dB)	130	132	131	133	124	120	115	110	106
Required Attenuation (dB/m)	7	7.2	7.1	7.3	6.4	6	5.2	5	4.6

Plotting this required attenuation on the design charts (Figures 10 and 11) the following conclusions may be drawn:

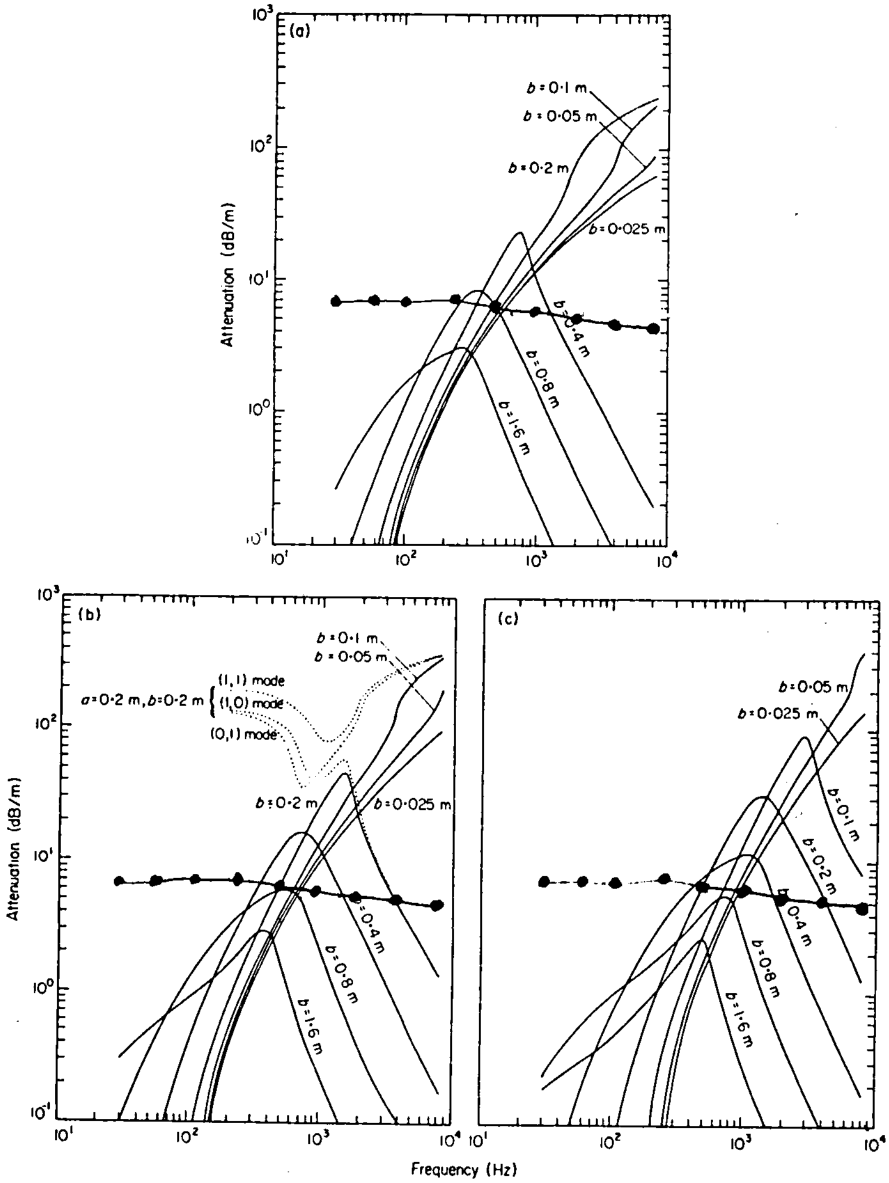
- At and below the 125 Hz octave band insufficient attenuation is attainable. This is true for the three values of flow resistance presented by Cummings for all channel widths and space factors.
- From 250 Hz upward, appropriate attenuation is attainable but only at channel widths in the range .1m < b < .4m. In this frequency band it is likely that some combination of channel widths and space factors will yield the best attenuation although widest frequency coverage at satisfactory attenuation levels is obtained at b = .2m.

It is interesting in the light of these observations to review some previous facts. The attenuation of the commercial muffler which utilizes bulk absorbers



Attenuation of the least damped mode in ducts with a 50% space factor. (a) $\sigma = 10^4$ MKS Rayls; (b) $\sigma = 2 \times 10^4$ MKS Rayls; (c) $\sigma = 4 \times 10^4$ MKS Rayls.

Figure 10. Muffler Design Charts for a Bulk Liner, 50% Space Factor



Attenuation of the least damped mode in ducts with a 75% space factor. (a) $\sigma = 10^4$ MKS Rayls; (b) $\sigma = 2 \times 10^4$ MKS Rayls; (c) $\sigma = 4 \times 10^4$ MKS Rayls.

Figure 11. Muffler Design Charts for a Bulk Liner, 75% Space Factor

discussed in Section 2.2 is also deficient in the low frequency region at and below 125 Hz. So also, is the attenuation of the muffler system for theANRL low pressure air system discussed in Section 2.3. Another interesting observation is that the acoustically treated splitter system of the ANRL muffler has a wall separation of the order of .16m. This compares well with the apparent best choice of splitter spacing (.2m) derived from Cummings' design charts.

Thus it may be concluded that a muffler employing standard commercial bulk absorbant material in a duct width of the order of .2m is the best choice over the frequency range 250-8000 Hz for the LFTRA. Below this frequency range standard commercial bulk absorbers do not appear adequate. Part of the reason for inadequate absorption of bulk liners at low frequencies may be deduced from the physics of the process. Here the entire bulk matrix moves with the passage of the sound wave reducing absorption due to micro-turbulent flow over individual bulk matrix fibers.

To study the effects of a resonant cavity liner for use in this frequency range, the model described in Section 2.1 was programmed for execution in an optimization loop using the Stewart-Davidson-Fletcher-Powell (SDFP) algorithm for non-linear optimization. The procedure was set up so that a duct liner could be optimized either at a single frequency or over a range of frequencies (Figure 12). Thus, it was possible to vary the Q-factor or selectivity of the attenuation band.

This program was exercised on numerous duct configurations of which only two are presented here. Figure 13 represents the attenuation of a lined duct consisting of nine separate tuned sections each two meters long. The duct, which is lined on walls in the Y-Z plane only, has a cross-section of Δx = .7m by Δy = 1.3m at the high frequency end and diverges at an angle of 2^o in the X-Z plane and 4^o in the Y-Z plane to a section of Δx = 2m by ΔY = 4m at the low frequency end. This duct was designed with diverging walls so that it might perform double duty in the LFTRA system as a diffuser as well as a muffler.

By contrast a uniform duct section of Δx = .7m gives substantially better sound attenuation per unit length as shown in Figure 14. This design consists of nine muffler sections each one meter in length. It may be seen that the overall attenuation curve is similar to that of Figure 13 although the

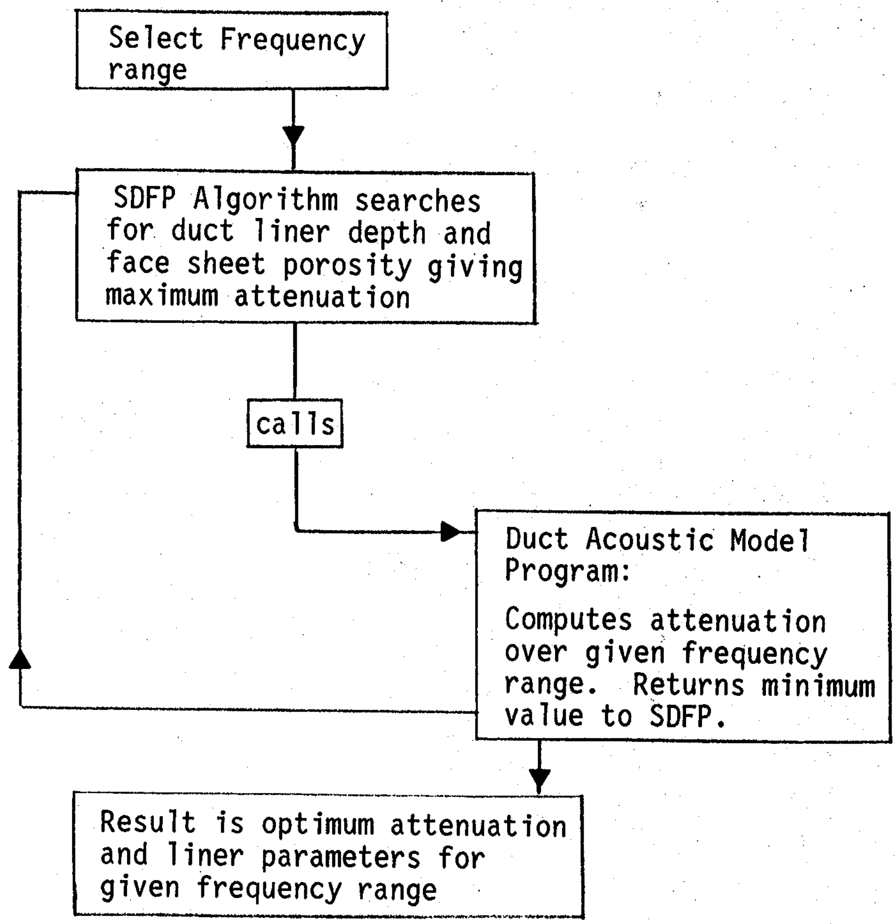


Figure 12. Design Procedure for a Single Muffler Section

Diffusing Angles: X-Z plane = 2^o, Y-Z plane = 4^o

Frequency (Hz)	20	26	29	39	52	69	92	123	165
Cavity Depth (M)	4.081	3.136	2.781	2.073	1.548	1.151	.843	.597	.431
Non-Dim. Flow Resistance	.06	.07	.08	.06	.04	.05	.06	.11	.21

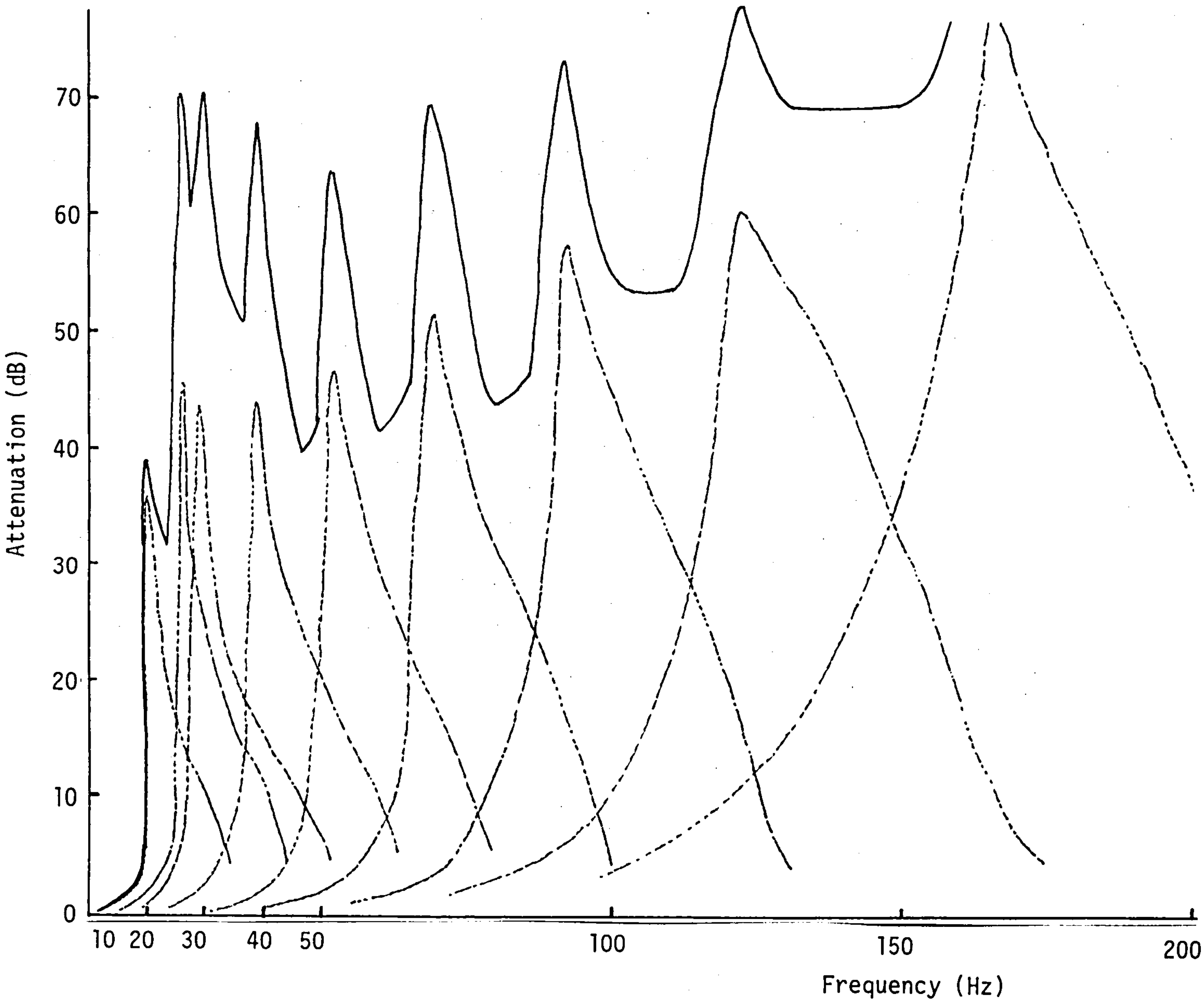


Figure 13. Performance of a Low Frequency Diffusing Muffler (Length 18m)

Diffusing Angles: X-Z plane = 0^{o}, Y-Z plane = 4^{o}

Frequency (Hz)	16	22	29	39	52	69	92	123	165
Cavity Depth (M)	5.3	3.8	2.9	2.1	1.6	1.2	.86	.64	.45
Non-Dim. Flow Resistance	.04	.04	.05	.05	.07	.07	.08	.05	.12

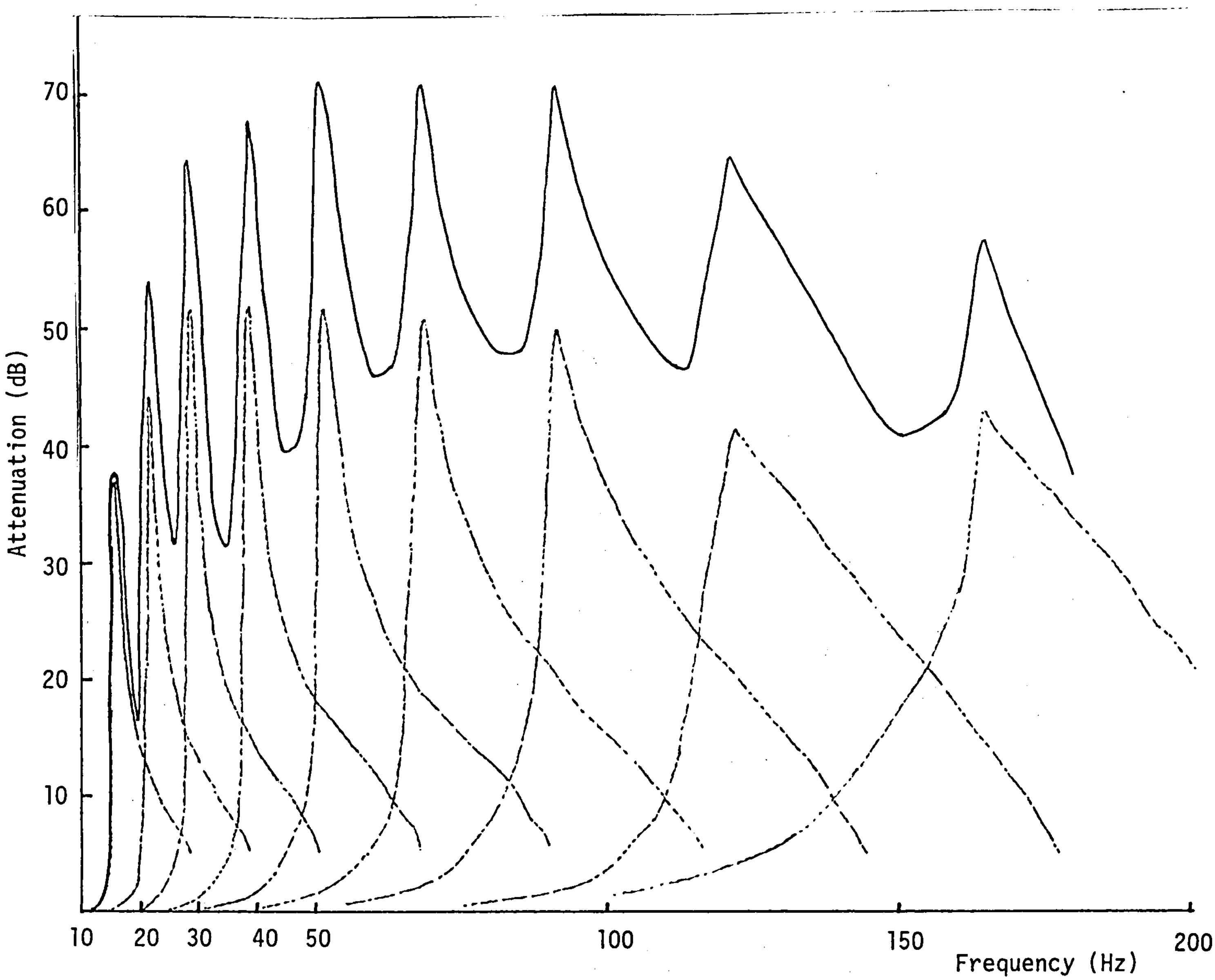


Figure 14. Performance of a Low Frequency Non-Diffusing Muffler (Length 9m)

total muffler length has been halved. Note that the muffler design may be allowed to diverge in the y-direction without affecting its performance.

Flow resistance of liner face sheets non-dimensionalized by ρc together with cavity depths in meters are also given in Figures 13 and 14 for each liner section. It may be seen that optional flow resistances lie in the range .04 to .2 while cavity depths are roughly equal to $\lambda/4$ and vary from approximately 5m at 16 Hz to .5m at 160 Hz.

In general, however, it may be seen from Figures 13 and 14 that attenuation in excess of 40 dB is attainable throughout the frequency range from 16-200 Hz using a series of tuned cavity duct liners. Thus a muffler of this type may be used to provide supplemental attenuation to a commercially available bulk absorbant type muffler for the low frequency regime.

Some notes related to construction of a tuned cavity liner are desireable in order to ensure that theoretical attenuations are achieved in practice.

1) It is important that cavity cross-section does not become too large with the possibility of exciting higher order modes in the cavity thus, it is suggested that cavity diameter not exceed $\lambda/10$.

2) Stiffness of cavity walls should be high enough to prevent significant structural transmission of sound. That is, first cylinder structural resonance should be above acoustic tuning frequency.

3) Cylinders should be isolated from each other by damping material to prevent structural transmission. Also, each tuned muffler section should be isolated from the succeeding one by a vibration break.

4) Mounting and flexibility of the perforated face sheet should be such as to ensure minimal movement under acoustic and aerodynamic loading Some stiffening of the face sheet will probably be required for all muffler sections to meet this requirement.

4.0 LFTRA SYSTEM DESIGN

4.1 Description of Potential Problems

At the beginning of this study, primary concerns regarding noise in the LFTRA test section were directed at muffling the anticipated high noise levels from the fan. However, as the study progress it became increasingly clear that interaction between the system components had a large input on the performance of these individual components.

An important parameter in this interaction is flow noise. For example, in design of the muffler, flow noise considerations mandate low flow velocities because as discussed in Section 2.6, noise radiated by separated and turbulent flow in a duct scales with the fourth or sixth power of flow velocity. For a bulk absorbant muffler designed to attenuate over the frequency range 250 Hz to 8000 Hz Section 3.0 showed that wall separation should be of the order of .2m with a length of perhaps 10m.

Unquestionably over this distance in such a narrow channel, flow would become fully turbulent and if the velocity were high, would radiate substantial acoustic power. Some of this noise would be absorbed by the muffler but some would also not. Thus, the logical muffler configuration for good absorption and low self noise would consist of a large cross-section for low velocity but divided by closely spaced parallel splitters, lined with bulk absorbant material and covered with a porous face sheet for good sound attenuation. Such a design is similar to the muffler in the ANRL low pressure air system.

Purely from the aspect of muffler design there are no problems with this concept and it may be expected to satisfactorily provide the 60 dB design levels. A problem arises, however, when this muffler is integrated into an LFTRA system via a diffuser (Figure 15). To reduce the velocity between the splitters to less than (say) 10m/sec. given a space factor for the splitters of 50% it would be necessary to diffuse from the test section of the LFTRA to a section approximately four meters square. Assuming a 3^{0} maximum diffusing angle on each wall, this results in a diffuser approximately 30 meters long, given a test section of .7m x 1.2m.

While it must be emphasized that these calculations are of an approximate nature, they nevertheless serve to illustrate the problem. Careful design

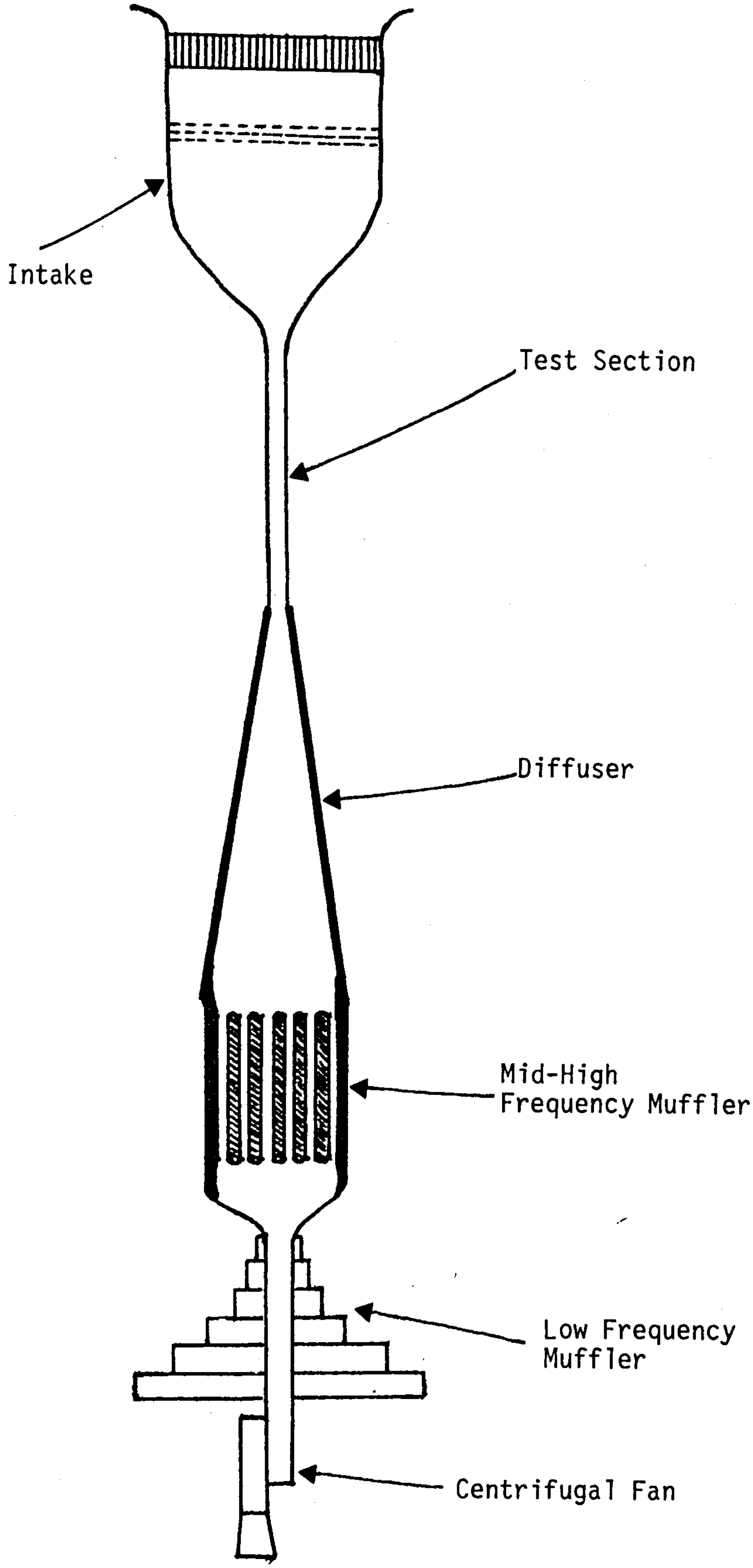


Figure 15. LFTRA System Configuration
(Example 1 - Unsuitable)

optimization should be able to reduce the diffuser length by a factor of two through use of tapered splitters and other refinements.

The principal difficulty with any diffuser, however, is that they are notoriously noisy. This is especially true for a long diffuser where a turbulent boundary layer may grow to substantial thickness and radiate noise back upstream into the test section. If the diffuser is shortened by allowing diffusing angles greater than 3^{o}, then separated flow will result with the generation of still more noise, in addition to decreasing the overall system efficiency.

Solely for system energy efficiency, a diffuser following the test section is desireable since it helps to reduce fan size and operating costs. However, the noise problem is difficult to resolve.

During initial conceptual design of the system an integration of the low frequency muffler with the diffuser (Figure 16) was seen as a possibility. Some benefit may be achieved from this concept because some sound generated by the turbulent boundary layer would be absorbed by the muffler in propatating towards the test section. However, calculations in Section 3 (Figures 13 and 14) show that a diffusing low frequency muffler would be twice as long as a non-diffusing one for the same sound attenuation. Also because each muffler section is tuned only to a specific narrow frequency band, the benefits desireable in the form of sound attenuation of boundary layer noise would be negligible, since boundary layer noise is radiated along the entire length of the diffuser.

From the preceeding discussion, it may be seen that integration of the acoustic muffler into the LFTRA is not a straightforward process. Both system configurations shown in Figures 15 and 16 are unsuitable even though both muffler arrangements provide adequate acoustic attenuation of fan noise. In addition, the muffler arrangement shown in Figure 15 has extremely low self-noise characteristics.

In evaluating the LFTRA system concepts shown in Figures 15 and 16, no consideration has yet been given to cost since both configurations are unsuitable on technical grounds. It is important to note, however, that the low frequency

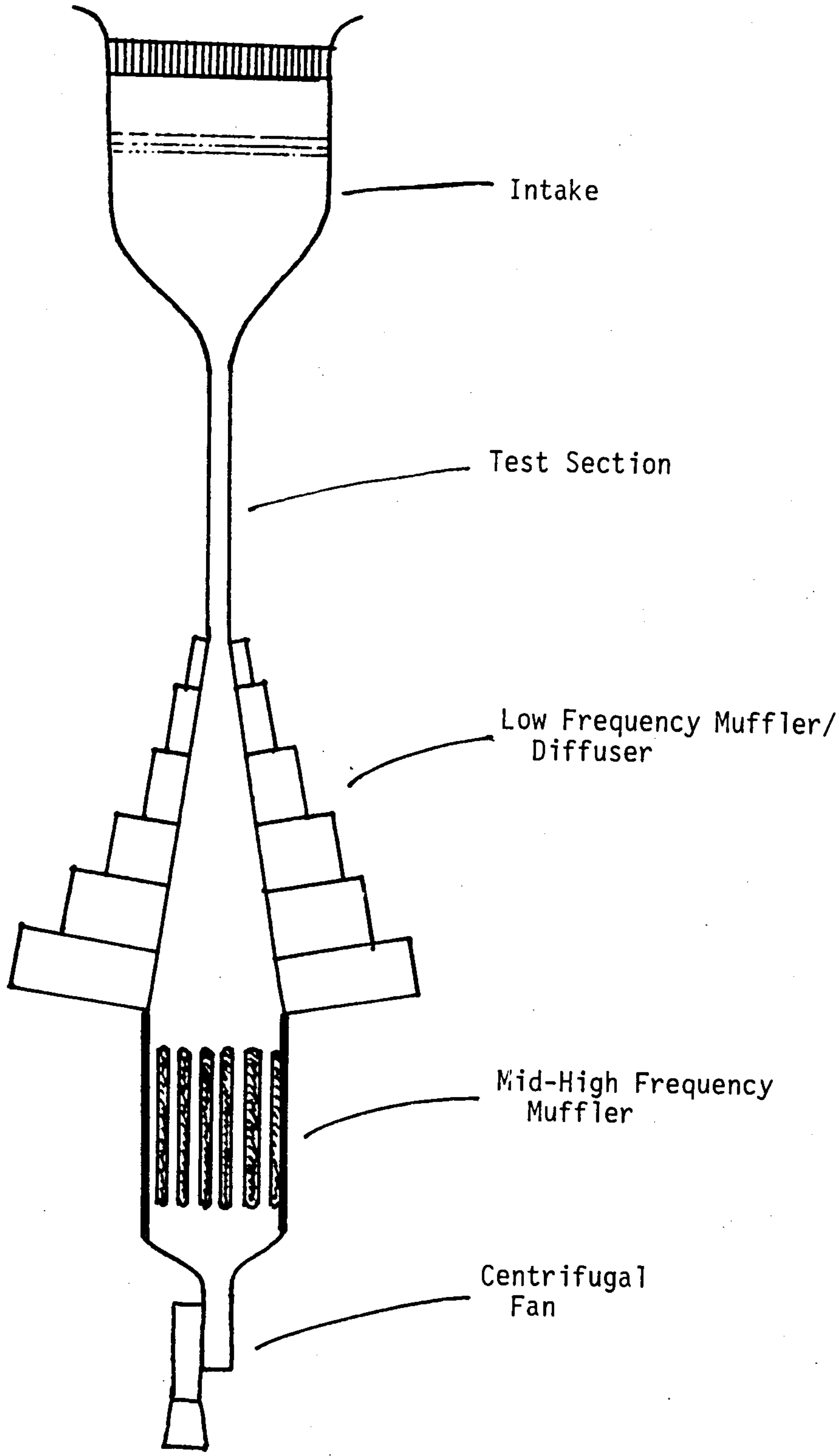


Figure 16. LFTRA System Configuration
(Example 2 - Unsuitable)

tuned cavity muffler is likely to be expensive to construct. On this basis alone, the configuration of Figure 15 is preferred to that of Figure 16 where the low frequency muffler length is doubled due to the greater wall separation.

4.2 Possible Solution

To solve the problems in LFTRA system design outlined in the previous section, it appears essential to introduce sound absorbing material into the diffuser.

One of the principal differences between sound generated in a duct by a fan and sound generated in a duct by a turbulent boundary layer is the location of the sound source. Mariano[26] has demonstrated that substantially improved attenuation may be obtained in a duct if sound sources are located close to a duct wall. Thus, the prospect of obtaining good attenuation in a diffuser without the aid of splitters appears less daunting. Mariano's analysis, which also includes the effects of a line sound source located in a boundary shear layer flow gives the results shown in Figure 17 for sound propagation in the opposite direction to the mean flow. This is, of course, the appropriate relative direction for sound propagation towards the test section in the LFTRA diffuser.

Thus, it would appear that significant benefits in reducing diffuser noise may be achieved by lining the diffuser with a bulk acoustic absorber faced with a perforated sheet. This additional attenuation is a non-linear effect and is at present incompletely understood. It has been discussed, however, by several researchers including Howe[19], Dean and Tester[27], and Ingard and Ising[28].

This concept for a quiet diffuser may be further augmented by suction through the bulk liner and perforated sheet along the entire length of the diffuser. Suction has not only the benefit of inhibiting growth of a turbulent boundary layer, but also may further assist attenuation of sound.

The best solution to integrated system design of the LFTRA, however, may be to attempt to integrate the low frequency muffler, mid-high frequency muffler, and diffuser together in a single unit.

This might be accomplished as shown schematically in Figure 18. Here the diffuser is divided essentially in two parts by a long central wedgelike splitter

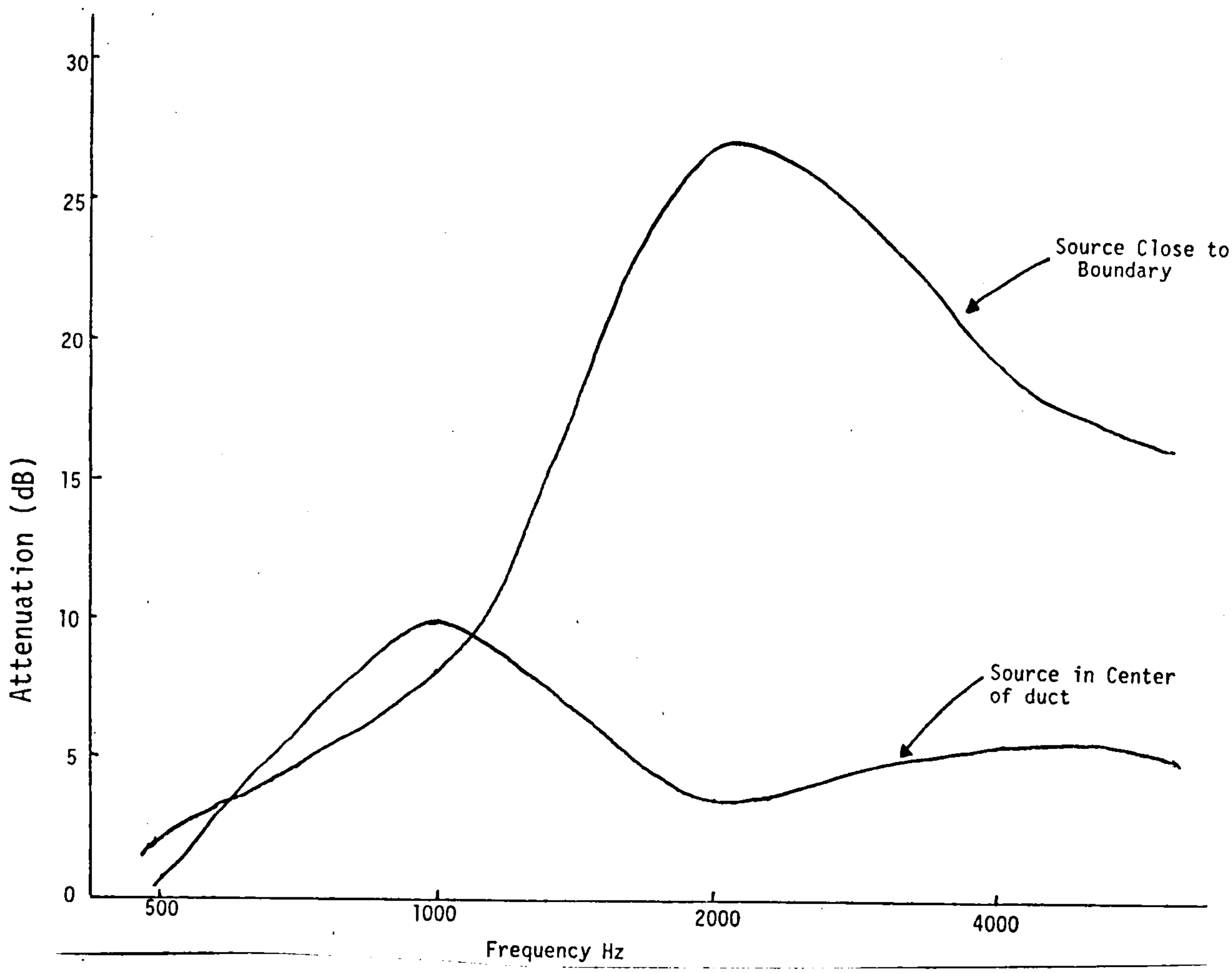


Figure 17. Comparison of Attenuation from a Duct Liner as a Function of Source Position

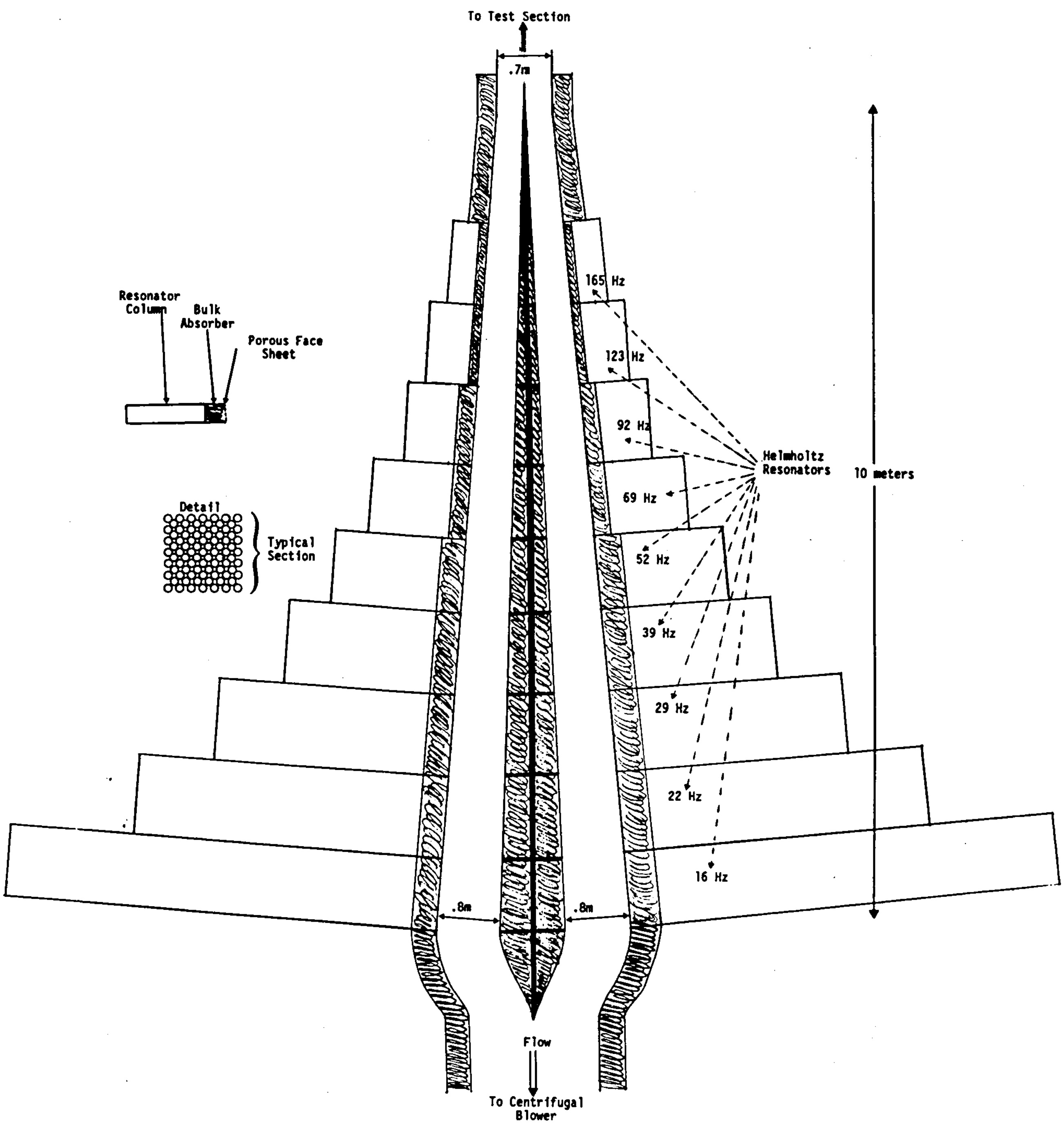


Figure 18. LFTRA System Design Concept Incorporating Diffuser, Low-Frequency Muffler and Mid-High Frequency Muffler

consisting of a rigid central wall, lined on each side with bulk absorbant material. The bulk material is partitioned at numerous axial stations to prevent axial transmission of low frequency sound, and faced with a perforated sheet. This central absorbant wedge would provide attenuation over the mid-high frequency region.

The outer walls of the diffuser comprise the low frequency muffler which is modified by the insertion of a small amount of bulk absorbant material in each cavity. The function of this absorbant material would be to broaden attenuation at the resonant peak while also allowing the low frequency reasonant cavity muffler to double as a bulk absorbing muffler for the mid-high frequency region. Estimated performance of this muffler is shown in Figure 19.

The advantages of this concept are as follows:

- Diffuser (turbulent boundary layer) noise is absorbed in the same muffler section as fan noise.
- Minimum length (and cost) is achieved by integrating diffuser, low frequency muffler and mid-high frequency muffler.

While this concept is well founded when judged according to state-of-the-art knowledge in muffler design, no muffler of this type has ever been constructed. Thus, it is recommended that an experimental evaluation be performed prior to detailed design and construction.

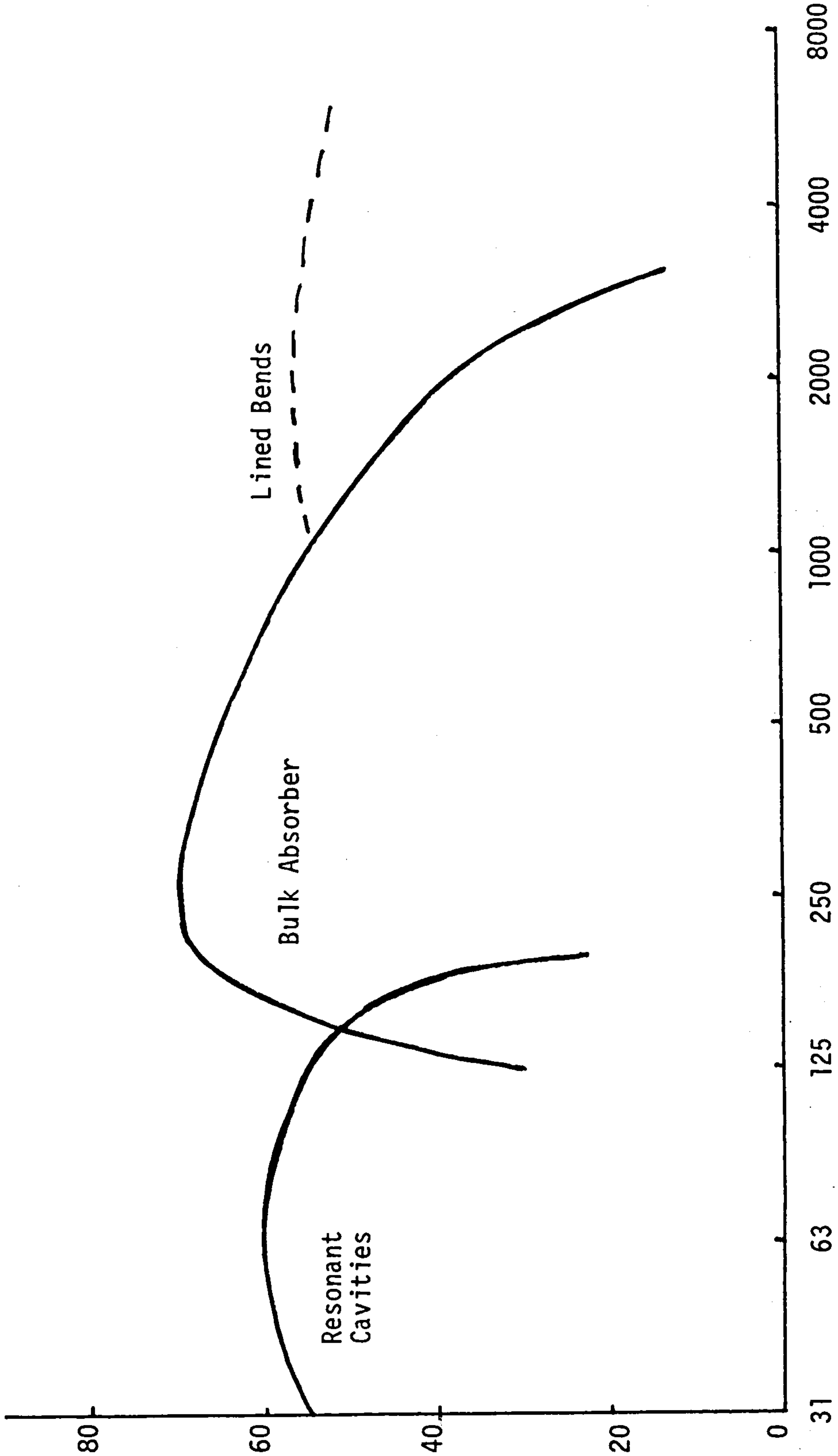


Figure 19. Estimated Performance of LFTRA Integrated Muffler shown in Figure 18.

5.0 CONCLUSIONS

This report has discussed the design of a muffler to attenuate fan noise in the proposed LFTRA. During the study it became evident that integration of the muffler into the LFTRA total system design radically impacted the characteristics of the required muffler.

The most promising concept which has emerged at this stage consists of an integrated low frequency muffler, mid-high frequency muffler and diffuser. Although the design goals are extremely challenging this design concept appears well founded when judged according to state-of-the-art knowledge in muffler design.

Significant unknown factors exist, however, which indicate that further study and experimental evaluation are required. Specific recommended areas of investigation are as follows:

- Low frequency absorption characteristics of resonant liners.
- Self noise of flow over perforates.
- Influence of bulk absorber inside resonant cavity liners.
- Effect of wedge in causing possible standing waves in tunnel test section.

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Appendix 1

Exact Solution of the Helmholtz Equation in a Rectangular Duct by Separation of Variables

In the absence of mean flow, acoustic waves propagating in a duct are governed by the three dimensional Helmholtz equation,

$$\nabla^2 p + k^2 p = 0 \qquad k = \frac{\omega}{c} \tag{1}$$

Suppose a harmonic solution in z of the form

$$p(x,y,z) = \sum_{m=1}^{\infty} \sum_{m=1}^{\infty} (A_{mn} e^{iK_{mn}z} + B_{mn} e^{-iK_{mn}z}) \; X_m(x) \; Y_n(Y) \tag{2}$$

Where X_m must satisfy the boundary conditions

$$\frac{\partial X_m}{\partial x} - ik\beta_4 X_m = 0 \text{ at } x = h \tag{3}$$

$$\frac{\partial X_m}{\partial x} + ik\beta_3 X_m = 0 \text{ at } x = 0 \tag{4}$$

and

$$\frac{\partial Y_n}{\partial y} - ik\beta_2 Y_n = 0 \text{ at } y = \ell \tag{5}$$

$$\frac{\partial Y_n}{\partial y} + ik\beta_1 Y_n = 0 \text{ at } y = 0 \tag{6}$$

(wall admittances β are shown in Figure 3)

Suppose $X_m = \operatorname{Cos} \gamma_m x + Q \operatorname{Sin} \gamma_m x$

$$\frac{\partial X_m}{\partial x} = -\gamma_m \operatorname{Sin} \gamma_m x + \gamma_m Q \operatorname{Cos} \gamma_m x$$

(4) $\quad -\gamma_m \sin\gamma_m x + \gamma_m Q \cos\gamma_m x + ik\beta_3 (\cos\gamma_m x + Q \sin\gamma_m x) = 0$

$$(-\gamma_m + ik\beta_3 Q)\sin\gamma_m x + (\gamma_m Q + ik\beta_3)\cos\gamma_m x = 0$$

$$\tan\gamma_m x + \frac{\gamma_m Q + ik\beta_3}{-\gamma_m + ik\beta_3 Q} = 0$$

now when $x = 0$ $\tan\gamma_m x = 0$

$$\Rightarrow \gamma_m Q + ik\beta_3 = 0$$

$$\Rightarrow Q = \frac{-ik\beta_3}{\gamma_m}$$

$$\Rightarrow X_m = \cos\gamma_m x - \frac{ik\beta_3}{\gamma_m}\sin\gamma_m x$$

(3) $\Rightarrow -\gamma_m \sin\gamma_m x - \gamma_m \frac{ik\beta_3}{\gamma_m}\cos\gamma_m x - ik\beta_4(\cos\gamma_m x - \frac{ik\beta_3}{\gamma_m}\sin\gamma_m x) = 0$

$$-\gamma_m^2 \sin\gamma_m x - ik\beta_3\gamma_m\cos\gamma_m x - ik\beta_4\gamma_m\cos\gamma_m x - k^2\beta_3\beta_4 \sin\gamma_m x = 0$$

$$(\gamma_m^2 + k^2\beta_3\beta_4)\sin\gamma_m x + ik(\beta_3 + \beta_4)\gamma_m\cos\gamma_m x = 0$$

Suppose we X_m and Y_r of the form:

$$X_m = \cos\gamma_m x + A\sin\gamma_m x$$

$$Y_n = \cos\alpha_n Y + B\sin\alpha_n Y$$

then (4) $\Rightarrow A = \frac{-ik\beta_3}{\gamma_m}$ and (5) $\Rightarrow B = \frac{-ik\beta_1}{\alpha_n}$

Substituting these expressions in (3) and (5) we get the transcendental equation:

$$(\gamma_m^2 + k^2\beta_3\beta_4)\sin\gamma_m h + ik(\beta_3 + \beta_4)\gamma_m\cos\gamma_m x = 0$$

$$(\alpha_n^2 + k^2\beta_1\beta_2)\sin\alpha_n \ell + ik(\beta_1 + \beta_2)\alpha_n\cos\alpha_n \ell = 0$$

Substituting (2), (7), (8), (9) in (1) gives the eigenvalues

$$K_{mn}^2 = k^2 - \gamma_m^2 - \alpha_n^2$$

The imaginary part of K_{mn} gives the decay rate in the z-direction for the mn'th mode.

Note that $\gamma_m = \alpha_n = 0$ is always a solution to equations (10) and (11) but that this solution gives the additional conditions that

$$k^2\beta_3\beta_4 + ik(\beta_3 + \beta_4) = 0, \text{ and,}$$

$$k^2\beta_1\beta_2 + ik(\beta_1 + \beta_2) = 0$$

Thus for this trivial case K = 0.

A general solution of equations (10) through (12) is only possible numerically, for example, using a Newton Raphson scheme[8]. An alternate procedure and the one used in this study was to use a 1-D finite element formulation of a 2-D representation of equation (1) in the x-z plane together with boundary condition equations (3) and (4) to obtain eigenvalues γ_m. A similar 1-D finite element formulation of a 2-D representation of equation (1) in the y-z plane together with boundary condition equations (5) and (6) gives α_n. The combined eigenvalues K_{mn} are then synthesized as before using equation (12). This procedure was suggested by Watson[9].

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16. Abstract

This report concerns acoustic muffler design of a new research tool for studying laminar flow and the mechanisms of transition, the Laminar Flow and Transition Research Apparatus (LFTRA). Since the presence of acoustic pressure fluctuations is known to affect transition, low background noise levels in the test section of the LFTRA are mandatory. The report discusses the difficulties and trade-offs of various muffler design concepts and indicates the most promising.

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