# 6. PERFORMANCE OF INLET SOUND SUPPRESSORS

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## SUMMARY

The sound power attenuation was experimentally measured for two inlet noise suppressor configurations and compared with the calculated attenuation from a theoretical model. From the agreement observed, it appears that the theoretical model is valid, at least in the range of broadband sound, and can be used in the design of a suppressor. A general description of the theoretical model and the effect of the duct and sound properties on the sound propagation and attenuation are presented.

## INTRODUCTION

Theoretical and experimental studies of inlet noise suppressors are being conducted at the Lewis Research Center as a part of the noise research program. The objective of the studies is to extend the understanding of the acoustic behavior of suppressors and of the factors that influence suppressor performance. The studies are general in scope rather than being applied to a specific engine.

In this paper results from the theory are presented which indicate the effects of several parameters on predicted suppressor performance. Some characteristics of the sound propagation found theoretically in a suppressor are also shown. In addition, a comparison is made between the experimental and theoretical sound power attenuation of two suppressor configurations. A detailed discussion of the theoretical calculation is given in reference 1.



## SYMBOLS

| | |
|---|---|
| $c$ | speed of sound |
| $D$ | duct diameter |
| $L$ | duct length |
| $R$ | wall resistance |

| | |
|---|---|
| r | radial distance |
| X | wall reactance |
| x | axial distance |
| A dB | sound power attenuation |
| A $dB_{max}$ | maximum sound power attenuation |
| X | sound wavelength |
| P | mass density |

## ANALYTICAL MODEL

Some of the conditions and assumptions for the analytical model are shown in figure 1. Briefly, the theory provides the solution to the linearized wave equation valid for acoustically treated or soft-walled circular ducts. It has been assumed that there was no steady flow, there were no wave reflections from the end of the duct, and the wall impedance was uniform throughout the duct. The solution has been specialized to consider a plane wave entering the duct by calculating the amplitudes of the radial modes such that their sum approximates the plane wave. For this purpose, the first 10 radial modes were sufficient to approximate the initial plane traveling pressure wave.

For these conditions the solution gives the pressure and velocity at any point in the duct. It is therefore possible to calculate the energy passing any cross section and, thus, the sound power attenuation. The parameters that appear in the theory are the ratio of duct diameter to sound wavelength $D/\lambda$, the ratio of duct length to diameter $L/D$, and the wall impedance composed of the wall resistance R, and the wall reactance X.

## THEORETICAL RESULTS

In figure 2 is presented the calculated behavior of the initially plane wave as it propagates down the duct. The sound pressure amplitude at any axial distance x relative to that of the plane wave at $x = 0$ is shown as a function of radial position. The calculations were made for a single value of acoustic impedance of the wall and for two values of the frequency parameter $D/\lambda$. For D/X = 10, the initially plane wave distorts as it propagates until a smooth profile exists at $x/D = 5$; also, the sound energy has been redistributed or beamed toward the duct center line. The pressure amplitude at the center line has doubled or increased by 6 dB, whereas the pressure amplitude at the wall has decreased by a factor of 10 or 20 dB. For these conditions, calculations show that the acoustic energy of the wave has been attenuated very little. For $D/\lambda = 1$, the wave remains essentially planar as it propagates but is appreciably attenuated. Thus, the results predict a strong influence of the parameter $D/\lambda$ on the sound power attenuation produced by a suppressor.

Figure 3 shows contours of constant sound power attenuation relative to the duct entrance plotted in the wall impedance plane. These calculations were for constant values of $D/\lambda$ of 1 and $L/D$ of 3. The contours are more or less circular and show an upper limit to the attenuation. In general, the upper limit occurs at a negative value of the wall reactance or in the capacitance (stiffness) controlled region. This result is in contrast to the approximate solution for a plane wave which predicts that the maximum attenuation will occur at zero reactance or at the liner tuned point (ref. 2). Generally, the point of maximum attenuation moves at a negative $45^{\circ}$ slope to higher wall resistance and more negative wall reactance as $D/\lambda$ is increased. This behavior is discussed in reference 1.

In figure 4, the dependence of the point of maximum attenuation on the frequency parameter $D/\lambda$ is shown for several values of $L/D$. The attenuation values have been divided by the $L/D$ value of each curve; this procedure approximately correlated the curves. It is seen from the figure that for $D/\lambda$ values less than about 1, the attenuation does not depend strongly on $D/\lambda$. However, for $D/\lambda$ values greater than 1, the attenuation depends strongly on $D/\lambda$. A possible reason for this behavior is that sound beaming rather than attenuation was found to occur when the duct diameter exceeded the sound wavelength. This behavior was noted in the discussion of figure 2.

Figure 5 shows calculated attenuation curves for two suppressors that differ only in their diameter. The length and wall impedance were the same for each suppressor. As shown by the curves, the attenuation is much greater for the smaller duct diameter. The increase in attenuation is due to the combined change in $L/D$ and $D/\lambda$ associated with the decrease in diameter. The curves illustrate the gain to be made by the use of configurations that have multiple passages such as those obtained with radial struts or splitter rings. The large attenuation predicted for $L/D = 5$ may not be observed in practice because of the masking effect of sound transmission through structural members or background noise such as the jet noise.

## COMPARISON OF EXPERIMENT AND THEORY

A limited number of experiments for comparison with the theory have been performed on the inlet of a full-scale 5-65 turbojet engine. The base-line configuration of the J-65 is shown in figure 6. It was equipped with a bellmouth and hard cowl inlet for static testing. Noise measurements were made by an array of far-field microphones in the front quadrant of the engine. In order to produce a more pronounced discrete tone at the blade passage frequency, the space between each third pair of inlet guide vanes was mechanically blocked. The resulting spectrum showed a peak about 10 dB above the background noise at the blade passage frequency and a lower peak at the first harmonic of blade passage frequency. Details of the experimental apparatus and procedures are given in reference 3.

At present, experimental and theoretical data have been obtained for two suppressor configurations. One was a simple cylindrical suppressor with no centerbody. The other was a seven-strut configuration shown without the bellmouth (fig. 7). In these configurations all the suppressor surfaces exposed to the flow were acoustically treated. The facing material in both suppressors was a perforated plate 0.02 in. thick with 0.05-in-diameter holes and an open area ratio of 0.08. The backing cavity depth was 1 in. except that for the struts which was 1/2 in. This backing cavity was partitioned with a honeycomb structure. The two sides of the struts were separated by an impervious septum. The resistance and reactance of this suppressor wall were calculated as a function of frequency according to the method of reference 4. Other conditions used in the wall impedance calculations were flow-by velocity of 150 ft/sec and a sound pressure level of 150 dB (re $2 \times 10^{-4}$ dyne/cm$^2$) corresponding, respectively, to the flow velocity and the overall pressure level in the hard cowl inlet.

The geometry of the cylindrical suppressor corresponded to that assumed for the theoretical calculations, A comparison of the calculated and experimental attenuations is shown in figure 8; the agreement is very good. It should be pointed out that there is no empiricism, in the form of an undetermined coefficient, needed for the comparison.

The passages in the seven-strut suppressor were not cylindrical so that the calculated attenuation was obtained essentially by using the wetted perimeter of the passages. The attenuation was calculated for a cylinder with a diameter equal to the actual passage height or width. The calculated result was corrected by the ratio of acoustically treated area to passage cross-sectional area for the actual passage as opposed to the circular passage. The $D/\lambda$ effect was thus accounted for by the exact plane wave theory in a cylinder. The noncylindrical correction was suggested by the approximate theory of Morse (ref. 2). The experimental and calculated attenuations for the seven-strut suppressor are shown in figure 9. At the lower frequencies, where the sound was essentially broadband noise, the agreement was again very good. At the higher frequencies, where discrete tones predominate, the experimental attenuation exceeded the calculated attenuation. A possible reason for this discrepancy may be that the sound at these discrete frequencies propagates as a rotating mode. According to Morse (ref. 2) rotating modes should attenuate faster than a plane wave of the same frequency.

## CONCLUDING REMARKS

The limited comparison of sound power attenuation obtained from experiment and theory is very encouraging. The theoretical studies are continuing to ease the restrictions imposed by some of the assumptions. These studies will include the effects of steady flow, wall impedance variation with suppressor length, and annular geometry. Additional experiments are being performed to further explore the validity of the theory and its extensions.

## REFERENCES


1. Rice, Edward J.: Attenuation of Sound in Soft Walled Circular Ducts. NASA paper presented at Symposium on Aerodynamic Noise (Toronto, Canada), May 20-21, 1968.
2. Morse, Philip M.: Vibration and Sound. Second ed., McGraw-Hill Book Co., Inc., 1948.
3. Smith, L. Jack; Acker, Loren W.; and Feiler, Charles E.: Sound Measurements on a Full-scale Jet-Engine Inlet-Noise-Suppressor Cowling. NASA TN D-4639, 1968.
4. Phillips, Bert: Effects of High-Wave Amplitude and Mean Flow on a Helmholtz Resonator. NASA TM X-1582, 1968.

## ANALYTICAL MODEL

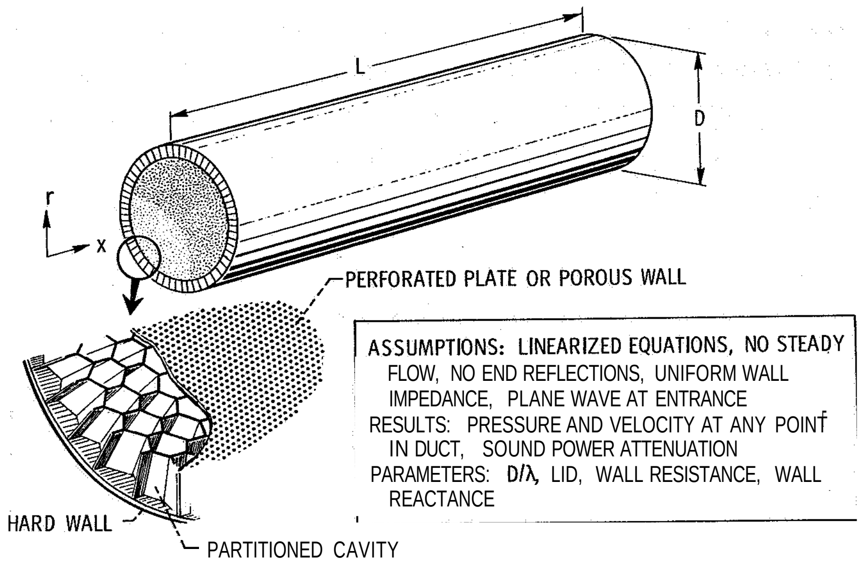


Figure 1

## EFFECT OF x/D ON RADIAL PRESSURE PROFILES FOR TWO VALUES OF D/λ

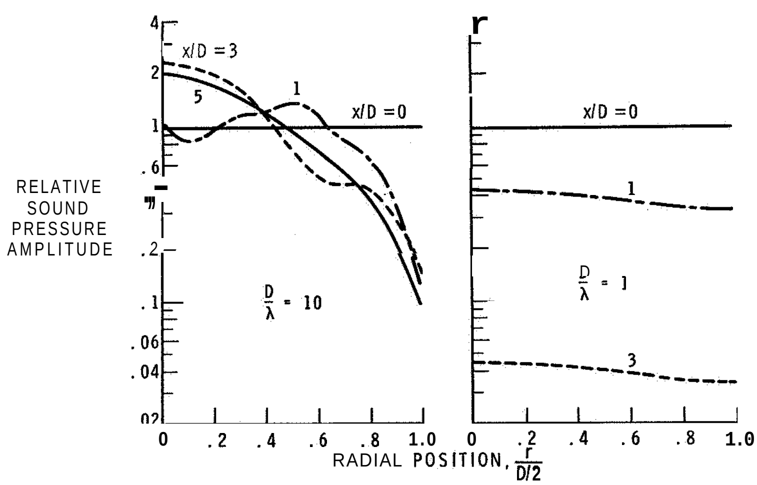


Figure 2

## SOUND POWER ATTENUATION CONTOURS

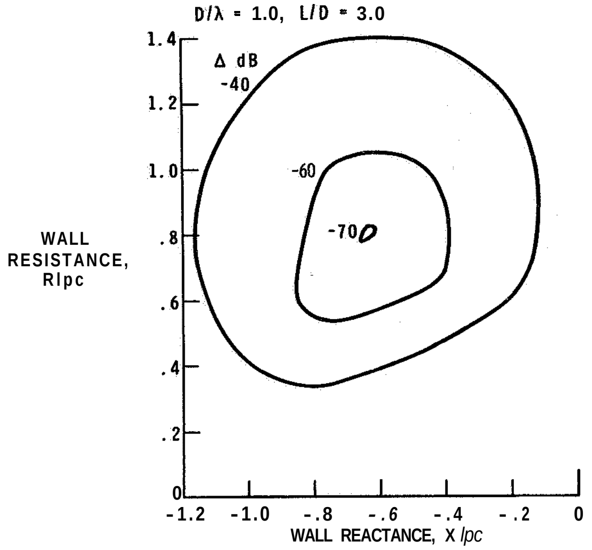


Figure 3

## EFFECT OF D/h ON MAXIMUM ATTENUATION

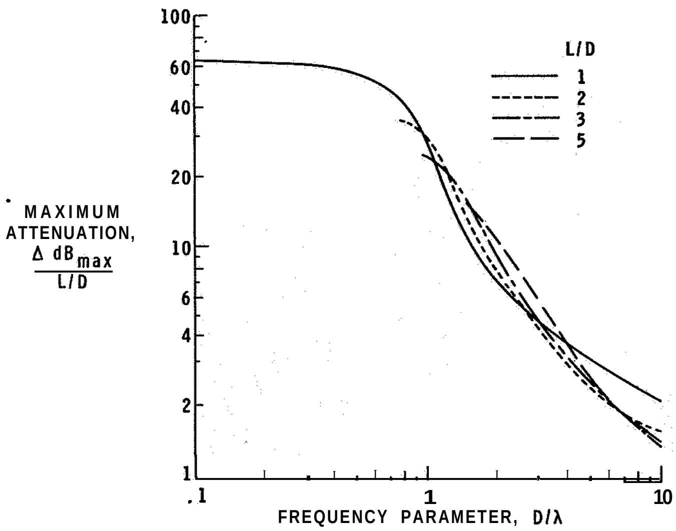


Figure 4

THEORETICAL SOUND POWER ATTENUATIONS FOR L/D=1 AND 5
WITH THE SAME WALL TREATMENT

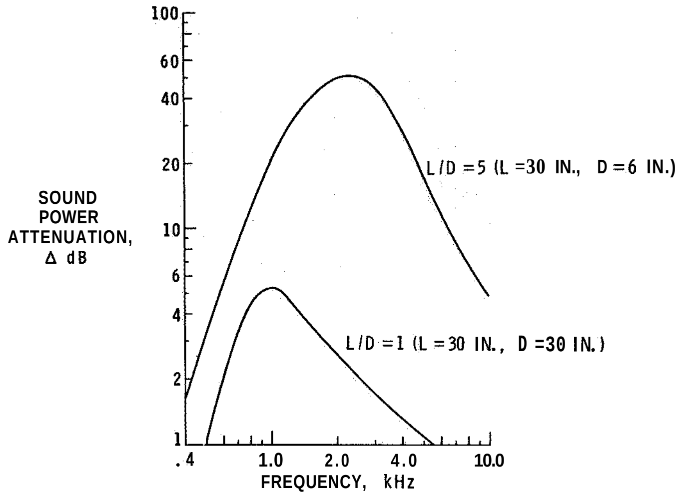


Figure 5

J-65 ENGINE WITH HARD COWL AND BELLMOUTH INLET

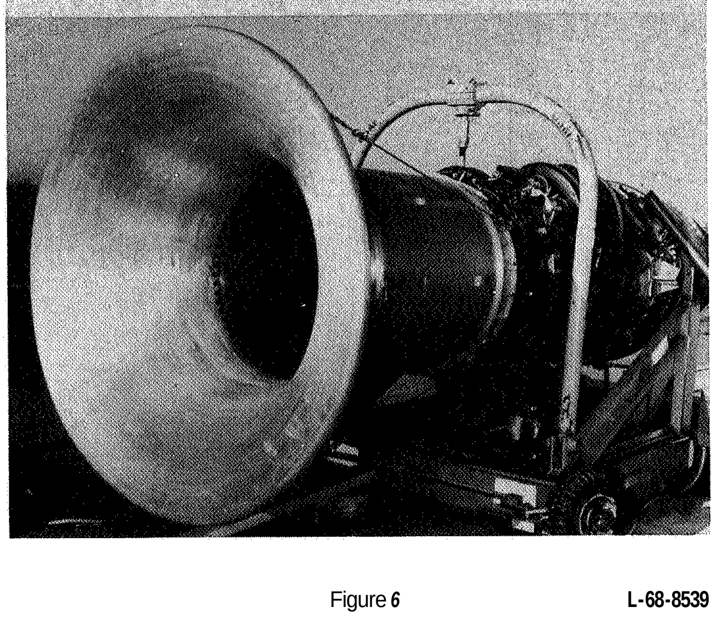

Figure 6 L-68-8539

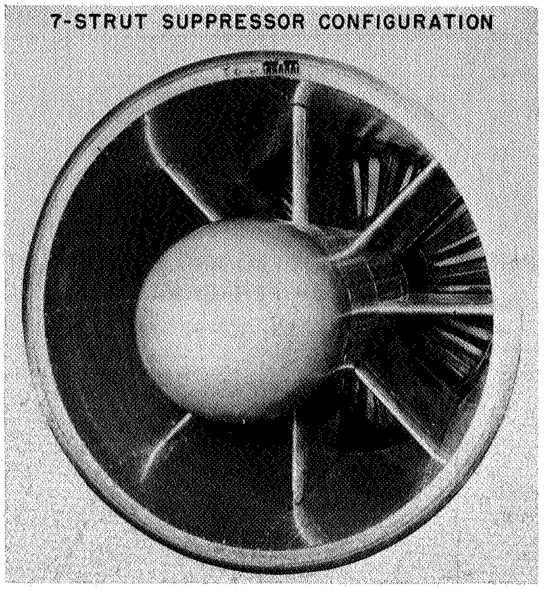


Figure 7 L-68-8540

COMPARISON OF PREDICTED AND EXPERIMENTAL SUPPRESSOR PERFORMANCE

CYLINDRICAL SUPPRESSOR; 1/3-OCTAVE DATA

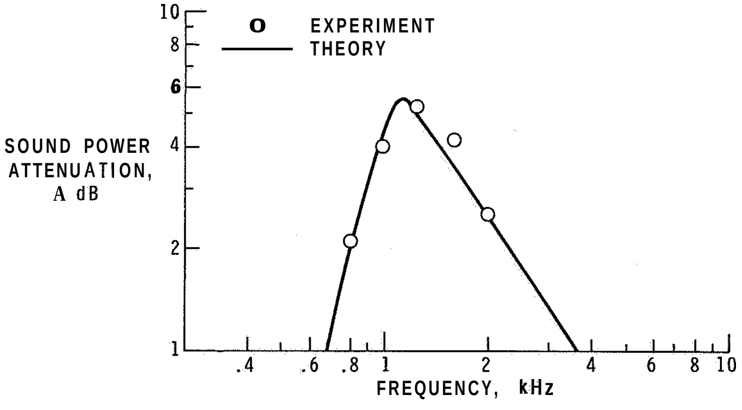


Figure 8

COMPARISON OF PREDICTED AND EXPERIMENTAL SUPPRESSOR PERFORMANCE

7-STRUT SUPPRESSOR; 1/3-OCTAVE DATA

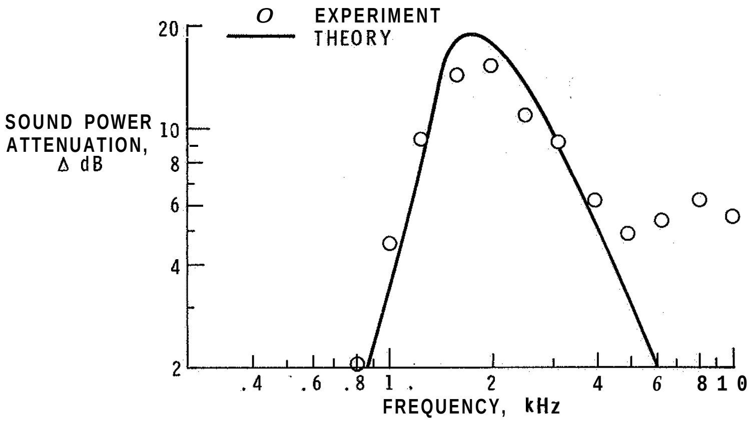


Figure 9