THE UNIVERSITY OF MICHIGAN

INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

SOUND POWER SPECTRA FROM SUBSONIC JETS

and

SOUND POWER STUDIES OF MODEL SILENCERS

Norman E. Barnett

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TABLE OF CONTENTS

LIST OF SLIDES

SOUND POWER SPECTRA FROM SUBSONIC JETS

INTRODUCTION

The experimental studies, described in this paper, were motivated by a desire to understand more fully the acoustical situation involved in silencing of jet engines during ground runup. Existing literature on this aspect of jet noise control is scanty and tends to neglect the physics of the silencing problem. In contrast, noise generation by ordinary jets has been studied extensively with the result that a comparatively complete physical description of noise-generation by a simple jet, unadorned with any auxiliary attachment, is available.

In the present case, we desired to conduct experimental silencing research using cold model jets and to employ the radiated sound power as the measure of acoustical performance. The reverberation room method for evaluating sound power permitted convenient measurement of the radiated power with complete freedom to ignore the directionality of the noise sources, whatever they may be, so far as these measurements were concerned.

The reverberation room method for accurate sound power measurement probably is not as familiar as the free-field method employed either out-of-doors or in an anechoic room. However, the reverberation room method rests upon a firm theoretical and experimental foundation. A detailed justification of the method lies outside of the scope of this paper except perhaps for the credence engendered by some data showing close agreement with free-field measurements.

The experimental arrangement consisted of installing both the nozzle end of the air-flow system and the microphone end of an audio spectrometer system in a reverberation room. Slide 1 shows schematically the air-flow system used to operate the small nozzles. High pressure air at roughly 100 psig passed through a pressure regulating valve (Reg. 1) into a float-type flow meter. By suitably adjusting the air pressure existing at the flow meter with respect to the air temperature existing there (that is, effectively adjusting the density of the compressed air), the flow meter was made direct reading with a full-scale value of 100 SCFM (standard cubic feet of air per minute). In this particular case, where the temperature ranged roughly from 60 to 130°F, the intermediate pressure was in the range of 40 to 50 psig. The directly-indicated mass rate of air flow was used as the primary measure of the aerodynamic condition prevailing at the nozzle. Beyond the flow meter, the air passed through a second pressure regulating valve (Reg. 2) and a calming chamber, lined with acoustical absorption and flow-smoothing screens, before arriving at the interchangeable nozzle. The maximum air-flow velocity in the calming chamber was less than two feet per second so that all interesting and noise-associated flow conditions occurred at or after the nozzle. The nozzle end of the settling chamber projected into the reverberation room.

The used air was exhausted from the reverberation room through a duct, one square foot in area and equipped with an acoustical filter. The filter served primarily to reduce the noise level in the room adjacent to the reverberation room for the comfort of the experimenters. The escape of the excess air from the reverberation room was essential in preventing short term rapid fluctuations in the barometric pressure there, which in turn would have caused troublesome problems with respect to microphone calibration.

The acoustical instrumentation is shown schematically on Slide 2 and is straightforward in concept. A one-half inch diameter condenser microphone was selected in order to enjoy the convenience of a flat diffuse-field response from low frequencies up to the vicinity of 20 kc. The microphone calibration was checked using an absolute diffuse-field reciprocity technique (Reference 1) and was found to agree with the manufacturer's calibration to within a fraction of a decibel. Sound pressure measurements were taken of the jet noise in the 24 one-third octave bands whose geometric-mean frequencies ranged from 100 cps through 20 kc. The levels of the individual bands were observed on a damped rms meter. Sound power determination by the reverberation room method requires a knowledge of the decay rate of the room, band for band, and Slide 2 indicates the decay rate instrumentation also.

At the bottom of Slide 2 is the formula relating sound pressure and decay rate to sound power (References 1 and 2), where:-

W = acoustic power per band, watts;

p^2 = space and time averaged sound pressure squared per band, measured with calibrated microphone;

D = space and time averaged decay rate per band, measured quantity;

V = volume of the reverberation room;

ρ, c = density and velocity of sound respectively for the atmosphere in the reverberation room;

$\bar{a}$ = Sabine absorption coefficient for the reverberation room, computed from decay rate and only of second order importance.

The broadband power, of course, is computed by summing the contributions from the individual one-third octave bands.

The experimentally determined decay rates were always utilized for computing sound power. Only negligibly small changes in these decay rates occurred during a group of test runs as a result of the compressed air being released into the reverberation room. However, significant changes in decay rates, especially at the higher frequencies, were encountered over longer time intervals and were due to the normally fluctuating moisture content of the atmosphere.

The stated upper-frequency limit of 20 kc may have aroused some concern about the validity of results. Measurements in reverberation rooms for architectural acoustical purposes are seldom extended upward as far as 10 kc. The University's 5400 cubic foot reverberation room seems to provide satisfactory, although less perfect, diffusion through 20 kc. Extension of the measuring range to still higher frequencies could not be achieved satisfactorily in a room of this volume.

Slide 3 shows a comparison of jet noise data obtained in the reverberation room with that obtained in an anechoic room. One of the nozzles used by W. C. Sperry (Reference 3) was duplicated and tested. The solid line is Sperry's partly theoretical, partly emperical reference curve. The circles are his experimental data and the crosses are the Michigan experimental data. The results agree almost perfectly - even the departures from the reference curve are duplicated.

ACCEPTED DESCRIPTION OF JET NOISE

As mentioned earlier, most of the definitive literature on jet noise refers to a simple subsonic jet exhausting into the surrounding atmosphere. It seemed wise, as a portion of this research program, to conduct some experiments whose results could be compared with the literature. This literature, taken as a whole, leads to a widely accepted description of simple subsonic jet noise. Many people have contributed to this description; M. J. Lighthill, Alan Powell, and H. S. Ribner, to mention a few. This description, as I understand it, is summarized on Slide 4. The total radiated acoustic power generated by a stationary cold ambient-air jet is:

$$W_{acoustic} = K\rho_o U^{+8} d^{+2} c_o^{-5} \text{ watts}$$

where:-

ρ_o, c_o = respectively the density and velocity of sound in the ambient air;

K = acoustical power coefficient, emperically evaluated and generally of order 6×10^{-5};

U = mean stream velocity of the jet flow;

d = diameter of the nozzle exit.

This equation is one form of the well-known eighth power law and states that the total radiated sound power varies as the eighth power of the efflux velocity and directly as the cross-sectional area of the jet.

A generalized spectrum shape is described as shown in Slide 4. It exhibits a broad maximum and when the frequency dependence is plotted in proportional band widths, it has a slope of roughly +9 db per octave below and -2 db per octave above the maximum. Also, the location of this maximum with respect to frequency has been found to obey a Strouhal number relationship:

$$f_m = S_m U/d$$

where the corresponding Strouhal number, S_m, is emperically about 0.2.

These features constitute the widely accepted description of simple jet noise: a spectral distribution whose shape is invarient to changes in size and velocity; the location of the frequency maximum, with respect to frequency, dependent upon the ratio of velocity to diameter and, with respect to level, dependent upon U^{+8} and area. (References 4 and 5)

RESULTS

A series of experiments were undertaken which attempted to operate within the above framework. Slide 5 presents the expected spectral curves for three small nozzles whose areas are in the ratios of 1 : 2 : 4, all operating at the same total mass flow rate of 100 SCFM. The dots represent the experimental data for three corresponding smooth-approach nozzles indicated at the right of the slide. Obviously, a general agreement exists between the expected and the observed behavior but there are also clearly discernible systematic differences.

First, consider the uppermost curve for a one-half inch jet at 100 SCFM, corresponding roughly to Mach one (actually 1.05 based upon observed pressure ratio). Using a value of $K = 6.0 \times 10^{-5}$, a total sound power level of 115.5 db reference 1×10^{-12} watt would be expected. The broadband sound power level calculated from band measurements through 20 kc is 114.4 db; a very close agreement considering the fact that even higher frequency bands would be expected to contribute something to the total power. Assuming a continued -2 db per octave behavior, another 1.3 db might be added to the experimentally observed power for a grand total of 115.7 db compared with 115.5 db. Certainly the total power observed for the one-half inch nozzle at sonic velocity is in very satisfactory agreement with the predicted value.

If the peak of the expected curve is located at 6300 cps, as suggested by the best visual fit to the experimental data, then the Strouhal number is 0.23; also in satisfactory agreement with an expected value of about 0.2.

Considering the two larger nozzles, we would anticipate successive power reductions of 2^{-7} or -21 db and downward frequency shifts of $2^{-3/2}$ or 4-1/2 one-third octave bands. Experimentally, power level reductions of first 18.9 db and then 15.4 db are observed; considerably smaller than expected. The spectrum shape flattens and it becomes difficult to state explicitly what happened to the spectral peak except that it did not behave as expected.*

Three different nozzles were used to obtain the data shown in Slide 5 and consequently there is a doubt about whether these nozzles really provided comparable situations. Perhaps some of the flow parameters, something related to boundary layer (Reference 6), for example, were altered inappropriately. Therefore, the one-half inch nozzle was tested again, only this time the mass flow rate was varied. Slide 6 illustrates the results. The solid curves again represent the predicted behavior and the dots the corresponding experimental data. Alternately open and closed dots have been used to aid in identification of the parameter.

As the mass flow rate (velocity) decreases, the deviations between the expected and the experimental results grow until at 25 SCFM, the curves would hardly be recognized as related. The deviations follow the same trends as illustrated on the previous slide (Slide 5). The spectrum flattens as the flow velocity decreases and proportionally more

*Dr. Alan Powell of U.C.L.A. suggests that the observed deviations are due to a flow separation occurring upstream of the nozzle in the calming chamber. If this is so it would change my interpretation of the observed results. This point has not been fully resolved at present.

of the energy appears at higher frequencies. On the basis of broadband power, agreement would appear quite good as low as 40 SCFM but the spectral data clearly evidence consistent though compensating departures at 60 SCFM.

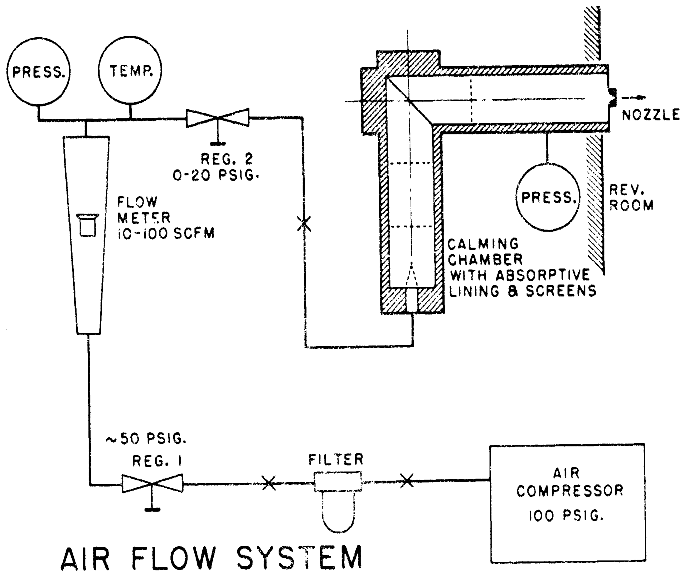
A more critical examination of these broadband data on Slide 7 reveals that a consistent trend existed here also. At the lower flow rates, more sound power is observed than expected on the basis of the eighth power law and the slope of the curve suggests a dependence on velocity to some power smaller than the eighth.

We might expect that at low enough velocities, the sound power characteristics of jets ought to tie up with those of ventilation system grills. I have not been able to find information on grills in appropriate form for a detailed comparison however there is some confirmation of the rising spectral shape and a velocity dependence to some lower power than the eighth. (Reference 7 and 8)

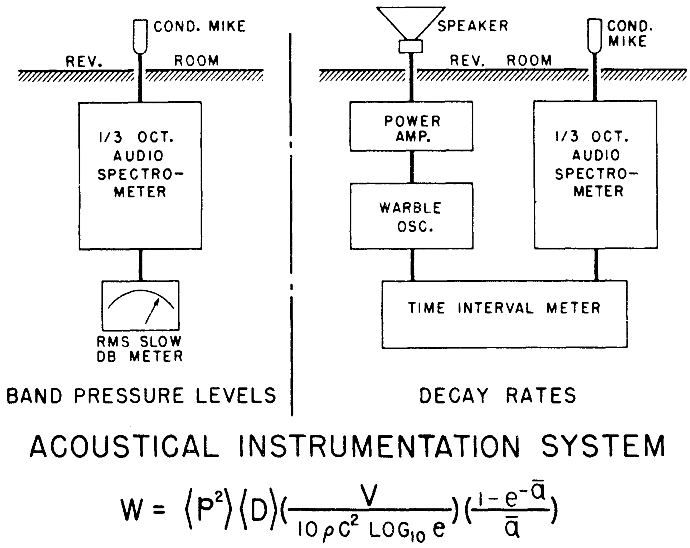
CONCLUSIONS

The experimental results reported here, taken in conjunction with existing literature on jet noise, suggest that the usual description of jet noise be reserved for the high subsonic velocity range and be incorporated as a special case within a broader description of jet noise. Such a description is only vaguely perceived at present but these results do provide some hint of what it must account for. One can even begin to visualize what the theory might contain. For example, regardless of the jet velocity (and therefore the rate of energy transport), the shear just outside the nozzle must be very large. As a consequence, an initial strong small-scale turbulence, radiating high frequencies, should be expected there. But whether or not powerful large-scale turbulence will develop might depend on the amount of energy available initially and on the competition among several dissipative mechanisms.

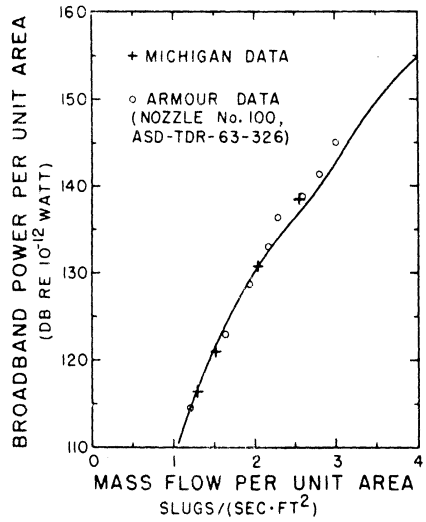
On the experimental side, much work remains to be accomplished particularly at low and medium velocities to delineate the physical behavior clearly. The present research program could not proceed further since its principal purpose lay in a different direction. It has however, provided very strong evidence of some unfinished acoustical business which hopefully some among us may be able to complete.



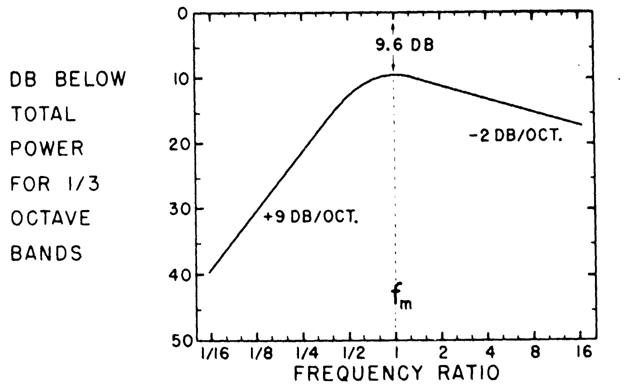
Slide 1. Air Flow System.



Slide 2. Acoustical Instrumentation System.



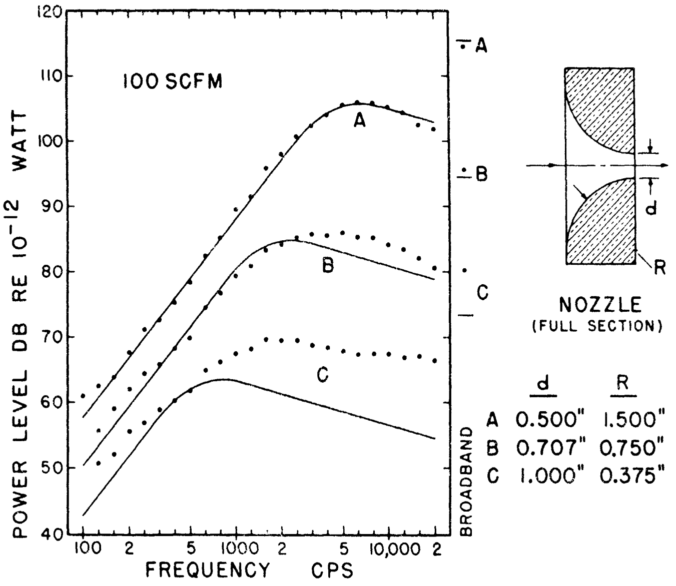
Slide 3. Diffuse vs Free-Field Results.



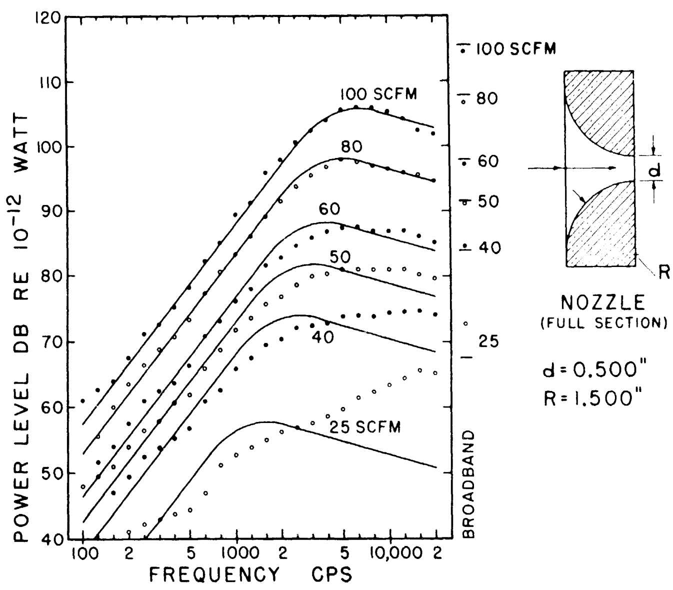
$$f_m = S_m U/d \text{ CPS}; \qquad S_m \simeq 0.2$$

$$W_A = K \rho_o c_o^{-5} U^8 d^2 \text{ WATTS}; \quad K \simeq 6 \times 10^{-5}$$

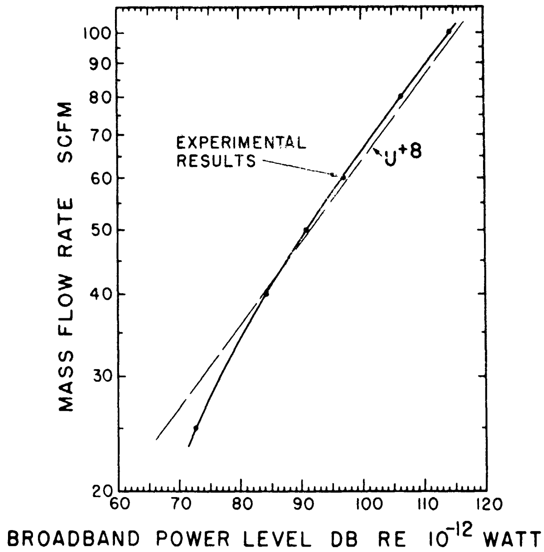
Slide 4. Description of Simple Jet Noise.



Slide 5. Jet Spectra; Different Nozzles.



Slide 6. Jet Spectra; Various Flow Rates.



Slide 7. Broadband Power vs Mass Flow Rate.

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SOUND POWER STUDIES OF MODEL SILENCERS

INTRODUCTION

The experimental studies, which are the subject of this paper, represent another portion of a larger research program related to the silencing of jet engines during ground runup. The preceding paper (Reference 1) contained a description of the experimental apparatus and the methods of measurement. The principal difference is that instead of being concerned with the noise generated by simple jets, here emphasis was concentrated on the acoustical effects accruing when objects are placed in, near, or around the jet efflux from some particular nozzle configuration. In these studies also, the mass rate of air flow through the nozzle system constituted the primary indication of a nozzle's aerodynamic condition while the radiated sound power determined by the reverberation room method yielded the primary acoustical data. As before, for the purposes of these experiments, directionality of the source or sources was ignored.

A rather wide variety of objects, placed in or around the jet flow, were investigated. These included plates, tubes, rods, wire screens, metal felts, and small glass beads. In all, some 680 sound power spectra were obtained. Only the wire screens and metal felts provided much promise of useful silencing.

EXPERIMENTAL RESULTS WITH SILENCER

From this large choice of material, I have selected a few results obtained with model silencers to present. (Probably some of the other results obtained with less complex configurations represent more basic physics but they are difficult to organize into a short paper.) Perhaps model silencer is a misnomer but it is used here to apply to a tubular object configuration which at least superficially resembles some existing runup silencers. In this instance, however, no attempt has been made to model the interior structure of any existing silencer.

Slide 1 shows the arrangement being considered. An extension piece has been appended to the one-half inch diameter smooth-approach nozzle (Reference 1) to move the nozzle's exit to a more accessible location. The model silencer consisted of two pieces of stainless steel tubing mitered and soldered together to produce the right-angled elbow as illustrated. Since many jet engines have a tail-pipe diameter of about 20 inches while this nozzle is one-half inch in diameter, the appropriate linear scaling factor is about 1 to 40 and dimensions scaled by this amount are shown on the slide. Initially the model muffler consisted of just the bare stainless steel tubing without any internal structure or acoustically-absorptive lining. Whether or not it represents an adequate model of any real silencer is debatable but there can hardly be any objection to it as just a laboratory configuration which, when tested in a certain way, gave the results presented below.

First, however, Slide 2 illustrates the acoustical consequences of adding the nozzle extension to the one-half inch diameter smooth-approach nozzle at two flow rates, 100 SCFM and 50 SCFM, which correspond roughly to Mach one and Mach one-half respectively. The results for the smooth-approach nozzle alone are drawn as a solid line while those for the addition of the extension are shown as dots. At the highest velocity, negligible change occurred but at the lower velocity, a significant increase occurred and so the higher values will have to serve as the base reference curve for the muffler experiments. With the extension in place, halving the mass flow rate only reduced the broadband sound power by 19.7 db instead of the 24 db expected from the description of simple jets or the 23.5 db found for the smooth approach nozzle alone.

Slide 3 shows the acoustical results for the bare solid-walled muffler compared to the extended nozzle alone. A tremendous increase occurred in the sound power radiated at low frequencies. At high frequencies and for the highest velocity flow, a small decrease occurred. Superimposed on the curves for the muffler are some bumps which are indicative of tonal generation corresponding to a pipe open at both ends. These remain fixed in frequency as the muffler is moved up or down stream but shift in pattern if the nozzle end of the muffler is closed off to yield an acoustical pipe closed at one end. The same general consequences occurred for both flow rates although the highest flow rate condition was affected most.

There are some theoretical reasons which would suggest that the results shown in Slide 3 were to be expected; something along the lines of "any solid surface in the vicinity of turbulence will enhance the acoustic radiation." In order to investigate this point, another muffler of identical dimensions was fabricated from perforated sheet metal. The solid area was reduced to 65 per cent by 1/8 inch diameter holes located in a triangular pattern, 0.185 inch on centers. Slide 4 illustrates the results. The solid curve is the reference curve for the extended nozzle alone. The dashed-curve is for the solid muffler and duplicates the results from the previous slide. The solid dots represent the acoustical results for the perforated muffler shell. It can be seen that these results lie intermediate between the extremes for the bare nozzle and for the solid shell.

In order to investigate the role of absorption without introducing acoustically opaque surfaces, the perforated muffler shell was wrapped on the outside with a layer of fine fiberglass. This fiberglass had a very thin coating of neoprene (I think) on the outside surface only. The acoustical results for this condition are also indicated on Slide 4 as open circles. This fiberglass (some that was lying around the laboratory and I have lost track of its original designation) would be expected to show appreciable absorption and transmission loss at high frequencies but negligible absorption and transmission loss at low frequencies (Reference 2). Indeed, these expectations are borne out. Reduced radiation is observed at high frequencies but the low-frequency behavior is reminiscent of a larger percentage of solid surface.

I do not think that the muffler walls are resonating to any appreciable extent. The increased low-frequency radiation is both at too low a frequency and too broadly distributed for such to be the case. This aspect of the problem can be investigated in another way. The solid muffler had impervious, fairly rigid, massive walls. Mr. Melvin Roquemore of the U.S. Air Force's Systems Engineering Group, who inherited project monitorship of this research, supplied a flexible-walled model muffler having approximately the same interior dimensions as the metal shells. It consisted of a double-walled structure of rubber-like material and was inflated with water. When filled with water, the model weighed about 17.51 kg. Thus it would seem to constitute an impervious, limp-walled massive structure. The acoustical results are shown on Slide 5 as open and closed dots and are essentially identical to those for the solid muffler (dashed curve). Only the detail of the low-frequency tones has shifted somewhat.

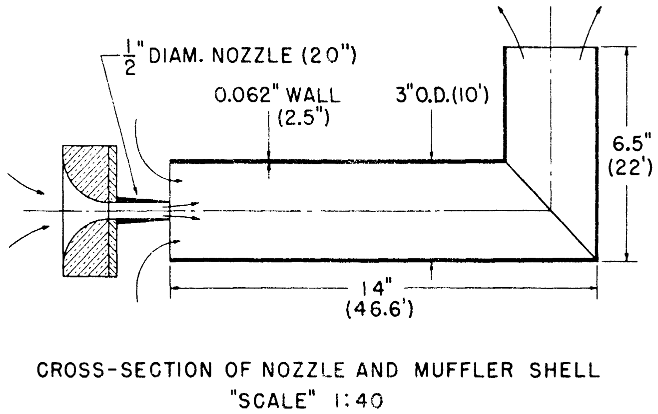
It would seem that if one attempts to fabricate a jet muffler using a solid shell, the first consequence is a large enhancement of especially the low frequency radiated sound power. Superimposed on this is the tonal generation to be expected for that particular geometry of enclosure. Possibly wall structural-resonances also will be superimposed if they occur but such have not been identified in this research. Then if one adds acoustical absorption inside the muffler shell, some reduction from the new higher base line of noise is to be expected. If one is clever enough and uses effective enough absorbing materials, some real reduction in sound power below that for the bare jet itself may accrue particularly at high frequencies but the low-frequency power is apt to remain larger than without a muffler shell.

I have one more slide to present. In this case, we started with the idea of introducing screens and metal felts while avoiding the introduction of solid surfaces in so far as possible. That was the starting point, however, Mr. Philip Kessel, who was conducting these particular measurements, began to make improvisations on the original theme using the screens, metal felts and other objects associated with our experimental program. Slide 6 shows the results obtained with one of his conglomerations. It did not raise the backpressure at the nozzle. The radiated sound power was reduced by an appreciable amount at all frequencies. The broadband power was reduced by 39 db or to about 1/10,000 of its original magnitude while some individual bands were reduced by 44 db.

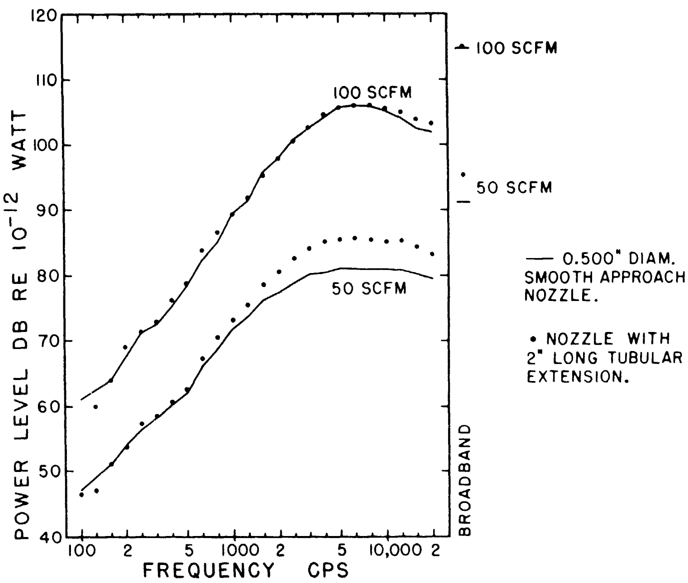
I do not think that this combination of materials is necessarily anywhere near optimum. Too little research time remained to take this combination of objects apart and to attempt to optimize the composite for each addition. The original idea of avoiding solid walls got lost somewhere and the rising trend at the upper frequency limit is slightly disturbing although presumably it could be controlled with conventional absorption. Probably a more compact geometry is possible without compromise of the acoustical performance although scaled up 40 times, it would only be 13 feet in diameter by 8 feet long.

Just why and how this muffler operates is not at all evident. By that I mean, we do not possess a detailed physical understanding of the silencing action of certain screens, metal felts, and other objects placed in the jet flow; much less understand how a combination of such objects functions or how they should be combined rationally to meet a specific silencing criterion.

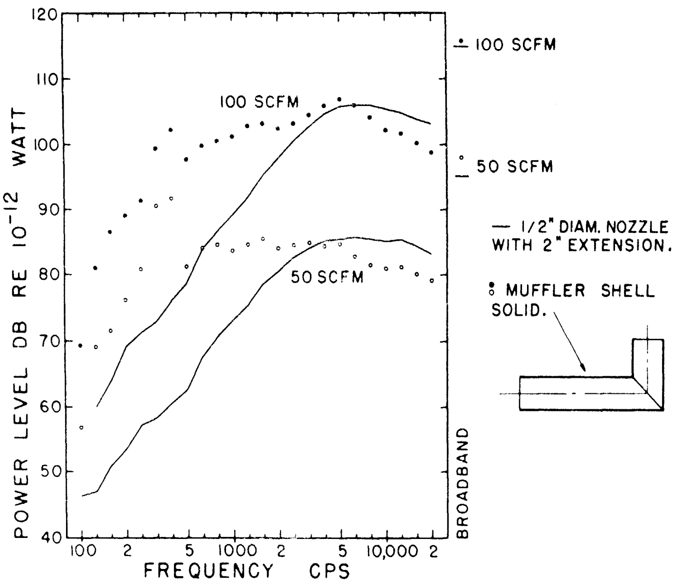
Nevertheless, the results shown on Slide 6 demonstrate beyond question that significant sound power reduction can be achieved. That is, physical processes exist by which large reductions can be achieved. These results are orders of magnitude more effective than the results for any single screen or metal felt tested; thus it appears that some form of cascading is applicable. However, these results with small cold nozzles merely affirm the possibility of compact runup silencers having large effect --- they do not provide a fully-engineered operational design. A considerable amount of basic research needs to be done to fully understand the physics of the acoustical processes associated with fluid flows and much development engineering rests thereupon to realize optimized, industrially-fabricated runup silencers.



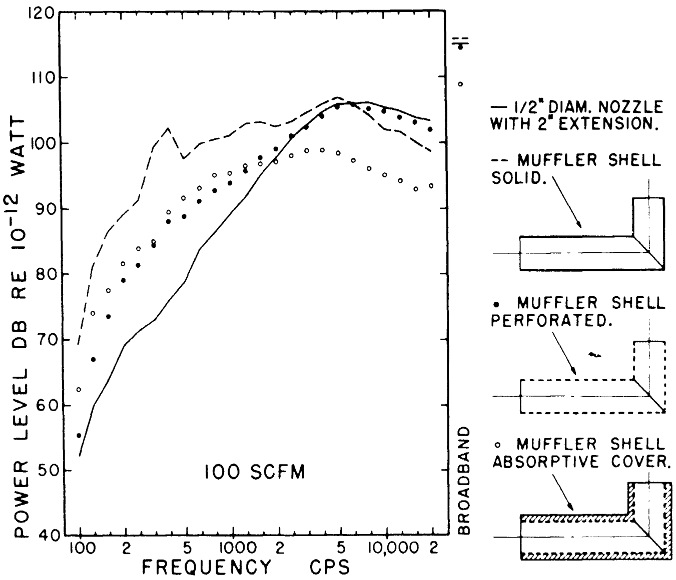
Slide 1. Experimental Configuration.



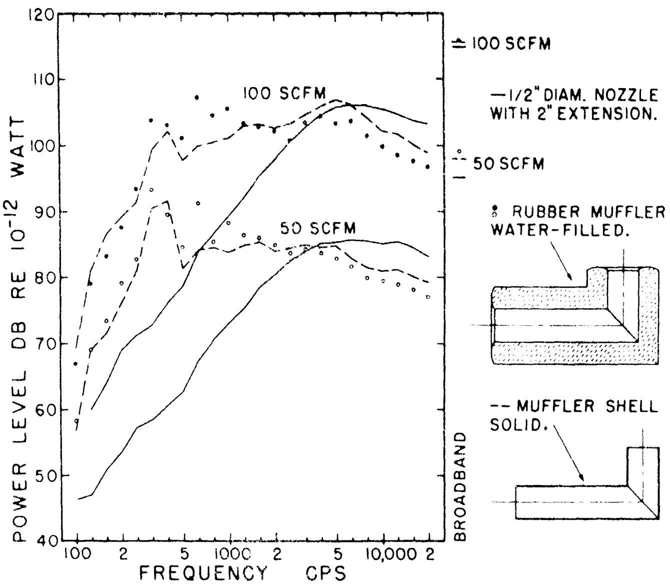
Slide 2. Effect of Nozzle Extension.



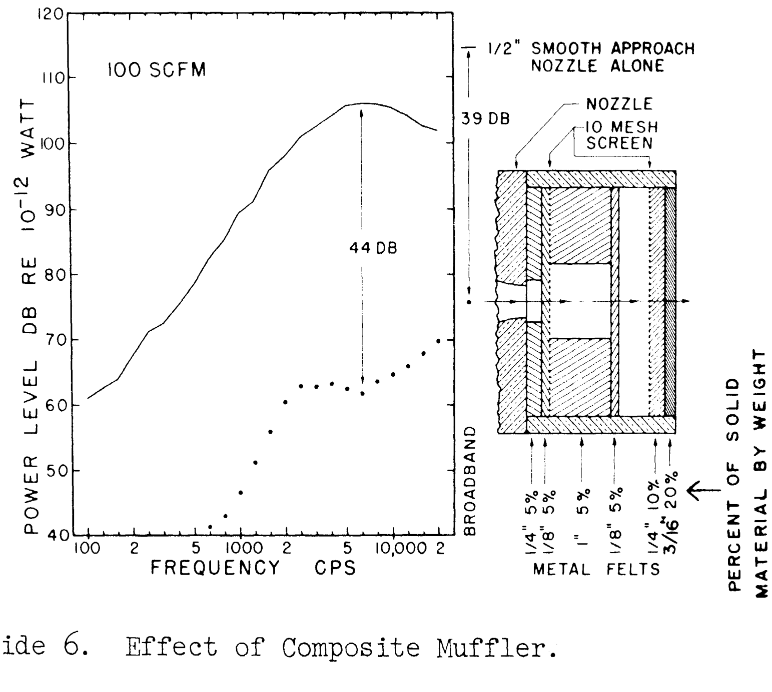
Slide 3. Effect of Solid Shell.



Slide 4. Effect of Different Shells.



Slide 5. Effect of Water-Filled Shell.



Slide 6. Effect of Composite Muffler.

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