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AN EXPERIMENTAL STUDY OF NOISE ATTENUATION BY LIQUID INJECTION

Prepared by:

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March 1971

Progress through Research

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16. Abstract

Fan and compressor noise represents an important part of the jet aircraft noise problem. This report describes an experimental program which began the evaluation of a novel noise reduction technique--liquid injection--which may be useful for fan and compressor noise reduction. The technique is based upon the fact that liquid droplets in an airstream can reduce noise in two ways: By altering the noise generation mechanisms, and by increasing the attenuation along the path of propagation.

The objective of this investigation was to define the magnitude of noise attenuation which can be produced by injecting evaporating droplets into a subsonic duct flow. This flow model, which is simple compared to the actual flow in an axial compressor or fan, was selected as a first step to explore the potential of noise attenuation by liquid injection. The reduction of noise generation by liquid injection was not studied; this must await testing on a fan or compressor.

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An Experimental Study
of Noise Attenuation by Liquid Injection

By Lewis A. Maroti and Gerald B. Gilbert

SUMMARY

Fan and compressor noise represents an important part of the jet aircraft noise problem. This report describes an experimental program which began the evaluation of a novel noise reduction technique--liquid injection--which may be useful for fan and compressor noise reduction. The technique is based upon the fact that liquid droplets in an air stream can reduce noise in two ways: by altering the noise generation mechanisms, and by increasing the attenuation along the path of propagation.

The objective of this investigation was to define the magnitude of noise attenuation which can be produced by injecting evaporating droplets into a subsonic duct flow. This flow model, which is simple compared to the actual flow in an axial compressor or fan, was selected as a first step to explore the potential of noise <u>attenuation</u> by liquid injection. The reduction of noise <u>generation</u> by liquid injection was not studied; this must await testing on a fan or compressor.

The basic mechanisms of attenuation by liquid droplets are viscous losses, thermal relaxation losses, and losses due to irreversible phase changes. To investigate the influence of these attenuation mechanisms, a test rig was designed to operate with several combinations of liquids and gases, with droplet sizes from 190μm to 1900 μm, and with pure tone frequencies from 100 Hz to 10 KHz. Orifice plates were used to create droplets with a narrow size distribution band, and a traversing microphone was used to measure attenuation with length along the test section.

The initial test runs employed water droplets in air. Tests with 1900μm droplets were performed at liquid-to-vapor mass fractions of 0.22 and 10.3

with the sound frequency ranging from 200 Hz to 6000 Hz. Tests with the 190μm droplets were performed at mass fractions from 0.5 to 2.2 with the sound frequency ranging from 200 Hz to 850 Hz. The sound pressure level (SPL) in the region near the acoustic generator and droplet injection plate dropped measurably--from 2 db at 200 Hz to 5-10 db at 6000 Hz--when liquid injection was used. No additional attenuation was measured along the duct. The reduction in SPL just downstream of the droplet injection plate may be due to the effect of the altered acoustic impedance on the driver, absorption of acoustic energy by the droplet break-up process, or alteration of microphone characteristics.

A review of the results of the initial tests employing newly-available analytical tools indicates that the drops tested were larger than optimum. Their evaporation rates were too small to achieve maximum attenuation. Additional tests, employing smaller droplets and more highly-evaporating liquids, should be performed to bracket the predicted peak attenuation regions. The results of such tests will define the magnitude of attenuation possible with evaporating drops, and will provide information to quide planning of tests of this attenuation technique in actual compressors or fans.

Section 1

INTRODUCTION

The jet engine has caused a revolution in the transportation industry by reducing dramatically the cost of air travel. Air travel today is growing at an unprecedented rate, and is today far in excess of the predictions made just a few years ago. The new business and new jobs in this industry have had a direct beneficial effect upon the nation, and indirectly, air travel has had an even larger effect by increasing the efficiency of businessmen in travel. The next generation of jets -- larger or faster -- promises to accelerate this.

Unfortunately, jet engines also generate objectionably large amounts of noise, and this noise is rapidly becoming a problem of such major proportions that the continued growth of air travel is literally threatened. Ground noise on aircraft ramps can now constitute a health hazard to ground support personnel. In every major city in the country, public indignation over jet engine noise is hampering the necessary expansion of airports, and it is seriously complicating the selection and development of sites for new airports to handle the expected air travel needs.

Means to reduce and to control jet engine noise are absolutely necessary if the growth of air travel is to keep pace with the needs of the nation for rapid, economical transportation.

This experimental program began the evaluation of a noise reduction technique-liquid injection- which may provide a means to effect significant reductions in the noise generated by jet engines.

Some recent experiments have shown that the presence of liquid droplets in an air stream can reduce noise levels in two ways: first, droplets can increase the rate of attenuation of noise as it passes through the air-droplet mixture; second, the presence of evaporating droplets can alter the mechanisms of noise generation in fluid flows so that less noise is generated at the source.

Our present understanding of these phenomena is meager. Some progress has been made in understanding the mechanisms of noise attenuation in non-evaporating fogs and in aerosols, but there is only a limited analytical explanation of the mechanisms by which evaporating droplets can cause noise attenuation. There is no information available, analytical or experimental, on the mechanism by which evaporating droplets can reduce the amount of noise generated in fluid flow devices.

The purpose of the experimental work reported here was to define the magnitude of noise attenuation which can be produced by evaporating droplets in a gas stream moving at a constant subsonic velocity in a duct. This flow model, which is simple compared to the actual flow in a gas turbine engine, was selected as a first step to explore the potential of sound attenuation by liquid injection.

Section 2

NOMENCLATURE

A	cross-section area	ft^2
C_m	liquid mass fraction	
C_n	number of particles per unit volume	
C_p	specific heat at constant pressure	BTU/lbm-° F
C_t	terminal velocity of drops	ft/sec
d	drop diameter	ft
D_v	molecular diffusivity	ft^2/sec
f	frequency	Hz
k	thermal conductivity of gas	BTU/sec-ft
L	length of test section	ft
L_e	Lewis number	
m	mass flow rate	lbm/sec
P_r	Prandtl number	dimensionless
Q	volume flow rate	ft^3/sec
R	gas constant	53.35 $\frac{ft\text{-}lbf}{lbm\ °R}$
S_c	Schmidt number	
SPL	sound pressure level	db re 0.0002 microbar
T	temperature	° F
α_T	thermal diffusivity	$ft.^2/sec.$
λ	latent heat of evaporation	BTU/lbm
ν	kinematic viscosity of gas	$ft.^2/sec.$
ρ	density	$lbm/ft.^3$
τ	relaxation time	seconds
ω	circular frequency	sec^{-1}

Subscripts

D	mass transfer effects
E	evaporation
L	liquid
R	residence time
T	thermal effects
v	viscous effects

Section 3

BACKGROUND

3.1 Related Work

The two major noise sources in a jet engine are the fan/compressor and the jet itself. Control of fan/compressor noise is presently sought by selecting design parameters to minimize noise generation and by applying acoustic treatment to the engine flow ducts. The jet noise of engines installed in aircraft is controlled by shaping the jet nozzle and by reducing the jet velocity.

Liquid injection has been used in the compressors of jet engines for thrust augmentation (e.g., reference A1) but no measurements of the resulting acoustic effects have been reported. Liquid injection has been used in the turbine section for cooling purposes (reference B1.) References (K3), (W1), and (W3)show that significant noise reduction can be achieved by injecting water into the jet exhaust stream. It is believed that the liquid droplets in the jet stream reduce the noise generated within the turbulent jet flow, and also increase the attenuation of the generated jet noise by a gas-liquid interaction. The relative magnitude of noise attenuation due to these effects has not been investigated as yet.

Experience with centrifugal refrigeration compressors has shown that when a liquid is injected into the inlet of high-performance compressors for temperature control, the compressor noise is reduced. In this case the liquid evaporates completely in the compressor so that no trace of liquid can be found in the discharge stream.

These two seemingly unrelated results indicate that liquid injection into the gaseous phase can influence the mechanism by which noise is generated, both when the noise is generated by turbulence, and when it is generated by inhomogeneous pressure fields due to energy exchange between gaseous fluid and rotating solid surfaces. The mechanisms of this noise attenuation cannot be explained satisfactorily from present knowledge. The available information on this effect is insufficient to evaluate the full potential of liquid injection for noise reduction in the axial compressor stages of jet engines.

Before discussion of the steps taken to explore how liquid injection can be applied, let us review what is known about the general subject.

The basic mechanism of sound attenuation by liquid droplets in a stationary gaseous fluid has been studied extensively by analytical and experimental methods for fogs or aerosols in air. References (S1), (E1), (O1), (U1), (K1), (E2), (Z1), (M2), (C1), (C2), (W3), (M1), (M3), and (M4) show the development during the past sixty years of analytical methods treating sound attenuation in two-phase flow. The various investigations assumed different sets of key parameters and different conditions for the analytical models which they used to describe the attenuation process. A list of these parameters is given in Table 1.

Reference (01) was the first to consider phase change. More recently, references (W3), (M1), (M3) and (M4) gave analytical descriptions of sound attenuation due to viscous, thermal, and phase change interactions in a gas/vapor - liquid flow. These references give the most thorough analytical description of attenuation in two-phase flow, although all of them include a number of restricting assumptions to reduce the mathematical complexity to a manageable level.

The analytical results of these references were used to guide the selection of test conditions for the present experimental program. It was assumed that those parameters which were found to be important in the analytical results will be important in the experimental program, but perhaps to a different degree.

3.2 Basic Considerations for Sound Attenuation in Two-Phase Flow

The basic mechanisms of sound attenuation by liquid droplets in a gas stream are as follows:

1. Viscous losses due to the relative motion of the air and liquid particles during the passage of acoustic waves.

2. Thermal relaxation, which can lead to significant losses of acoustic energy in the air alone. This effect can be enhanced by heat transfer between the liquid particles and the air.

3. Phase change, due to evaporation from the droplet or due to condensation on the droplet, causing a relaxation-type loss in the periodic process.

The properties of the gas/liquid systems tested can be characterized in terms of relaxation times corresponding to the attenuation mechanisms described above. These relaxation times are as follows:

τ_V = relaxation time for viscous effects (Stokes flow)

$$\tau_V = \frac{d^2}{18\nu} \frac{\rho_L}{\rho}$$

d = drop diameter
ν = kinematic viscosity of gas
ρ_L = liquid density
ρ = gas density

τ_T = relaxation time for thermal effects

$$\tau_T = \frac{d^2}{12\alpha_T} \frac{\rho_L}{\rho}$$

α_T = thermal diffusivity = $\frac{k}{\rho c_p}$
k = thermal conductivity of gas

τ_D = relaxation time for mass transfer effects

$$\tau_D = \frac{d^2}{12 D_v} \frac{\rho_L}{\rho}$$

D_v = molecular diffusivity

Two other characteristic times are of interest:

τ_E = droplet evaporation time constant (Stokes flow)

$$\tau_E = \frac{\rho_L \lambda d^2}{8k_f \Delta T}$$

λ = latent heat of evaporation
k_f = thermal conductivity of film
ΔT = temperature difference

τ_R = residence time in the test section

$$\tau_R = \frac{L}{\frac{Q}{A} + C_t}$$

L = length of test section

Q = gas volumetric flow rate

A = cross-section

C_t = terminal velocity of drops.

Other parameters important in the description of the process are:

Liquid volume fraction (flowing)

$$\frac{Q_L}{Q} = \frac{ft^3/\text{sec liquid}}{ft^3/\text{sec gas}}$$

Liquid mass fraction (flowing)

$$\frac{m_L}{m} = \frac{\text{lbm/sec liquid}}{\text{lbm/sec gas}}$$

Latent heat factor

$$\frac{\lambda}{RT_o}$$

λ = latent heat of liquid

R = gas constant

T_o = gas temperature

The droplet evaporation time (τ_E) describes the steady evaporation process which takes place in the flow. The relaxation time for mass transfer (τ_D) describes the periodic evaporation and condensation due to acoustic pressure variations at the surfaces of the liquid droplets. The residence time (τ_R) defines the maximum time which the liquid droplets spend in the flow and sound field inside the test section.

The ratios of relaxation times define the following dimensionless numbers, well known in fluid mechanics and heat transfer:

$$\frac{\tau_v}{\tau_T} = \frac{2}{3} \; Pr$$

Pr = Prandtl Number

$$\frac{\tau_D}{\tau_v} = \frac{3}{2} \ Sc$$ Sc = Schmidt Number

$$\frac{\tau_D}{\tau_T} = Le$$ Le = Lewis Number

Section 4

EXPERIMENTAL APPARATUS

4.1 System Operating Characteristics

To investigate the influence of liquid droplets on noise attenuation, it was necessary to design an experimental arrangement which can operate with several combinations of liquid and gases. The following ranges of operating parameters were desired:

• 3 droplet sizes	10 μm to 2000 μm
• 3 liquids	water, Freon, liquid nitrogen
• 2 gases	air, Freon
• air initial conditions	70°F to 170°F, saturated and unsaturated
• water initial conditions	70°F to 170°F
• gas flow rate	50 cfm to 250 cfm
• liquid flow rate	1.0 gpm to 5.0 gpm
• sound frequency	100 Hz to 10k Hz
• sound pressure level	10 to 20 db above background

Plans were made to test various combinations of parameters and flow rates to cover a broad range of the variables that were expected to influence the attenuation of noise at several selected pure tone frequencies.

The system needed for this test program, in addition to supplying, controlling, and measuring the quantities listed above, also had to satisfy the following requirements.

1. A narrow band of droplet size distribution had to be achieved for each basic droplet size.

2. A flow "adjustment region" had to be provided so that both gas flow and liquid droplets could be introduced uniformly into the test section.

3. A 10 foot long test section was necessary.

4. A traversing microphone system was required in order to measure noise attenuation with length along the test section.

5. A plenum was necessary at the end of the test section to act as an anechoic termination, a liquid separator, and an isolation plenum for closed-loop tests with Freon gas.

Sketches of the experimental system designed to satisfy these requirements are shown in figures 1 and 2. Photographs of the experimental rig are shown in figures 3 to 9. A blower, orifice system, and electrical heater were used to supply heated gas to the plenum at the top of the vertical test section. A perforated plate inside the plenum was used to distribute the gas flow uniformly into the test section. The liquid droplets were injected vertically downward into the gas stream from a pressure vessel with a large number of uniformly spaced holes of equal diameter.

The gas and liquid droplet mixture entered the test section and flowed past the acoustical driver and microphone. At the bottom of the 10 foot test section, the gas-liquid mixture passed through the anechoic termination and into a wooden box. This box can be closed at the top when a recirculating test is run. A microphone carriage and motor-driven pulley system were used to traverse the microphone through the test section. The length of the microphone traverse was 105 inches.

4.2 Design of Critical Components

1. Gas Blower System

The desired gas flow rate for the experimental program was 50 to 250 cfm. An available blower with a capacity of 600 cfm was selected. This blower easily met the air flow requirements. However, the same blower and motor when used with Freon could supply only about 50 cfm because of the higher density of Freon and the limited allowable operating current of the motor. This reduction in maximum possible Freon gas flow results in a lower maximum liquid Freon flow rate (.635 gpm).

2. Nozzle Spray System

The original plan for liquid injection was to use commercially available liquid spray nozzles. This type of liquid injection was found to have serious drawbacks for this application.

a. The droplet size distribution is very wide for commercially available liquid spray nozzles. For example:
   - a 2000μ nominal spray nozzle has 5% volume of liquid smaller than 840μ and 5% volume of liquid larger than 3700μ.
   - a 520μ nominal spray nozzle has 5% volume of liquid smaller than 250μ and 5% volume of liquid larger than 920μ
   - a 220μ nominal spray nozzle has 5% volume of liquid smaller than 125μ and 5% volume of liquid larger than 360μ.

b. Commercial spray nozzles spray the liquid in a conical pattern which will intersect the walls of the 8 inch diameter test section and run down the walls. The minimum available cone angle is a 30° included angle. A large percentage (maybe 25%) of the liquid would end up on the walls of the test section. If several nozzles with overlapping fields were used, the liquid droplets would be unevenly distributed over the cross-section of the test duct.

c. Pneumatic spray nozzles can produce 10μ nominal size droplets. However, 50 to 250 nozzles would be required to produce the desired 1 to 5 gpm flow rate due to the low flow capacity of such nozzles.

A new method of droplet formation was selected to eliminate the undesirable characteristics of the commercial nozzles. The principle of droplet formation used was the Rayleigh breakup of a liquid jet. This instability phenomenon is fully discussed in reference (M5) where non-dimensional correlations of jet breakup characteristics are presented. The correlations include the effect of liquid properties. Single nozzle tests were also conducted in the laboratory with water, Freon 11, and liquid nitrogen to verify the operating limits indicated by reference (M5).

According to the Rayleigh break-up theory and as substantiated by tests, the average droplet size formed by a liquid jet under normal operating conditions will be 1.89 times the jet diameter independent of the liquid properties. This value was used to calculate the nozzle plate hole sizes. The number of holes in each nozzle plate was selected to pass the desired liquid flow rate with the supply pressures available. This practical condition made it necessary to increase the minimum droplet size from 10μm to 190μm.

Table 2 shows the geometry and operating characteristics of each plate and liquid combination.

The holes in each nozzle plate were uniformly distributed inside a circle of 7.5 inches diameter. The stainless steel nozzle plate was attached to a 200 cubic inch liquid plenum chamber which can be pressurized to 100 psig. This nozzle plenum chamber was mounted on top of the gas plenum so that the liquid jets pointed directly downward. The jet breakup into droplets occurred within one foot of the nozzle plate.

For the water tests, the liquid plenum was pressurized by a pump to supply up to 5 gpm. For the Freon tests, the Freon flow of 0.635 gpm was supplied by filling the vessel and pressurizing to the desired level with high pressure air.

The liquid nitrogen flow rate had to be limited to 0.436 gpm when injected into 250 cfm of 170^{o} air so that the outlet air temperature would not fall below the moisture freezing point. The flow of liquid nitrogen was obtained by filling the vessel and using the boiled-off vapor to pressurize the container.

The nozzle plates had to be recalibrated for Freon and liquid nitrogen in order to relate flow rate to the applied pressure. Calibration was accomplished by timing the liquid discharge rate at selected plenum pressure levels.

3. Microphone Traverse System

A light-weight freely-moving carriage was designed and made from 1/16 inch diameter rods and 3 small rubber wheels to fit snugly inside the 8 inch diameter test duct. The microphone was elastically suspended on the carriage and shielded from the liquid so that it would be unaffected by carriage bounce, external vibration, and the falling liquid. A motor-driven pulley system was developed to move the carriage at rates from 5 fpm to 50 fpm down the test section.

4. Acoustic Termination

An acoustic termination made of three fiberglass wedges was installed at the lower end of the test section. This termination was intended to result in plane propagation waves in the duct in the frequency range between the cut-off of the termination and the lowest cross-mode of the test section (i.e., from 150 Hz to 850 Hz.) A thin plastic cover was placed over the wedges to prevent soaking by the liquid flow.

5. Sound Generating System

Pure tone sinusoidal signals were generated by an audio oscillator (figure 2). The signal was amplified by a single-channel power amplifier and fed to an acoustic driver. The sound pressure level in the test section was required to be at least 10 db higher than the background noise of all other equipment. The diaphragm of the acoustic driver had to be insensitive to moisture or to 100% relative humidity in order to operate properly in the test environment.

6. Acoustic Instrumentation

A piezoelectric ceramic type microphone was selected because of its low temperature coefficient and its insensitivity to the humidity. The microphone was isolated from effects of vibration, liquid droplets, and heat transfer by using a

"Scotchguard" covered foam enclosure and an elastic suspension. A single-lead shielded cable was used to connect the traversing microphone to the sound level meter (figure 2.)

The sound level meter was used for calibration and for reading the over-all sound pressure level. The long microphone cable picked up electrical noise from fluorescent lights and from the variable speed motor used to traverse the microphone. This noise resulted in a 90-96 db SPL reading on the C scale. Frequency analysis of the noise spectrum revealed that 60 Hz and 180 Hz components were dominating the background noise. By using a 1/3 octave band filter on the sound analyzer centered at the frequency of the audio oscillator, the background noise was 20 db lower than the sound pressure level of the test tone at all test conditions.

The graphic level recorder was used to record the microphone signal versus time. Since the microphone was traversed at constant speed, the recorded traces describe the spatial variation of the sound pressure level along the test section. The signal from a 16 inch microphone traverse was recorded over a 1 inch chart length by proper selection of the traverse speed and the paper speed combination.

4.3 System Operation

Table 3 summarizes the flow rate and temperature capabilities of the test rig. The operating range for air was achieved by throttling the blower flow and varying the power input to the 10 kw electrical air heater. The air system can be operated as an open system where almost 10 kw is needed to heat a flow of 250 cfm to 180^{o}F, or it can be operated as a closed recirculating system requiring very little heater power. The Freon system must be operated in a closed, recirculating mode at atmospheric temperature with the flow rate controlled by the throttle valve.

The water loop as shown on figure 1 included a reservoir, pump, rotameter, throttle valve, heater, filter, nozzle plenum and nozzle spray plate. Two rotameters in parallel allowed flow measurement over a range from 0.3 gpm to 6.7 gpm. An 18 kw main heater was used to increase the system water temperature slowly to $180^{o}F$. When the desired water temperature was reached, the heater power was shut off and the water temperature was allowed to stabilize. The test was started by switching two valves to divert the recirculating water flow into the nozzle chamber. The reservoir volume permitted 3 or 4 tests to be run off before recharging and reheating.

The capabilities of the experimental arrangement as described above and tabulated in tables 2 and 3 made experiments possible over a range of operating parameters. The operating parameters can be expressed in terms of the key parameters which govern sound attenuation in two-phase flows. These key parameters were discussed in Section 2.2. Table 4 shows the range of these key parameters which can be obtained with the experimental arrangement.

Section 5

EXPERIMENTS

5.1 System Check-out Tests

A number of tests were completed to check the operation of controls and instrumentation, to measure the performance of the sound generator and recording equipment, and to determine the optimum run time for the test apparatus.

It was found that the acoustic termination was not effective. Standing waves were observed along the test section . The measurement of sound attenuation in the test section still was possible despite the standing waves because the sound pressure level at successive antinodes is a function of the attenuation coefficient.

Table 5 shows the predicted number of standing waves along the test section; the sound records agree within 1%. The data in Table 5 shows that, at frequencies higher than 850 H_z, the half-wavelength is less than the duct diameter so cross-mode vibration will develop. Therefore, at frequencies higher than 850 Hz, the microphone should be traversed over the cross-sectional area of the test section so that cross-mode effects can be taken into account when determining the average sound pressure level. Cross-traverse capability was not incorporated in the present system.

The accuracy limits for the measured data are estimated to be as follows:

Temperature	=	$\pm 2^{o}F$
Pressure	=	$\pm$ 0.05 in. H_2O
Flow rate	=	$\pm$ 3%
Frequency	=	$\pm$ 1%
Time	=	$\pm$ 0.2 sec.

5.2 Air-Water Tests

The conditions for the air-water tests and the associated key parameters are given in Table 6. The first two tests were completed using the largest droplet size, 1900μm, the maximum and minimum mass fraction, and sound frequency f = 200 Hz. Tests 1 - 2 to 1 - 5 are a set of tests for ambient initial conditions with maximum droplet size and maximum liquid mass fraction. In test 1 - 6, the background noise was recorded with the complete system operating without sound.

Tests 2-1 to 2-4 were completed with the smallest liquid drop size possible with the present liquid spray method, 190 μm. The air and water were at ambient initial conditions for these tests.

Test 2-5 was a "warm-up" test performed during the period required to heat up the air and water to 80°F above the ambient temperature. Tests 2-6 and 2-7 were not fully recorded because the microphone carriage rubbed against the wall of the test section. The rubbing was caused by thermal distortion of the carriage frame during the warm up period.

Tests 2-8 to 2-11 were conducted at elevated initial water and air temperatures. The liquid flow rate during these tests decreased from 1.0 gm to 0.2 gpm due to plugging of 80% of the holes in the nozzle plate. Since a 1μm filter is installed in the water loop upstream of the nozzle box, the material which caused the plugging must have been released from the short section downstream of the filter.

Section 6

RESULTS AND DISCUSSION

The air-water test conditions are described by the records of operating conditions (Table 6) and by traces of the sound pressure level (SPL) variation along the length of the test section. A typical set of traces is shown in figure 10 for test 2-3 with and without liquid injection.

The traces show two types of changes in SPL; one is an initial decrease when the liquid injection is turned on, and the other is the variation of SPL at the nodes and antinodes of the standing waves along the test section. The initial SPL at the top microphone position is 107db for both traces. At the instant the water injection began, the sound-pressure level dropped 3 db on the upper trace. This initial SPL drop is also shown by the 3-4 db downshift of the entire trace for the traverse. The downward displacement of the trace was maintained as long as liquid injection continued.

This type of SPL change was observed at all sound frequencies used for testing. The SPL change increased with frequency from 1-2 dB at 200 Hz to 5-10 dB at 6000 Hz.

The SPL maxima and minima of the standing waves show a 1 to 2 dB reduction over the lower two-thirds of the test section when liquid is injected. This is the type of SPL variation expected for these tests, although the magnitude of attenuation is lower than expected.

The abrupt reduction of SPL just downstream of the injection nozzle when the droplets are introduced may be due to one or more of the following reasons:

1. The SPL produced by the acoustic driver may be lower for the two-phase (air-water) flow due to the difference in acoustic impedance.

2. Sound energy may be absorbed by the jet break-up process. The liquid leaves the holes of the nozzle plate in the form of continuous jets, but due to surface tension instability the

jets break up into uniform droplets. Liquid jets exposed to acoustic vibration may break up in a shorter length, in which case the work done by the acoustic waves against surface tension can result in absorption of acoustic energy.

3. The microphone sensitivity may change under the 100% R.H. condition. The microphone is properly protected from liquid drops, and it is rated for 100% R.H. operation. However, it is still possible that a change in sensitivity is experienced at this entrance relative humidity condition.

These comments show that the measured reduction in sound level at the upper end of the duct may have been due to attenuation or to changes in equipment operation when droplets were introduced. If attenuation indeed occurred, the major portion of the attenuation due to liquid injection took place over an axial distance of 18 inches between the acoustic driver and the top position of the traversing microphone. Since a sort of "near field effect" was expected in the vicinity of the acoustic driver connection to the test section, the top position of the microphone was selected to be about two test section diameters away from the driver connection. Therefore, more detailed measurements of the noise reduction within the 18" section could not be made.

The additional attenuation measured along the length of the test sections is almost negligible for all cases tested. The limited number of tests performed in the test program were not run at conditions which were optimized for maximum attenuation.

Figure 11 summarizes the test conditions in relation to the analytically-predicted peak attenuation conditions. In this figure, the combination of sound frequency and liquid drop diameter is shown. The three rectangular regions outlined with dashed lines show the test conditions originally selected.

In order to assure uniform droplet distribution, special nozzle plates had to be used as discussed in Section 4.2; these nozzle plates set 190μm as the minimum drop size for our tests. The circle symbols in figure 11 show the test conditions with the present liquid nozzle plates.

When the liquid drops are strongly evaporating, the diameter of the drops decreases along the test section. Thus, in figure 11, local conditions are defined by a horizontal line starting at the initial drop diameter and extending to the left, ending at the diameter at which the drops leave the test section.

The predicted regions of peak attenuation are shown in figure 11 by dash-dot and continuous lines. These predictions are based upon the analytical results of references C8 and W3 for the air-water two-phase flow. Although the present experimental conditions do not match all of the assumptions made in these references, it is expected that the peak attenuation should take place in the general area of the predictions.

Figure 11 shows that the initial diameter of the drops used in the test program is larger than optimum. The rate of evaporation is too low to achieve complete evaporation within the test section. Therefore, the conditions for peak attenuation were not achieved within the length of the test section.

The analytical methods of reference C8 and W3 were used to predict the attenuation which could be expected to occur in the test system if it were filled throughout its length with droplets of optimum size and mass loading. The predicted maximum attenuation achievable is 4-6 db. Additional attenuation may be possible as a result of the variation in drop size and mass loading which occurs in the test rig as the droplets are carried down the duct and evaporate. Almost no attenuation was achieved along the length of the duct in our tests, probably because the drops tested were too large and the evaporation rates were too small.

The results obtained to date are inconclusive for determining the maximum sound attenuation which can be obtained by liquid injection. Additional tests, employing smaller initial drop diameters and more highly evaporating liquids, should be performed to cover a wider range of conditions and to bracket the expected peak attenuation regions. The results of such tests employing optimized conditions can define the magnitude of sound attenuation possible by evaporating liquid drops and can provide information for practical application of this noise reduction technique to gas turbine engines.

APPENDIX

Correlation Parameters for Sound Attenuation Coefficients in Two-phase, Gas-liquid Flow

- Reference C8 (Cole and Dobbins) used the following parameter to describe the attenuation coefficient.

$$\bar{\tau} = \frac{\omega \tau_T}{C_m}$$

A value of $\bar{\tau}$ near 1 is predicted to yield maximum attenuation.

$\omega = 2\pi f$	-	circular acoustic frequency
$\tau_T = \dfrac{d^2}{12\,\alpha_T} \dfrac{\rho_L}{\rho_o}$	-	thermal relaxation time
$C_m = \dfrac{d^3\pi}{6} \dfrac{C_n \rho_L}{\rho_o}$	-	liquid mass fraction

ρ_L - liquid density

ρ_o - mixture density

C_n - number of particles per unit volume

α_T - thermal diffusion coefficient

d - liquid drop diameter

f - frequency

In the present investigation, the effective number of liquid drops per unit volume can be defined as:

$$C_n = \frac{\text{(No. of drops produced per second) (Residence time in test section)}}{\text{(Total volume of test section)}}$$

A practical form of $\bar{\tau}$ is:

$$\bar{\tau} = 0.332 \frac{f\,[\,Hz\,]\left(\frac{d\,[\mu m]}{100}\right)^2}{q_L\,[gpm]\;\tau_R\,[sec]} \qquad (A1)$$

- Reference W3 (Wooten) calculated an attenuation coefficient as a function of $\omega\,\tau_v$. When $\omega\,\tau_v$ is near 1, Wooten predicts the attenuation to be a maximum.

$\omega = 2\pi f$ - circular acoustic velocity

$\tau_v = \frac{d^2}{18\,\nu}\;\frac{\rho_L}{\rho}$ - velocity relaxation time

d - drop diameter

ν - kinematic viscosity of gas

ρ_L - liquid density

ρ - gas density.

REFERENCES

A1 Ardans, P. M., and Stephenson, D. W., "An Analytical Method for Estimating the Performance of a Gas Turbine Engine with Water-Methanol Injection" SAE-Paper 7002DV, March 1970

A2 Arnold, L., and Slutsky, S., "Attenuation of Plane Waves of Sound by Suspension of Resonating Particles" NASA SP-207, July 1968

B1 Barilovich, V.A. et al, "Using Suspended Moisture for Cooling High-Temperature Gas Turbines," Air Force Systems Command Report FTD-MT-24-147-69

C1 Chow, J. C. F., "Attenuation of Acoustic Waves in Dilute Emulsions and Suspensions," J. Acoust. Soc. Amer. 36, 2395 - 2401 (1964)

C2 Chu, B. T., and Chow, J. C. F., "On a Macroscopic Theory of a Two-Phase Fluid," Paper No. 65-8, AIAA 2nd Aerospace Sciences Meeting, New York, January 1954 (unpublished)

C3 Carrier, G. F., "Shock Waves in a Dusty Gas," J. Fluid Mech. 4, 376 - 382 (1955)

C4 Courtney, W. G., "Kinetics of Condensation of Water Vapor," Chem. Phys. 36, 2018 - 2025 (1962)

C5 Carslaw, H. W., and Jaeger, J. C., Conduction of Heat in Solids Oxford University Press, London (1959) p. 112

C6 Cole, J. E., and Dobbins, R. A., "Propagation of Sound Through Atmospheric Fog," J. of Atmospheric Science 27 (3), (May 1970)

C7 Cole, J. E. et al, "Time Resolved Measurement of Droplet Size and Concentration in Cloud Chambers," J. Applied Meteorology 9 (4), (August 1970)

C8 Cole, J. E., and Dobbins, R. A., "Measurements of the Attenuation of Sound by a Warm Air Fog, " (to be published)

D1 Dobbins, R. A., and Temkin, S., "Propagation of Sound in a Gas-Particle Mixture and Acoustic Combustion Instability," AIAA Journal 5 (12)

E1 Epstein, P. S., "On the Absorption of Sound in Suspensions and Emulsions," Contributions to Applied Mechanics, Theodore von Karman Anniversary Volume, California Institute of Technology, Pasadena (1941) p. 162 - 188

E2 Epstein, P.S., and Carhart, R.R., "The Absorption of Sound in Suspensions and Emulsions: 1. Water Fog in Air," J. Acoust. Soc. Amer. 25 (3), 553 - 64 (May 1953)

G1 Goldman, E., "Absorption and Dispersion of Ultrasonic Waves in Inert Monatomic Gas Mixtures," J. Acoust. Soc. Amer. 44, 708 - 713 (1968)

G2 Goldman, E., "Absorption and Dispersion of Ultrasonic Waves in Mixtures Containing Volatile Particles," J. Acoust. Soc. Amer. 47 (3), 768 - 776 (1970)

G3 Goldman, E., and Sirovich, L., "Equations for Gas Mixtures," Phys. Fluids 10 (9), (September 1967)

H1 Harris, C.M., "Absorption of Sound in Air Versus Humidity and Temperature," NASA CR-647, January 1967

H2 Harris, C.M., "Absorption of Sound in Air in the Audio-Frequency Range," J. Acoust. Soc. Amer. 35 (1)

H3 Hirth, J.P., and Pound, G.M., "Condensation and Evaporation, Nucleation and Growth Kinetics," Progress in Materials Science 11, Macmillan, New York (1963)

K1 Knudsen, V.O. et al, "The Attenuation of Audible Sound in Fog and Smoke," J. Acoust. Soc. Amer. 20 (6), 849 - 857 (November 1948)

K2 Kinsler, L.E., and Frey, A.R., Fundamentals of Acoustics, John Wiley & Sons, New York (1967)

K3 Kurbjun, M.C., "Limited Investigation of Noise Suppression by Injection of Water into Exhaust of After-Burning Jet Engine," NACA RML57L05 (February 1958)

K4 Knudsen, M., "Die maximale verdampfungsgeschwindigkeit des Quicksilbers," Ann. Phys. 47, 697 (1915)

L1 Lamb. H., Hydrodynamics, Dover Publications, New York (1945) p. 656 - 661

L2 Liu, J.T.C., "Some Comments on the Use of Additives (Small Solid Particles) in the Jet Noise Problem," NASA SP-207, July 1969

M1 Marble, F.E., "Some Gasdynamic Problems in the Flow of Condensing Vapors," Astronaut. Acta. 14, 585 - 611 (1969)

M2 Marble, F.E., "Dynamics of a Gas Containing Small Solid Particles," Proceedings of the Fifth AGARD Combustion and Propulsion Colloquium, Pergamon Press, New York (1963)

M3 Marble, F.E., "Gas Dynamics of Condensing Vapors," U.S. Air Force Aerospace Research Laboratory Report - ARL 69-0040 March 1969

M4 Marble, F.E., and Wooten, D.C., "Sound Attenuation in a Condensing Vapor," Daniel and Florence Guggenheim Jet Propulsion Center, California Institute of Technology, Pasadena, California

O1 Oswatitsch, K., "Die Dispersion und Absorption des Schalles in Wolken," Phys. Z. 42, 365 (1941)

P1 Porter, J.W., "The Interaction of Acoustic Waves with Solid-Gas and Liquid-Vapor Interfaces," to be published in J. Acoust. Soc. Amer.

R1 Richards, F.J., "Research on Aerodynamic Noise from Jets and Associated Problems," J. Aero. Soc. 57 (509), 318 - 342 (May 1953)

S1 Sewell, C.J.T., "The Extinction of Sound in a Viscous Atmosphere by Small Obstacles of Cylindrical and Spherical Form," Phil. Trans. Roy. Soc. London (A) 210, 239 (1910)

T1 Temkin, S., "Attenuation and Dispersion of Sound by Particulate Relaxation Processes," Brown University Technical Report, February 1966

T2 Temkin, S., and Dobbins, R.A., "Measurements of Attenuation and Dispersion of Sound by an Aerosol," Brown University Technical Report, May 1966

T3 Temkin, S., and Dobbins, R.A., "Attenuation and Dispersion of Sound by Particulate Relaxation Processes," J. Acoust. Soc. Amer. 40, 317 - 324 (1966)

T4 Tyler, J.M., and Perry, E.D., Jet Noise Report Number PWA Inst. 451, Pratt and Whitney Aircraft 1954

U1 Urick, R.J., "The Absorption of Sound in Suspension of Irregular Particles," J. Acoust. Soc. Amer. 20 (3), 283 - 289 (1948)

W1 Watters, B.G. et al, "Acoustical Behavior of Some Engine Test Cell Structures," J. Acoust. Soc. Amer. 27 (3), 449 - 456

W2 Wisniowski, H.V., "Radial Grids with Water Injection for Suppressing the Noise of Jet Engines," Can. Aero. J. (January 1961)

W3 Wooten, D.C., "The Attenuation and Dispersion of Sound in a Condensing Medium," Ph.D., Thesis 1967, California Institute of Technology, Eng. Mech. Department

Z1 Zink, J.W., and Delasso, L.P., "Attenuation and Dispersion by Solid Particles Suspended in a Gas," J. Acoust. Soc. Amer. 30, 765 - 661 (1958)

Table 1

List of Key Parameters and Conditions Governing Sound Attenuation in Two-Phase Flow

1. Mode of Interaction between Phases or Method of Dissipation

- viscous - friction
- thermal - heat transfer
- phase change - mass transfer, evaporation, condensation

2. Particle Definition

- rigid (solid)or deformable (liquid)
- fixed in space or free to move
- shape is spherical, cylindrical, or other
- all particles are the same size
- number of particles per unit volume is constant or variable
- particle distribution is homogeneous
- phase change periodical and/or steady
- volume of particles is negligible or not negligible
- mass of particles is/is not significant relative to the mass of gas per unit volume.

3. Flow of Gaseous (vapor) Phase

- no flow
- subsonic flow
- supersonic flow

4. Interaction of the Acoustic Field and the Particles

- continuum treatment - particle diameter is much smaller than the wavelength
- single particle treatment
- ratio of particle diameter to gas displacement amplitude due to sound waves
- ratio of particle diameter to particle displacement amplitude.

Table 2

Nozzle Plate Geometry and Operating Conditions

Liquid	Droplet Size	Hole Size	Number of Holes	Design Conditions Flow Rate	Design Conditions Plenum Pressure	Jet Velocity	Flow Rate Range
	microns	inches		gpm		fps	gpm
Water	1900	0.040	100	5.17	1.9 psig	13.2	0.5 - 6.0
Water	642	0.0135	170	5.04	50.0 psig	66.5	0.5 - 6.0
Water	190	0.004	400	1.03	58.0 psig	65.5	0.5 - 1.36
Freon 11	1900	0.040	50	0.635	3.2"Freon	3.25	0.5 - 0.715
Freon 11	642	0.0135	170	0.635	1.18"Freon	8.4	0.414 - 1.1
Freon 11	190	0.004	400	0.635	32.7" Freon	40.5	0.3 - 0.93
Liquid N_2	1900	0.040	25	0.435	6.07"Liq. N_2	4.46	0.435 only
Liquid N_2	642	0.0135	80	0.435	1.35" LN_2	12.22	0.35 - 0.435
Liquid N_2	190	0.004	400	0.435	8.28" LN_2	27.7	0.22 - 0.435

Table 3

Capabilities of Test Rig

Gas System	
Air:	
Range of Flow Rate	50 - 250 cfm
Temperature Range	70 - 180°F
Freon Gas: (R-11)	
Range of Flow Rate	25 - 50 cfm
Temperature Range	70°- 75°F
Liquid System	
Water:	
Range of Flow Rate	0.5 - 6 gpm
Temperature Range	70°~ 180°F
Freon Liquid: (R-11)	
Range of Flow Rate	0.3 ~ 1.1 gpm
Temperature	75°F
Nitrogen Liquid:	
Range of Flow Rate	0.2 ~ 0.4 gpm
Temperature	-320°F
Droplet Sizes :	
	190μm
	642μm
	1900μm

Table 4

Key Parameters for Two-Phase Systems

	Air-Water	Air-Freon	Air-Liquid Nitrogen	Freon-Freon
Mass Fraction	22 ~ 1300%	20 ~ 360%	14 ~ 70%	20 ~ 30%
Volume Fraction	.03 ~ 1.6%	.02 ~ 0.3%	.1 ~ 0.3%	.08 ~ 0.3%
Temp Difference	0 ~ 100°F	0 ~ 100°F	490°F	0°F
Latent Heat Factor	23 ~ 29	2.2	8.9	10.2
Density Ratio	830	1230	670	280
τ_V, sec.	0.1 ~ 10	.14 ~ 14	0.2 ~ 20	.27 ~ 27
τ_T, sec.	0.1 ~ 20	.16 ~ 16	.09 ~ 9	.27 ~ 27
τ_E, sec.	6 ~ 700	.7 ~ 70	.09 ~ 9	∞
τ_R, sec.	0.4 ~ 2.0	0.6 ~ 3.0	0.3 ~ 1.5	0.3 ~ 1.2

Table 5

Acoustic Standing Waves in the Test Section

$D = 8''$ diameter of test section

$L_m = 105''$ length of microphone traverse

$\lambda_o = \frac{13500}{f}''$ wavelength in air at 68^oF

f	λ_o	L_m/λ_o	$2D/\lambda_o$
H_z	in	-	-
200	67.5	1.56	0.24
600	22.5	4.66	0.71
750	18.0	5.84	0.89
850	15.9	6.60	1.00
950	14.2	7.40	1.13
1000	13.5	7.78	1.19
2000	6.75	15.60	2.37
6000	2.25	46.6	7.11
10,000	1.35	77.8	11.87

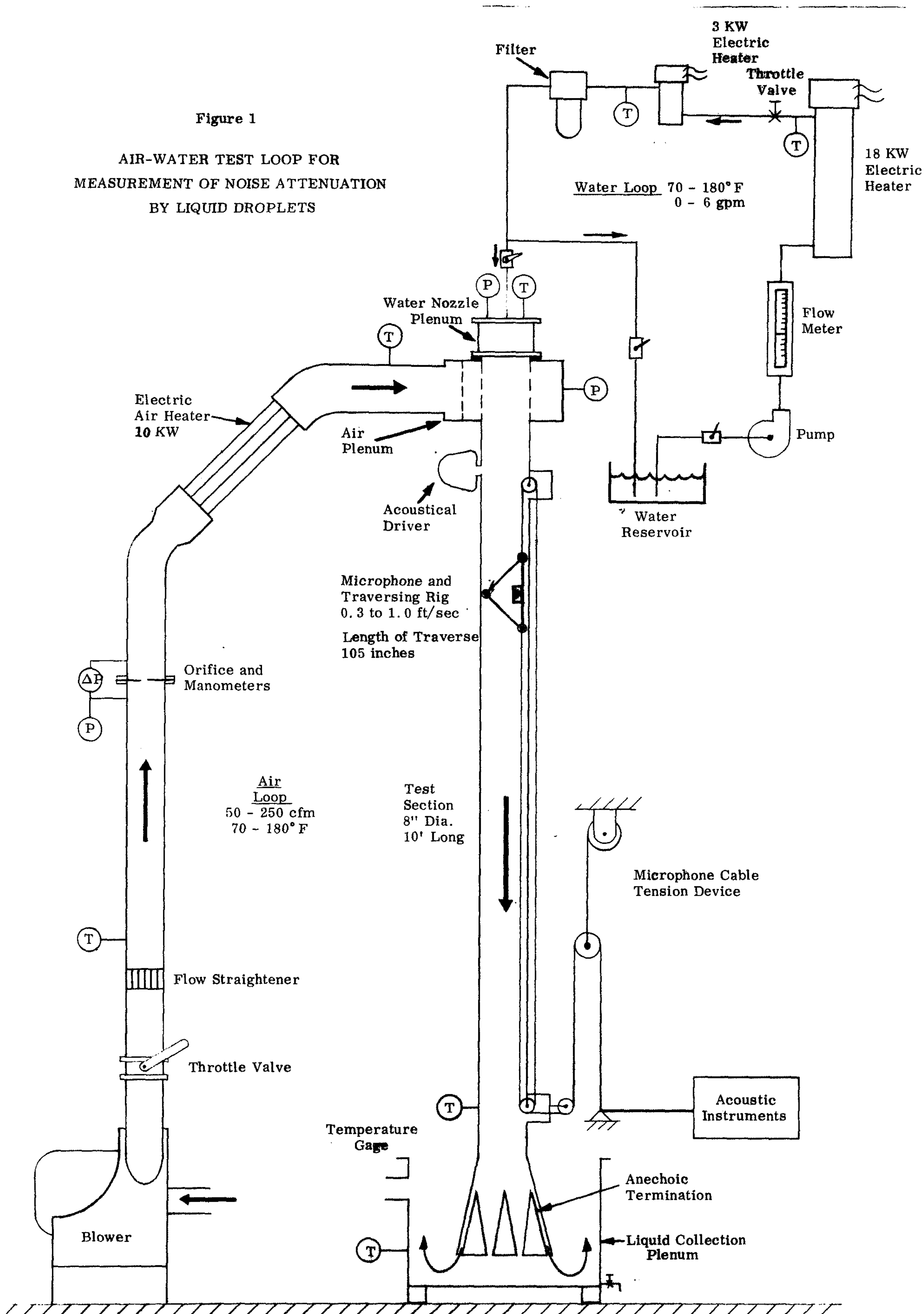
Table 6

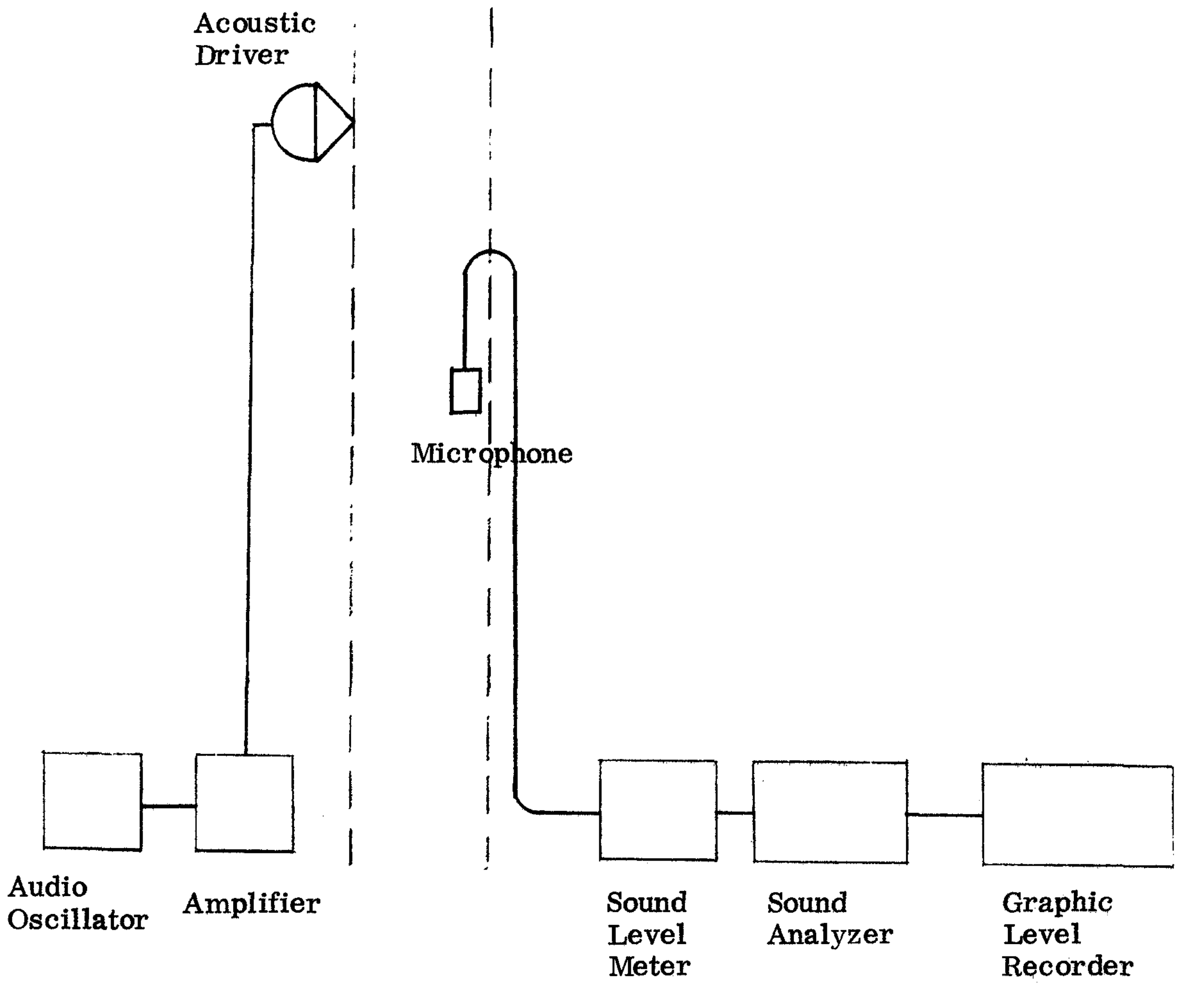
Air Water Test Conditions

Test No.	Droplet Dia.	Initial Air	Cond. Liquid	Frequency of Sound				q_L	Q	m_L/m_v	τ_v	τ_T	τ_R
	μm	^oF	^oF	Hz	Hz	Hz	Hz	gpm	cfm	-	sec	sec	sec
1-1	1900	88	68	200				5.1	250	.22	10.9	11.2	0.4
1-2	1900	83	70	200				5.1	57	10.3			0.7
1-3	1900	84	65			850		5.1	57	10.3			
1-4	1900	87	65		600			5.1	57	10.3			
1-5	1900	89	66				6000	5.1	57	10.3			
1-6	1900	90	69					5.1	57	10.3			
2-1	190	80	70	200				1.00	50	2.2	.11	.12	2.0
2-2	190	84	70		600			0.95	50	2.1			
2-3	190	87	71			850		0.90	50	2.0			
2-4	190	90	72				6000	0.90	50	2.0			
2-6	190	-	-	200				.80	57	-			
2-7	190	-	-	200				.75	57	-			
2-8	190	157	140	200	200			.3	47	0.8	0.9	.11	
2-9	190	160	150	200				.2	47	0.5			
2-10	190	162	152		600			.2	47	0.5			
2-11	190	164	157			850		.2	47	<0.5			

Figure 1

AIR-WATER TEST LOOP FOR
MEASUREMENT OF NOISE ATTENUATION
BY LIQUID DROPLETS





(Sound analyzer is centered at the audio oscillator frequency using 1/3 octave band filter bandwidth.)

Figure 2

SCHEMATICS OF ACOUSTIC EQUIPMENT

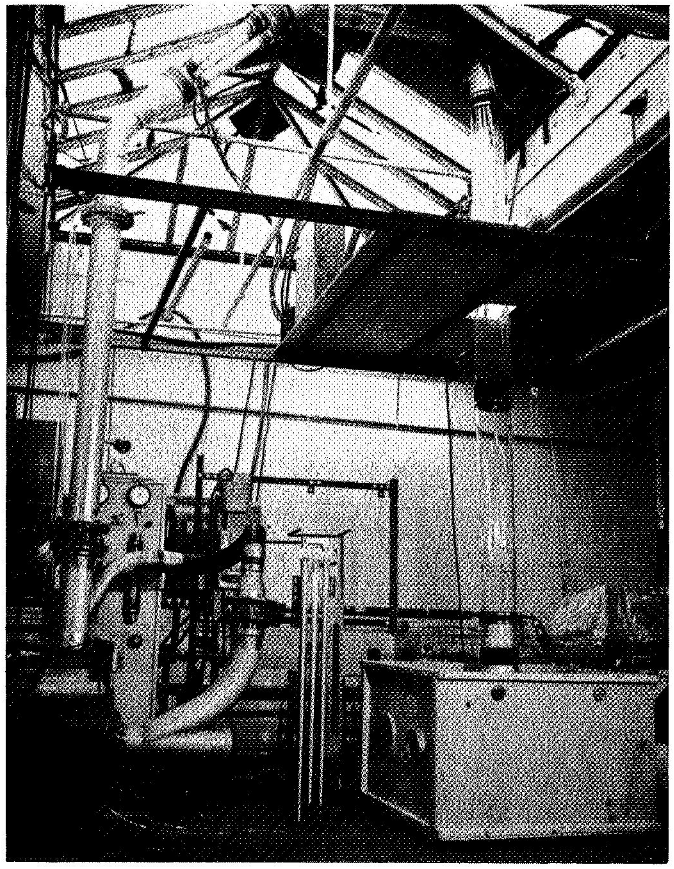


Figure 3

AIR LOOP, OPEN LOOP ARRANGEMENT

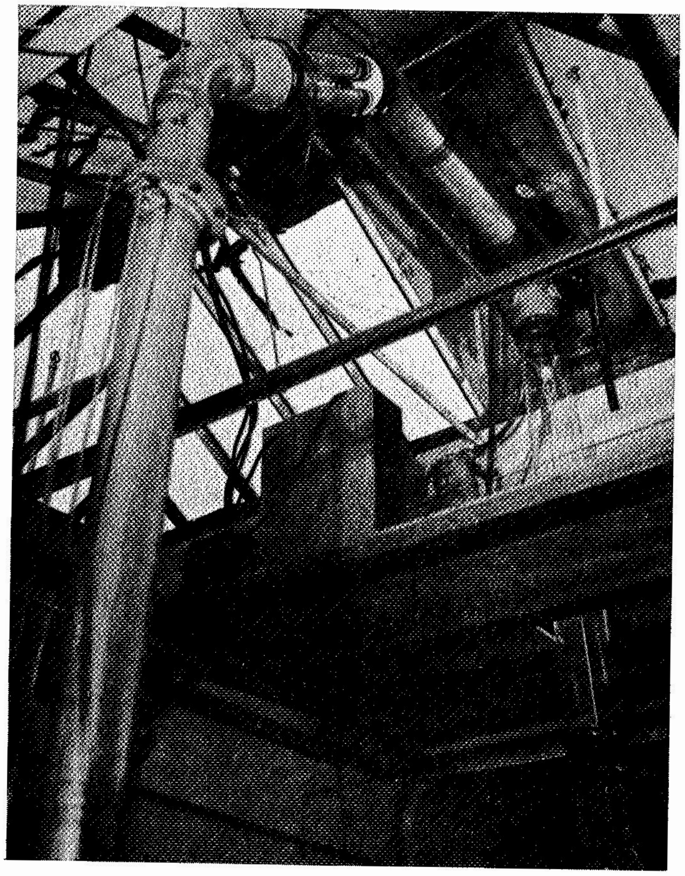


Figure 4

AIR METERING ORIFICE, AIR HEATER, AIR PLENUM, AND UPPER HALF OF TEST SECTION

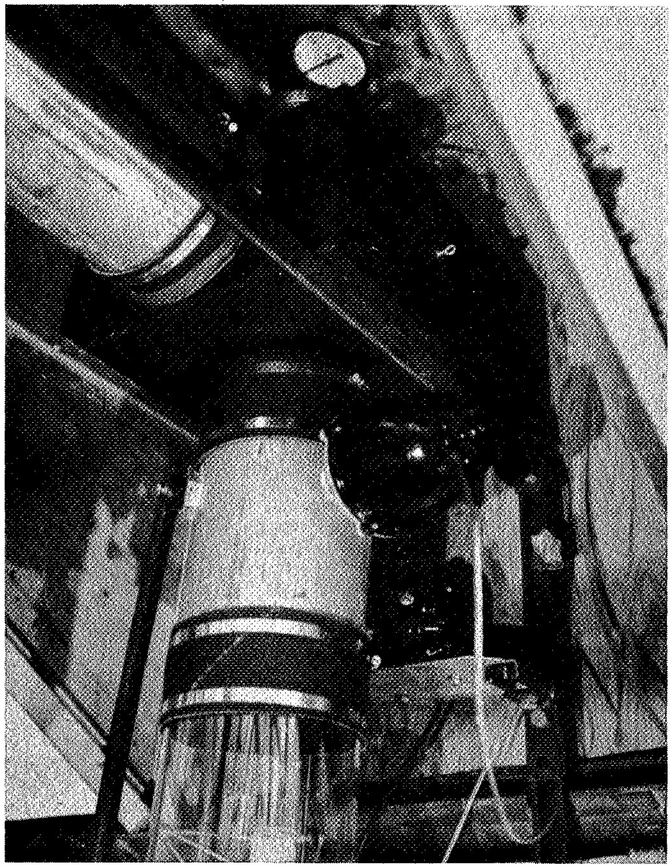


Figure 5

AIR PLENUM, ACOUSTIC DRIVER, AND
MICROPHONE TRAVERSE MOTOR

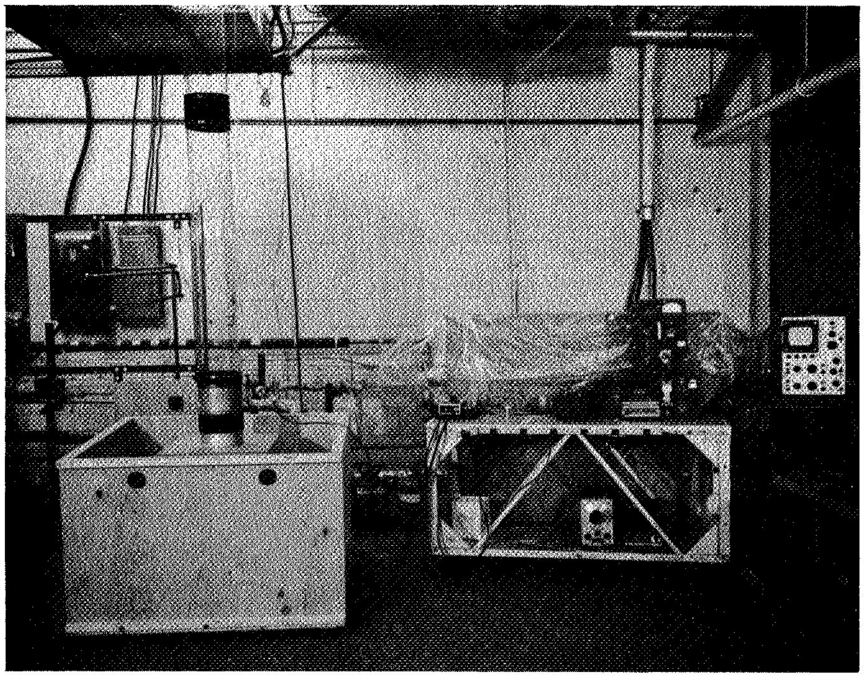


Figure 6

LIQUID COLLECTION PLENUM,
LOWER HALF OF TEST SECTION, AND
ACOUSTIC EQUIPMENT

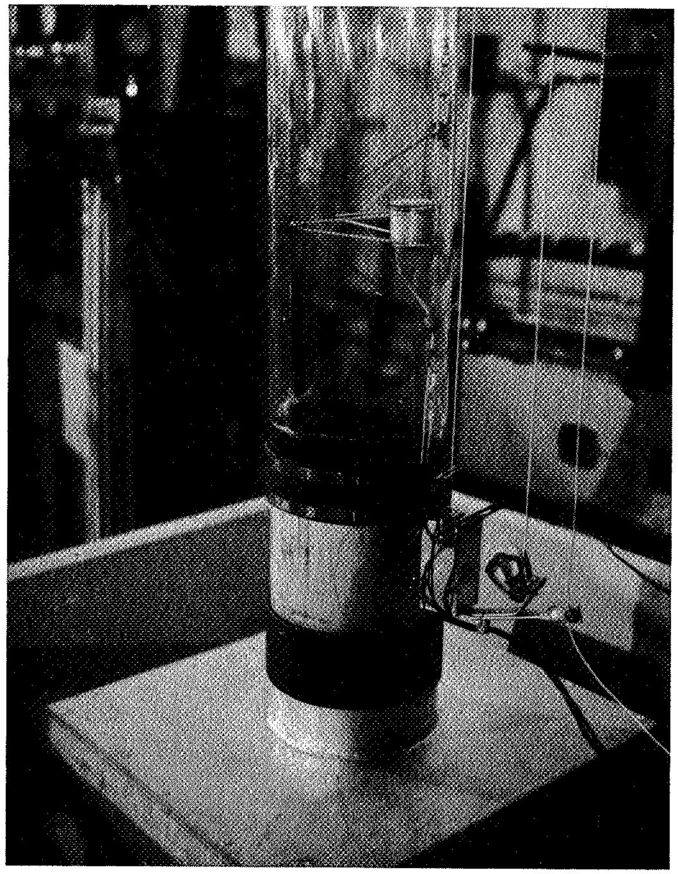


Figure 7

MICROPHONE AND TRAVERSING RIG

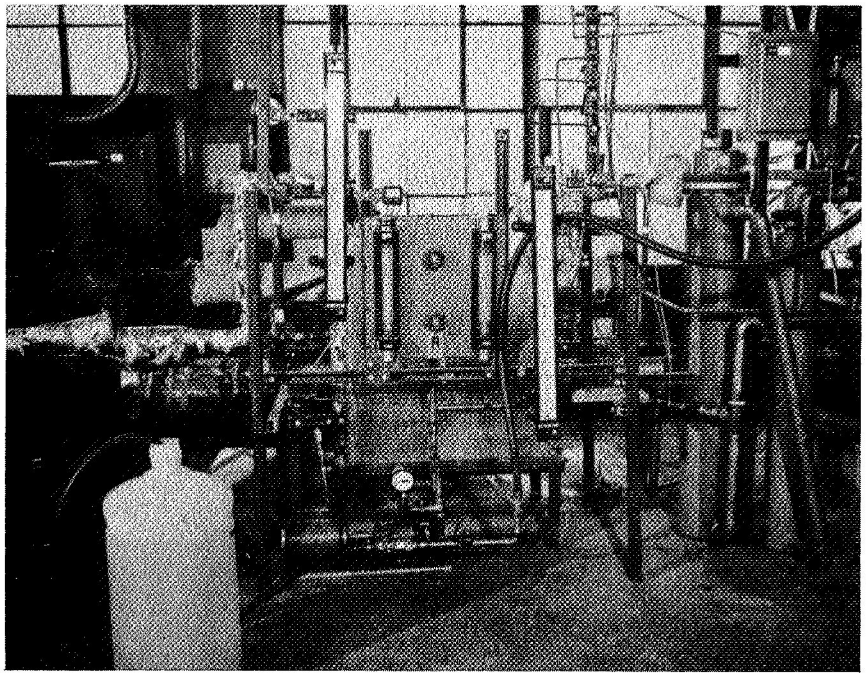


Figure 8

WATER LOOP, PUMP,
FLOW METERS, AND WATER HEATERS

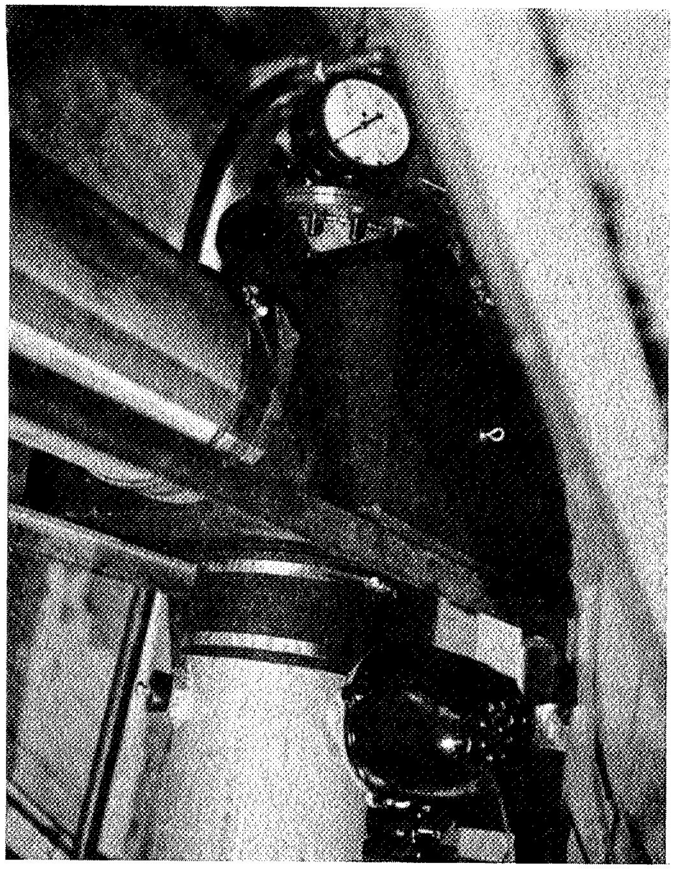


Figure 9

WATER NOZZLE PLENUM ON
THE TOP OF THE AIR PLENUM

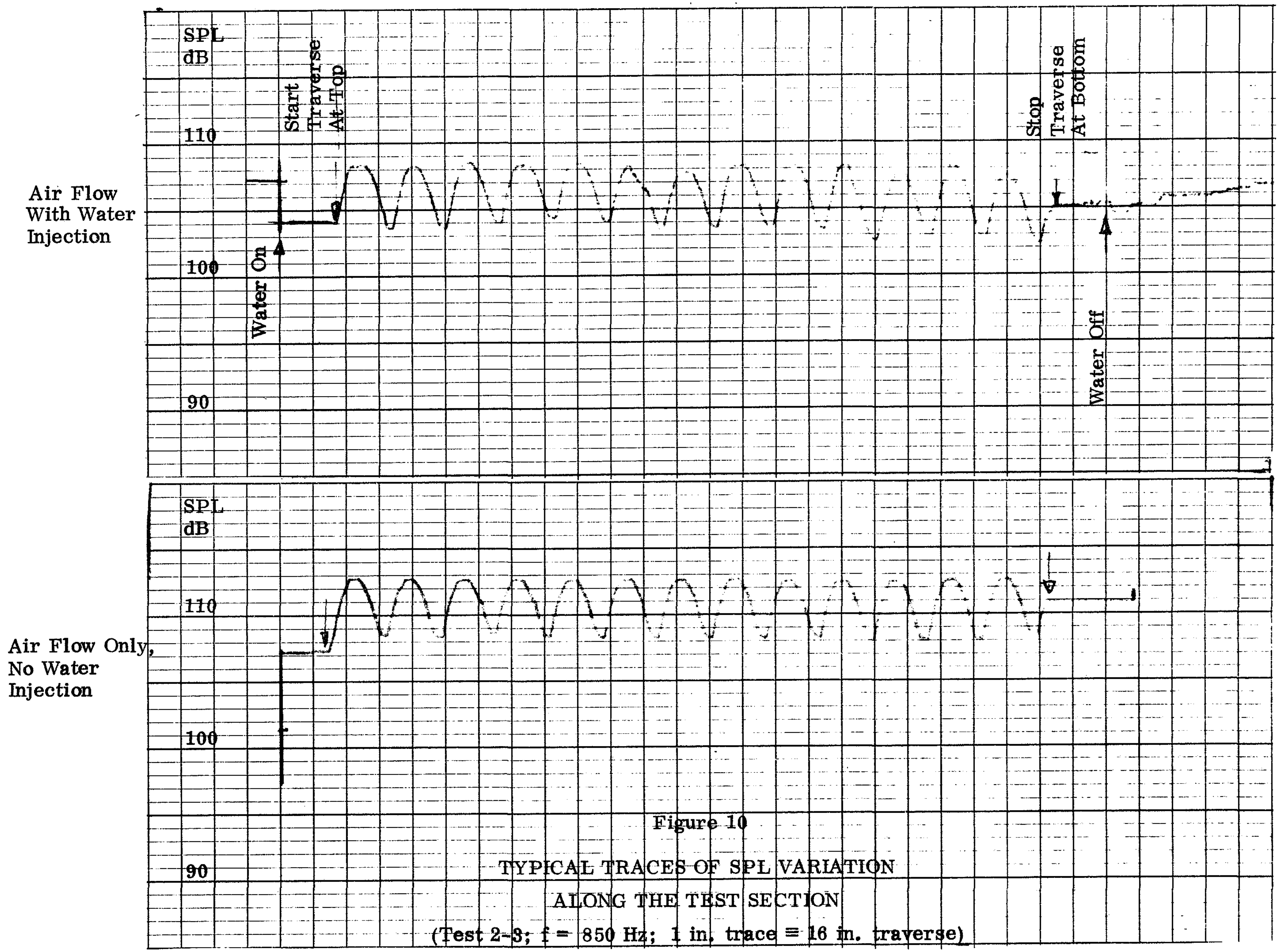
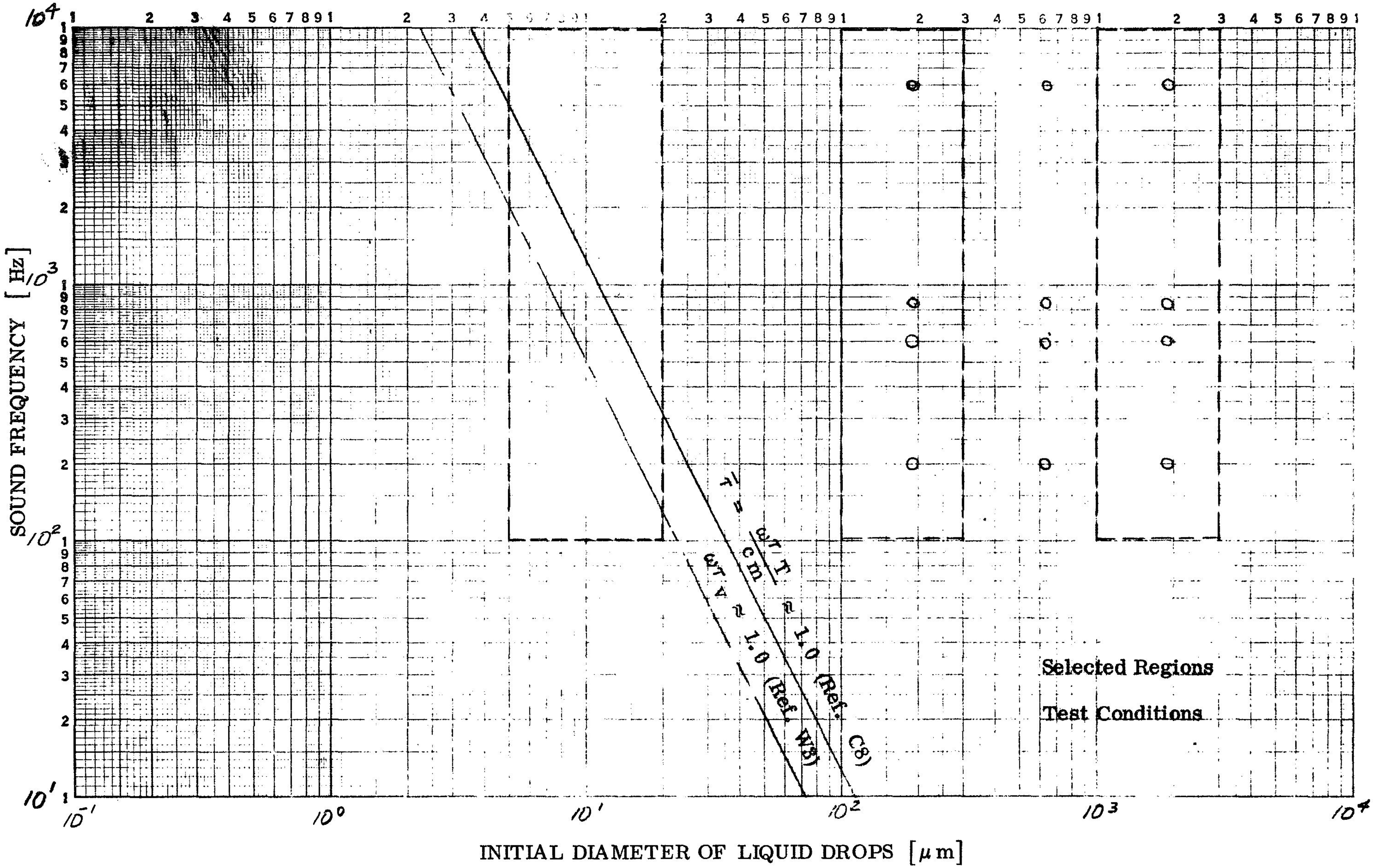


Figure 10

TYPICAL TRACES OF SPL VARIATION
ALONG THE TEST SECTION
(Test 2-3; f = 850 Hz; 1 in. trace ≡ 16 in. traverse)

Figure 11

Regions of Sound Frequency and Particle Diameter Combinations



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