GEORGIA INSTITUTE OF TECHNOLOGY
OFFICE OF CONTRACT ADMINISTRATION
SPONSORED PROJECT INITIATION

Date: 10/7/78

Project Title: Study of Poultry Plant Noise Characteristics

Project No: A-2231

Project Director: R. A. Cassanova

Sponsor: NASA-Lewis Research Center

Agreement Period: From 10/1/78 Until ~~9/30/79~~ ~~1/1/80~~ 5/31/81

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Defense Priority Rating: None

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SPONSORED PROJECT TERMINATION

Date: 6/2/81

Project Title: Study of Poultry Plant Noise Characteristics

Project No: A-2231

Project Director: J. Craig Wyvill

Sponsor: NASA Lewis Research Center

Effective Termination Date: 5/31/81

Clearance of Accounting Charges: 5/31/81

Grant/Contract Closeout Actions Remaining:

- _ Final Invoice and Closing Documents
- x Final Fiscal Report S.F. 272
- x Final Report of Inventions
- x Govt. Property Inventory & Related Certificate
- _ Classified Material Certificate
- _ Other ______

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A-2231

Progress Report

for the period

October 1, 1978 to March 31, 1979

STUDY OF POULTRY PLANT NOISE CHARACTERISTICS AND POTENTIAL NOISE CONTROL TECHNIQUES

NASA Research Grant No. NSG3228

from the

National Aeronautics and Space Administration
Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135

by the

Georgia Institute of Technology
Engineering Experiment Station
Atlanta, Georgia 30332

R.A. Cassanova, Principal Investigator
J.C. Wyvill
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L.J. Moriarty

I. BACKGROUND

Poultry processing plants present a unique challenge in arriving at applicable noise control methods. Ordinary noise control methods such as sound absorbing and vibration damping materials cannot be applied due to the high humidity environment and the daily, rigorous cleaning with high pressure steam and detergents. Typical noise levels in poultry processing plants are known to be high and in many areas exceed the noise limits established by OSHA.

Cursory sound pressure level surveys have been performed in the past by government agencies and plant personnel. However, very little data is available on noise level contours throughout a typical plant and no detailed data is available on the frequency distribution of the noise.

The current project is aimed at providing basic data on the noise environment in poultry processing plants so that a more concise and progressive direction may be given to reducing noise levels. This report summarizes the project efforts for the period October 1, 1978 to March 31, 1979 and presents typical examples of the data which were acquired in a poultry plant.

II. DATA ACQUISITION

The measurement instrumentation setup is presented in figure 1. The system consisted of three 1/2" free field, condenser microphones mounted on an aluminum boom attached to a camera tripod. Each microphone station was equipped with a separate preamplifier and power source and simultaneous recording of each of the three stations was made via the three direct channels of a Hewlett Packard tape recorder. Wind screens were placed on the microphones to prevent moisture from shorting the system. Equipment specifications are presented in Appendix A.

The evisceration area of the plant was chosen for survey since it represents exposure to approximately 150 people/shift. Due to the extensive use of mechanization in the plant, the majority of personnel exposed in this area are situated on the two production lines running the full length of the area (see figure 3).

The evisceration area was sectionalized into a grid pattern for measurement purposes (see figure 3). Every effort was made to take readings every 3 feet. However, in a few instances, equipment and personnel blocked areas precluding the acquisition of data completely to plan. In addition, the production line spacing forced measurements to be taken in aisle areas only. This necessitated the spacing between rows to be 9 feet (see figure 1).

Data gathering required portability. Consequently, the entire measurement system was placed on a cart and powered by a 12 volt automobile battery through an A/C power inverter. A calibration tone was placed on each channel prior to data gathering and at periodic intervals to set the recorder gain and to insure consistent system response during measurement.

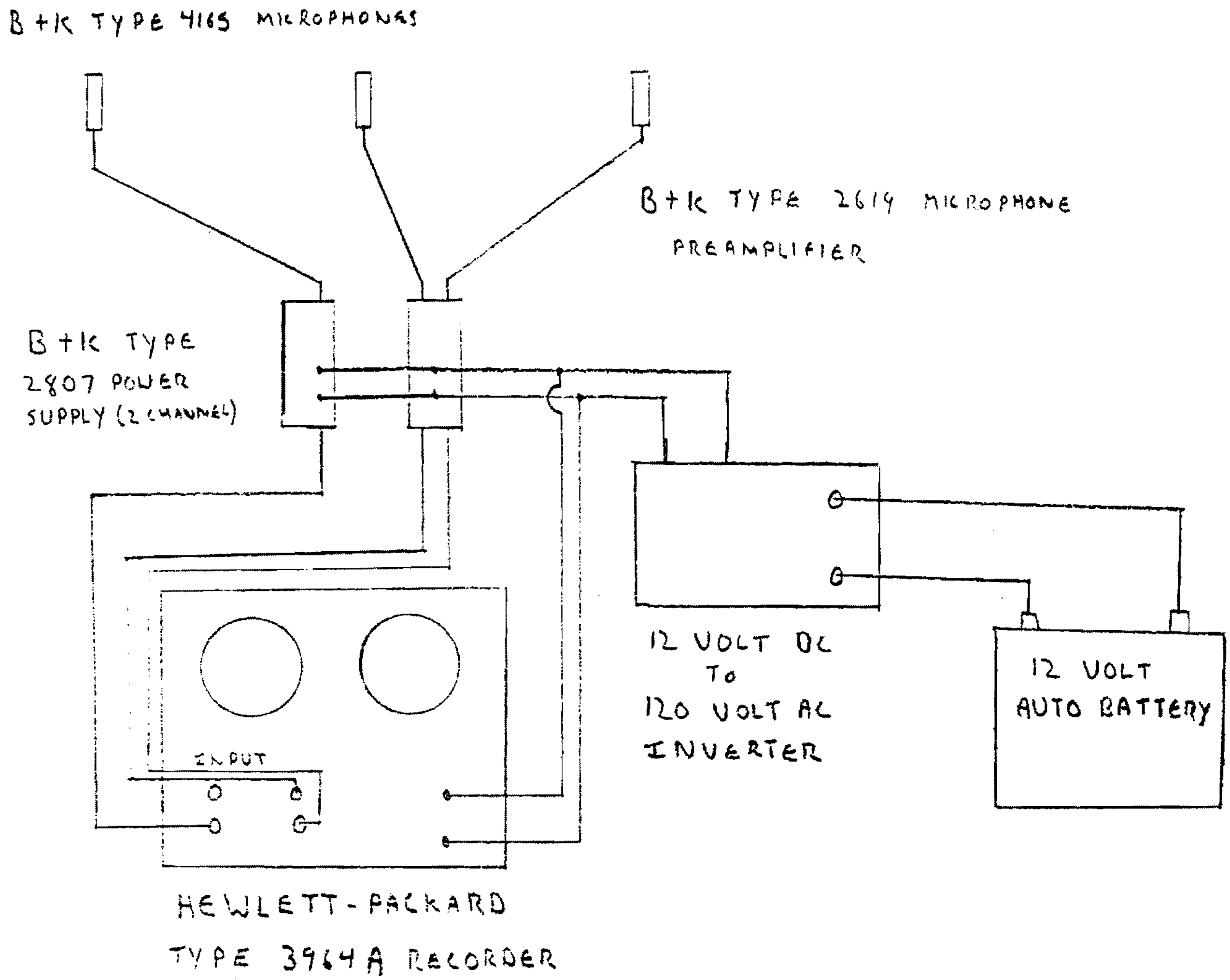
Observed Environment

The plant was composed primarily of block walls and a metal corrugated ceiling. Girder supports for the roof were boxed with metal covers and were spaced along the wall.

Plant personnel work in 8-hour shifts with 1/2 hour for lunch. The production line requires approximately 60 minutes for a bird to reach the packing area (immediately beyond the evisceration area) after it is first hung. Several minutes lapse between the hang area and entry to the evisceration area.

FIG 1

DATA RECORDING

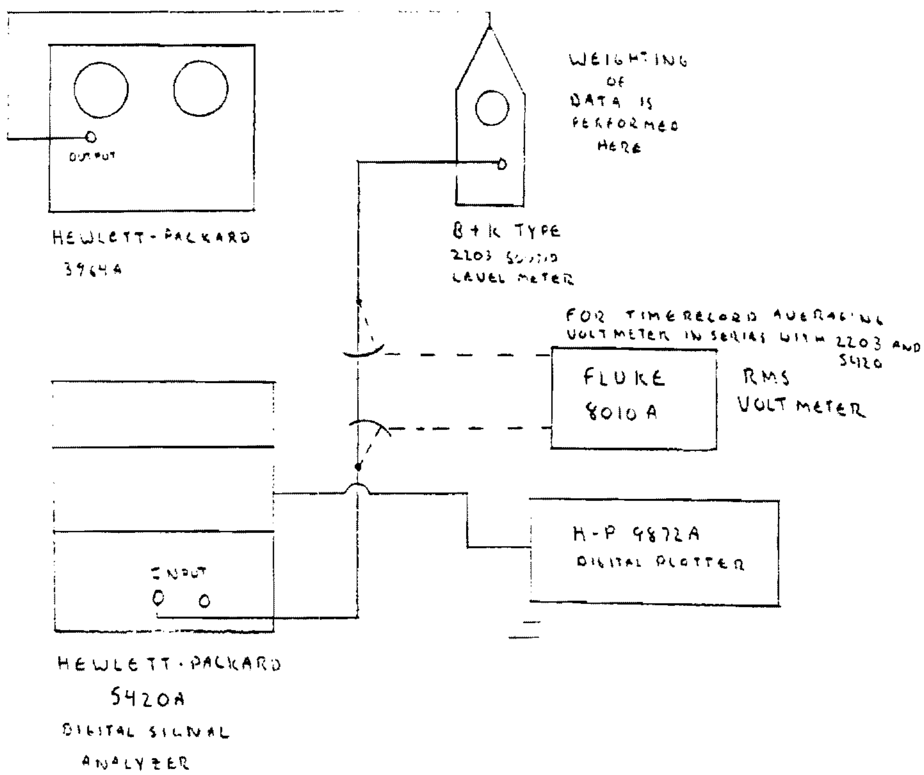


III. DATA ANALYSIS

The analysis system setup is presented in figure 2. The system consisted of the Hewlett Packard tape recorder used to gather data, a B&K type I sound level meter (for A-weighting), a Fluke RMS voltmeter for time averaging, a Hewlett Packard digital signal analyzer, and an X-Y plotter. The digital RMS voltmeter was modified to provide an analog output for compatibility with the signal analyzer. Equipment specifications for the digital RMS voltmeter are presented in Appendix B. The analyzer calibration procedure is discussed in Appendix C.

FIG. 2

DATA ANALYSIS



NOISE CONTOURS

Utilizing the 78 measurements made in the plant, noise contours were developed. All data used in this development was A weighted and time averaged. The resultant contours are presented in figure 3.

Contour lines are useful in pinpointing major noise sources. From figure 3, the four predominant noise sources present are the lung guns, the ice dump for the chiller, the circulating fan, and the conveyor portals from the picking area. Of additional interest is the localized noise buildup occuring between the gizzard inspection station and the lung guns apparently caused by the interaction of noise from the lung guns and ice dump. The entire work area seems to exhibit a "hard" reverberant quality evidenced by the slow attenuation rate occuring away from the lung gun toward the east wall.

TIME HISTORY

Six measurement points were singled out for purposes of evaluating how the noise levels in the plant varied with time. These points are identified as A,B,C,D,E, and F on figure 3. Figure 4 presents the time history of noise in the area near the lung guns (point A). Variations in noise level of 6 dbA were recorded at this station.

Figure 5 shows a relatively quiet region of the plant near the USDA trim (point B). Variations in noise level of 4 dbA were recorded at this station.

Figure 6 shows the time history of the noise buildup area between the gizzard inspection station and the lung guns (point C). Here variations in noise level of only 4 dbA were recorded.

Figure 7 shows data for the area directly in front of a circulating fan (point D). Cyclic noise variations of 7 dbA were recorded in this region.

Figure 8 (point E) shows the area near the conveyor portals from the picking area. The initial 20 seconds typifies the variation in noise level when the door to the picking area is opened. The average level rose nearly 8 dbA. Variation in noise level with the door closed reached 9 dbA.

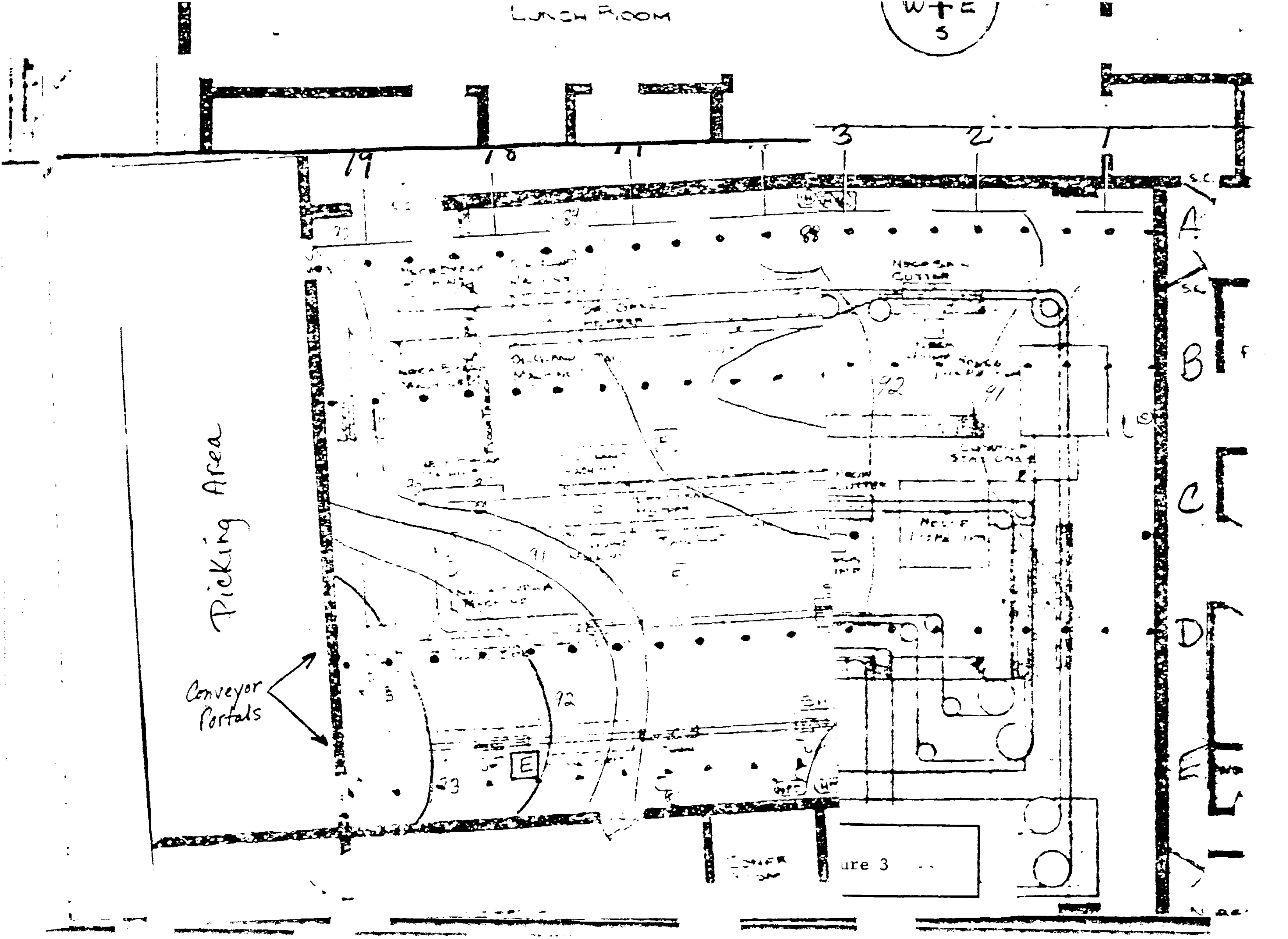
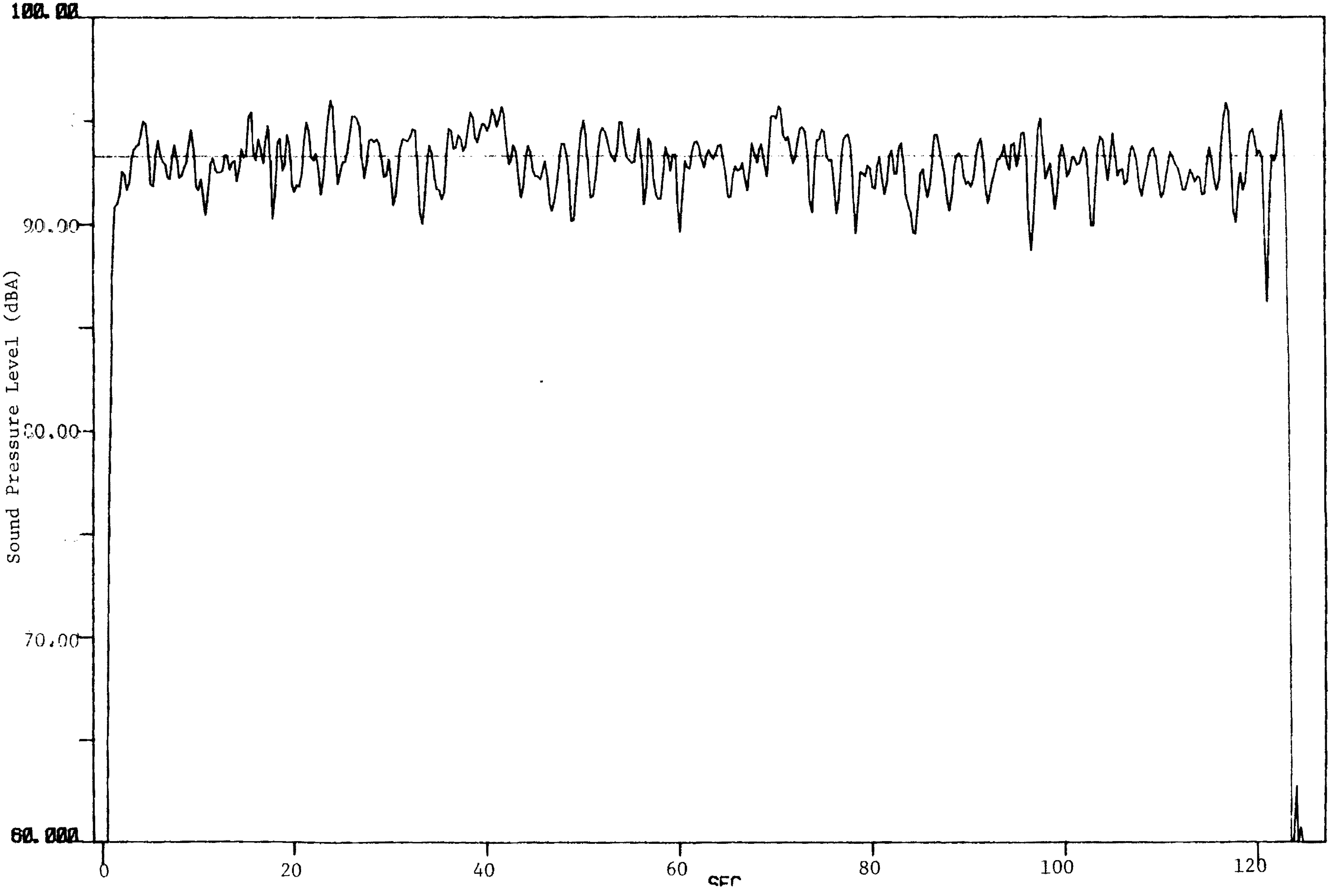


Figure 3

FIGURE 4

POINT A - 93.300 dBA



POINT B -88.000 dBA

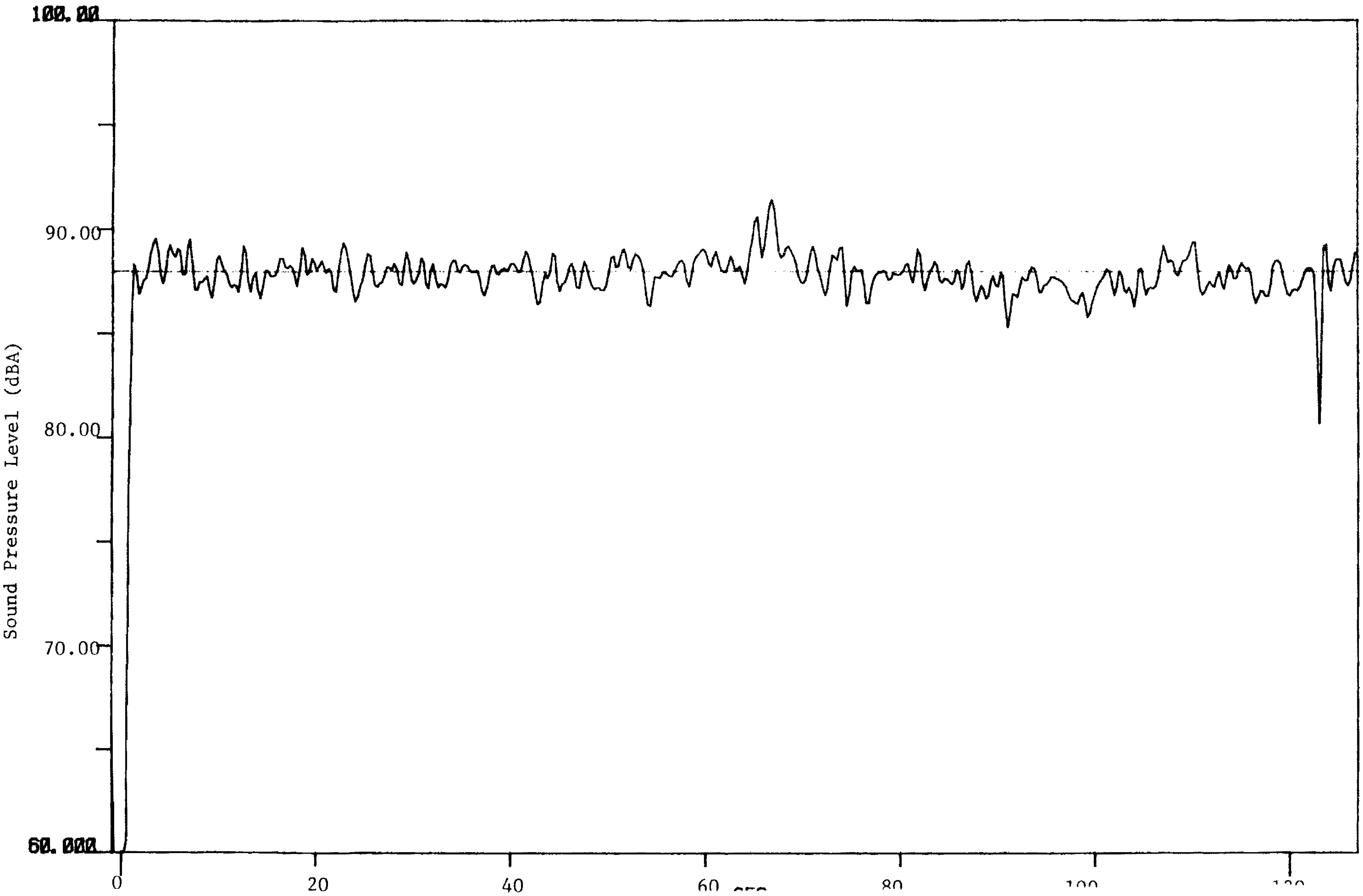
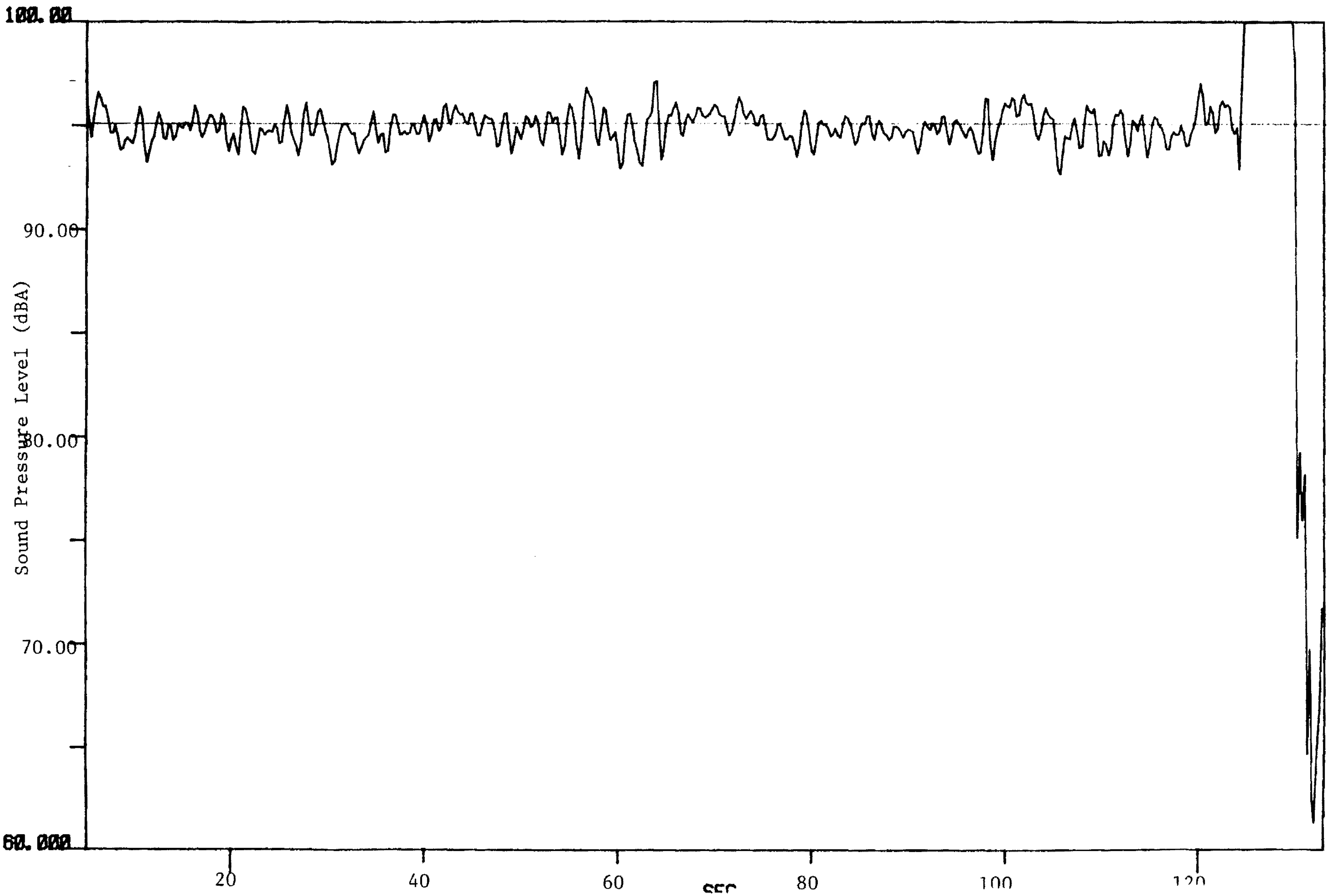
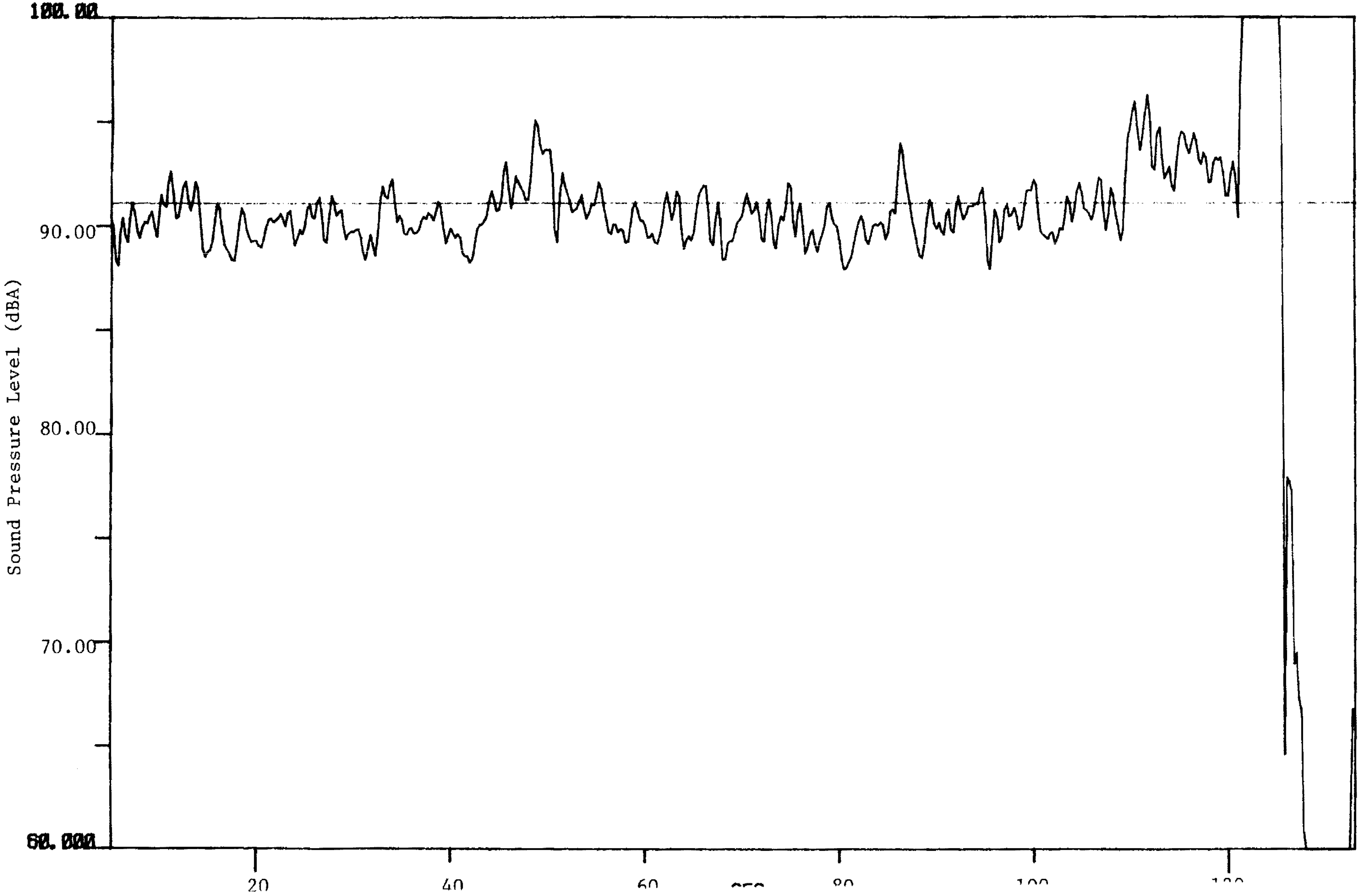


FIGURE 6

POINT C - 95.100 dBA



POINT D - 91.100 dBA



POINT E - 92.400dBA

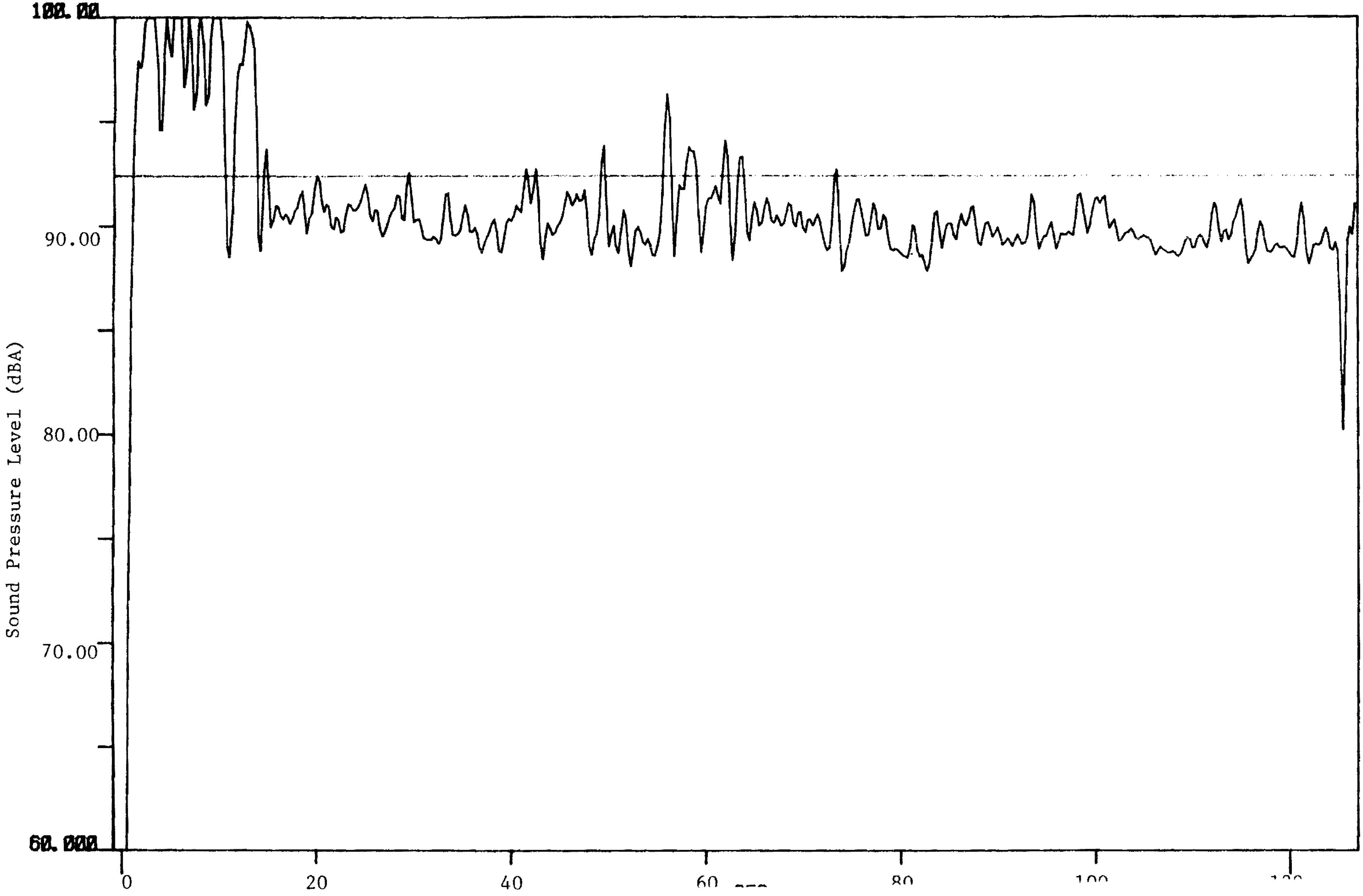


Figure 9 shows the area near the ice dump for the chiller (point F). A cyclic pattern is evident in this data reflectant of the ice dump cycle. Variations in noise level of 6 dbA were recorded at this station.

FREQUENCY ANALYSIS

The same six points used for the time history analysis were spectrum analyzed both on a "linear" and "A-weighted" basis. Figures 10 and 11 show the linear and A-weighted spectra for the lung gun area (point A). A peak at approximately 550 Hz is evident in both figures. The A-weighting network emphasizes a secondary peak at approximately 1600 Hz.

Figures 12 and 13 show the linear and A-weighted spectra for the quiet area near the USDA trim (point B). The A-weighting network tends to emphasize primary and secondary peaks at 1000 and 600 Hz respectively.

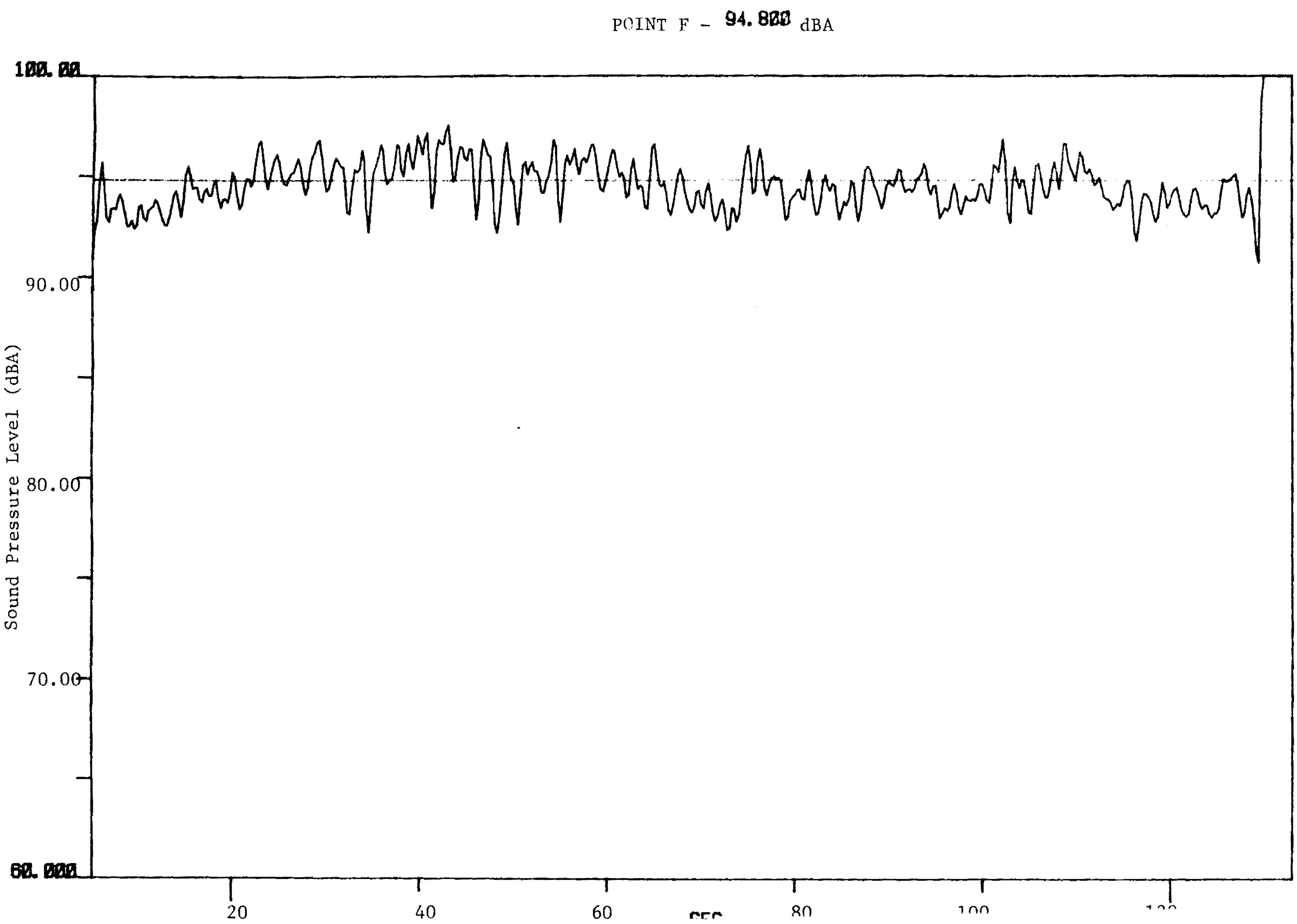
Figures 14 and 15 show the linear and A-weighted spectra for the noise buildup area between the lung guns and the gizzard inspection stations (point C). Prevalent in the linear spectra are peaks at both 250 Hz and 500 Hz. Both peaks are evident in the ice dump spectra; however, the 500 Hz peak is broader, possibly reflecting the broad 550 Hz peak of the lung guns.

Figures 16 and 17 show the linear and A-weighted spectra for the area directly in front of a circulating fan. A linear peak at approximately 350 Hz is smoothed by the A-weighting leaving primary peaks at 500 and 1000 Hz.

Figures 18 and 19 show the area near the conveyor portals from the picking area. These spectra typify an interval when the door to the picking area is closed. A primary peak at approximately 550 Hz is evident in both the A-weighted and linear spectra. Secondary A-weighted peaks appear at 700, 1000 and 1100 Hz. A noticeable secondary peak at 3200 Hz also is evident in both the linear and A-weighted spectra.

Figures 20 and 21 show the area near the ice dump for the chiller. A sharp peak near 500 Hz appears on the A-weighted spectra. A secondary A-weighted peak occurs at 250 Hz.

FIGURE 9


POINT F - 94.800 dBA
100.00
90.00
80.00
70.00
60.000
Sound Pressure Level (dBA)
20
40
60
80
100

POINT A - Linear (no weighting)

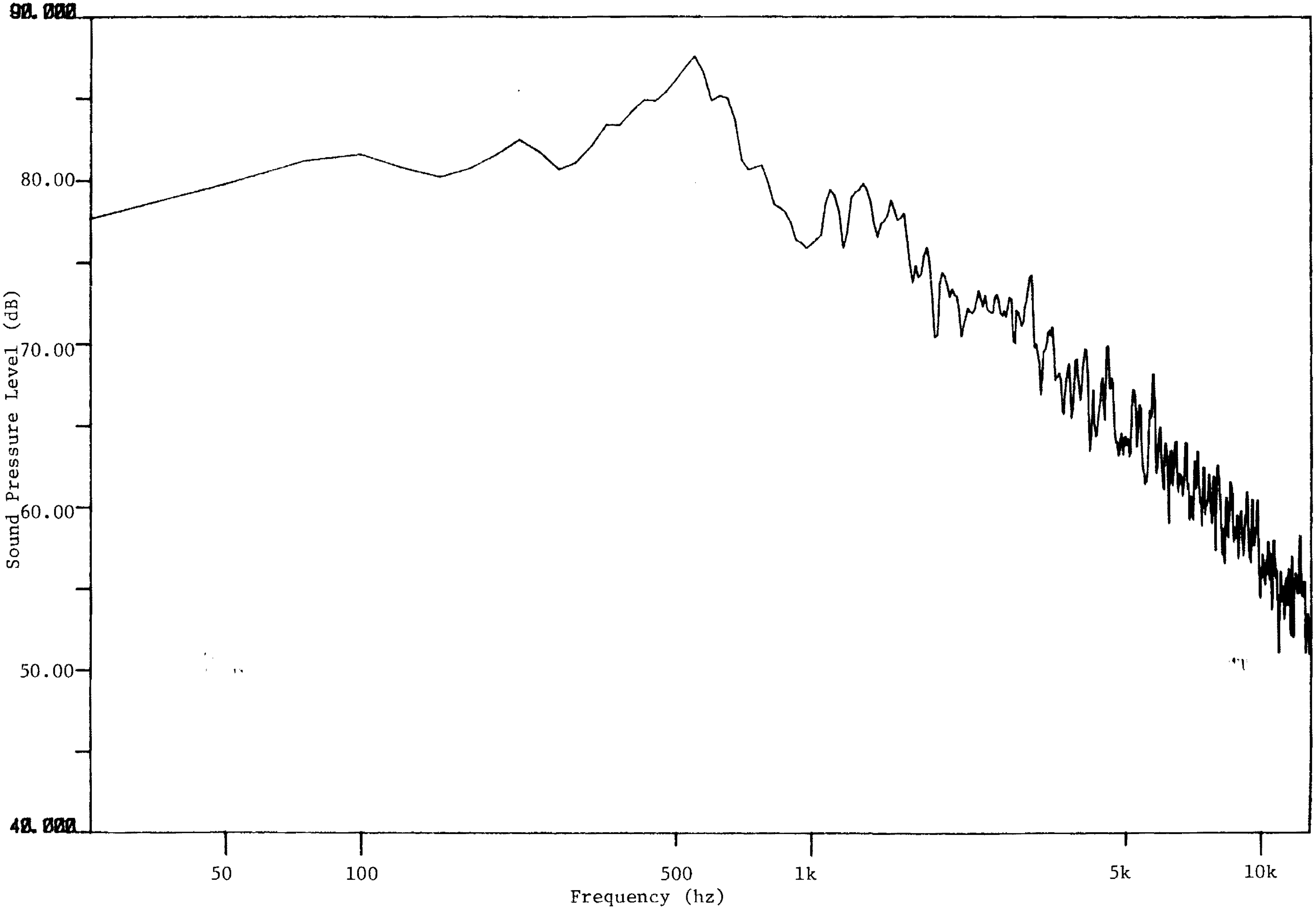


FIGURE 11

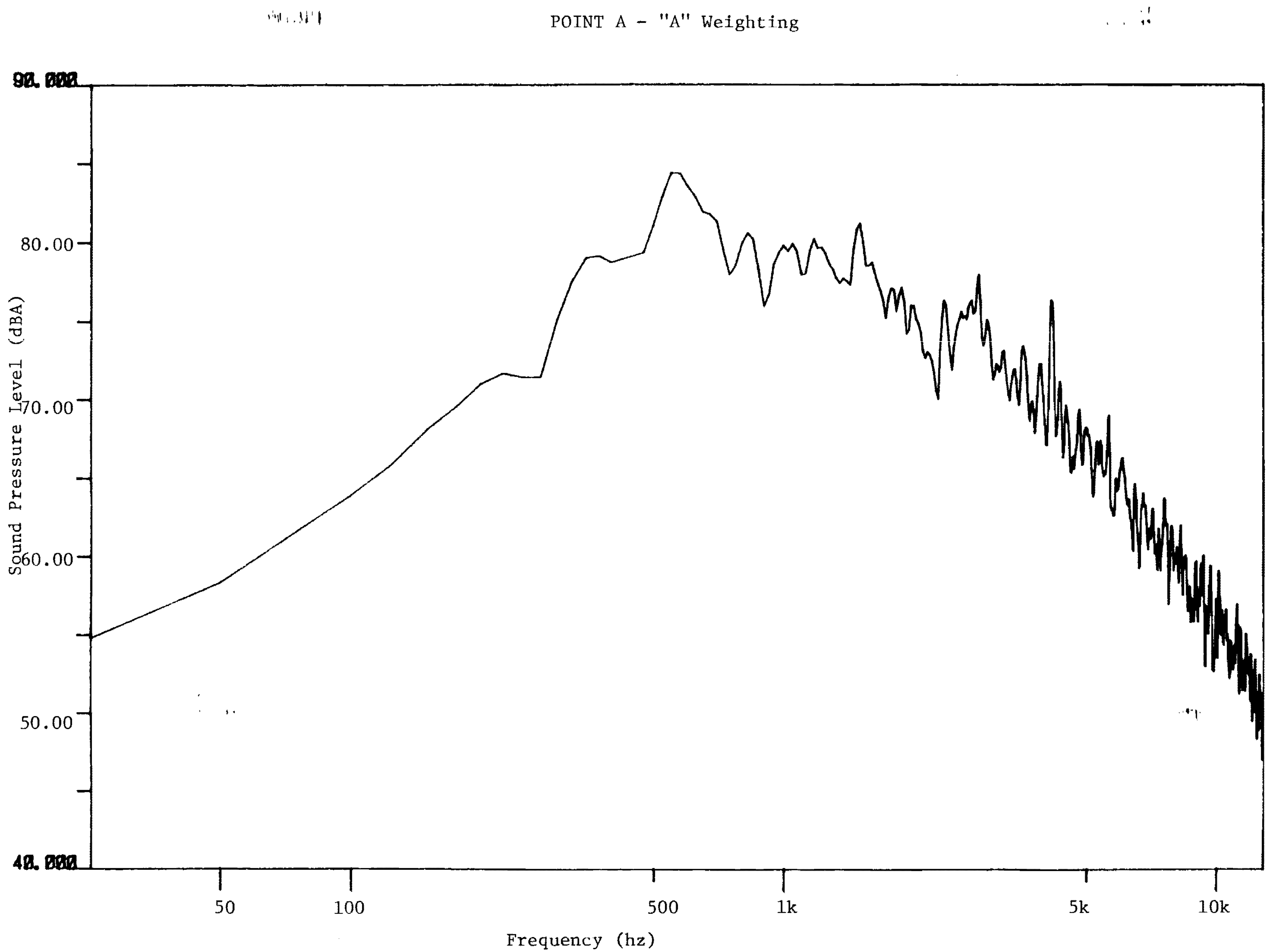
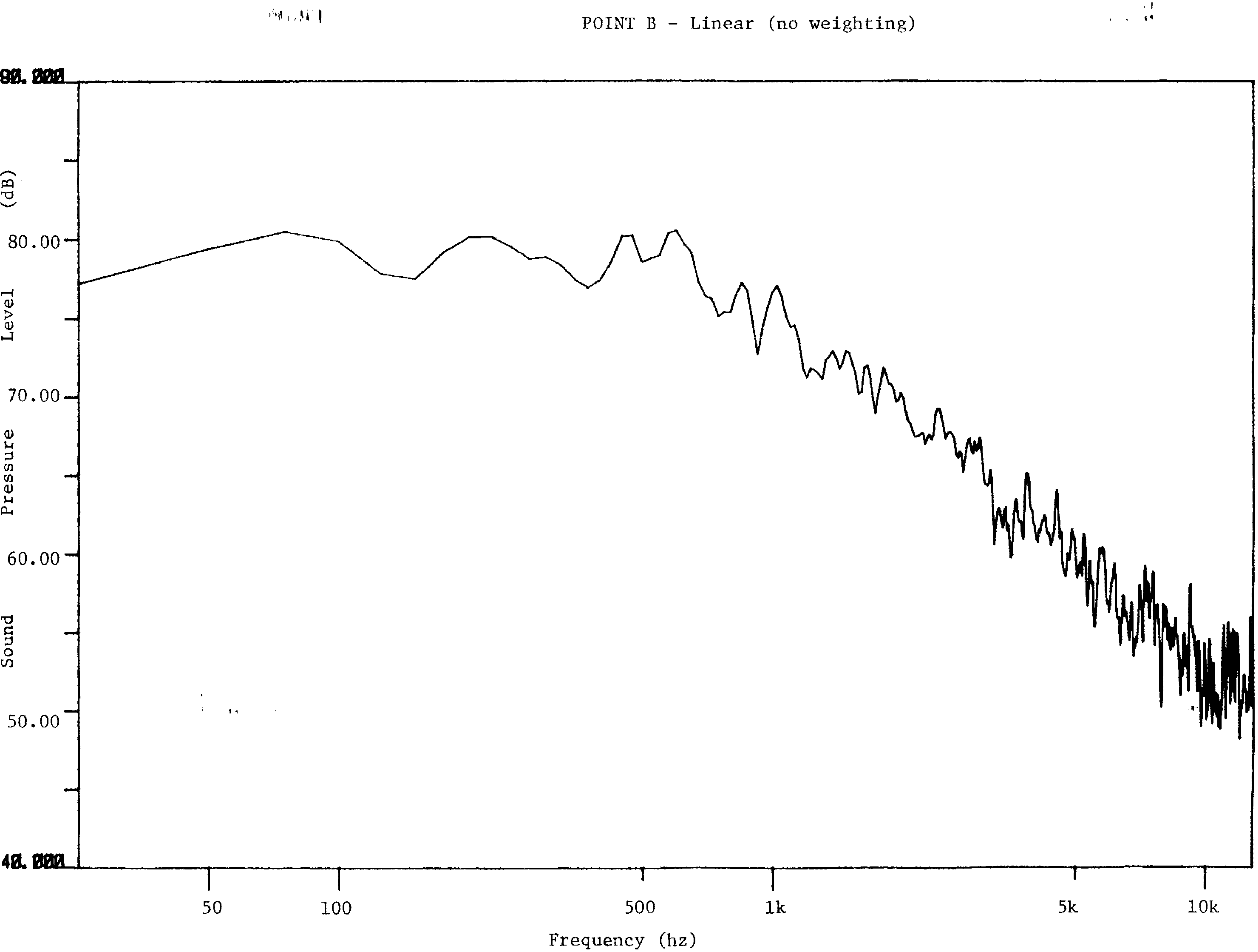

POINT A - "A" Weighting
Sound Pressure Level (dBA)
80.00
70.00
60.00
50.00
50
100
500
1k
5k
10k
Frequency (hz)

FIGURE 12



POINT B - Linear (no weighting)
Sound Pressure Level (dB)
80.00
70.00
60.00
50.00
50
100
500
1k
5k
10k
Frequency (hz)

FIGURE 13

POINT B - "A" weighting

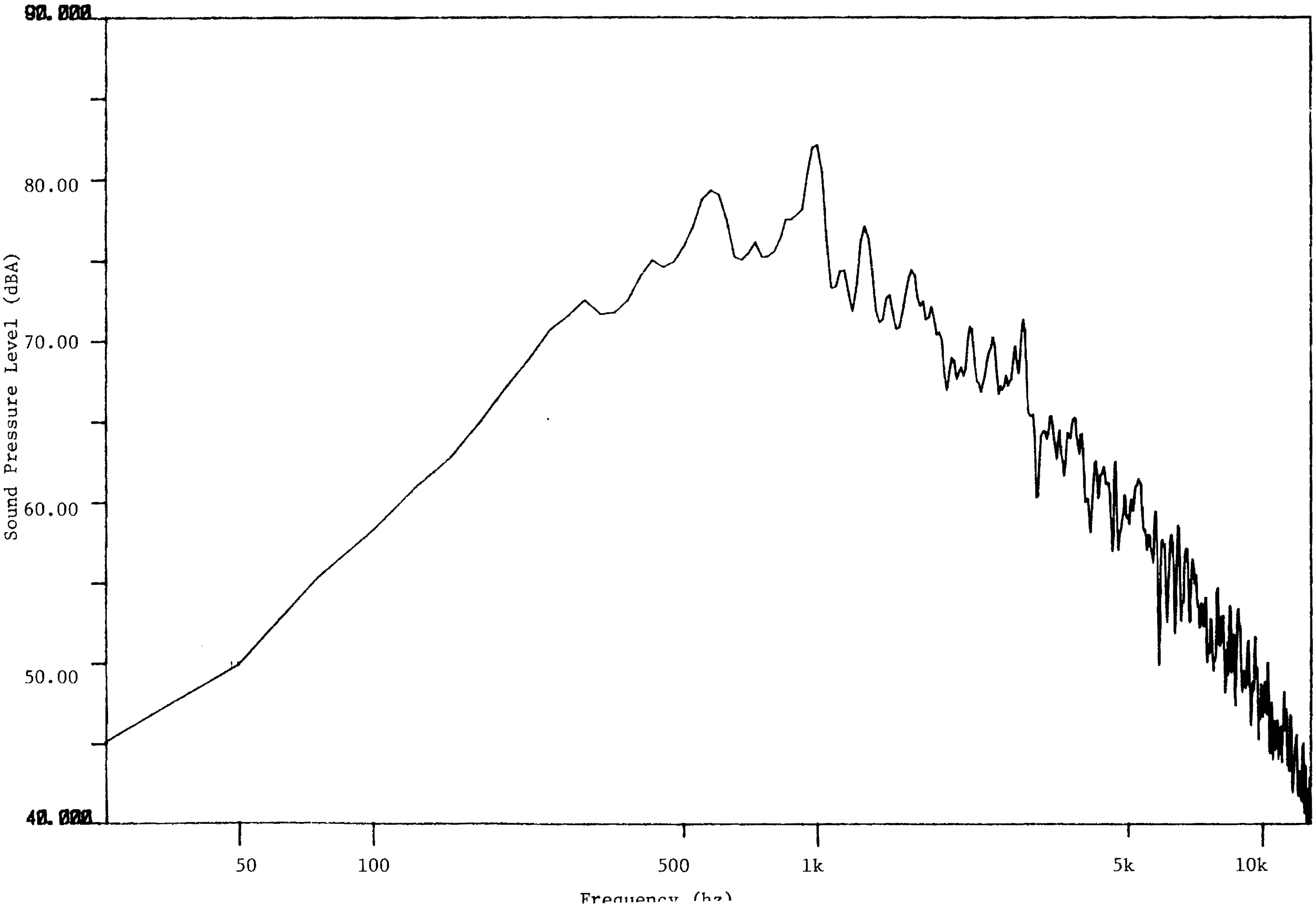
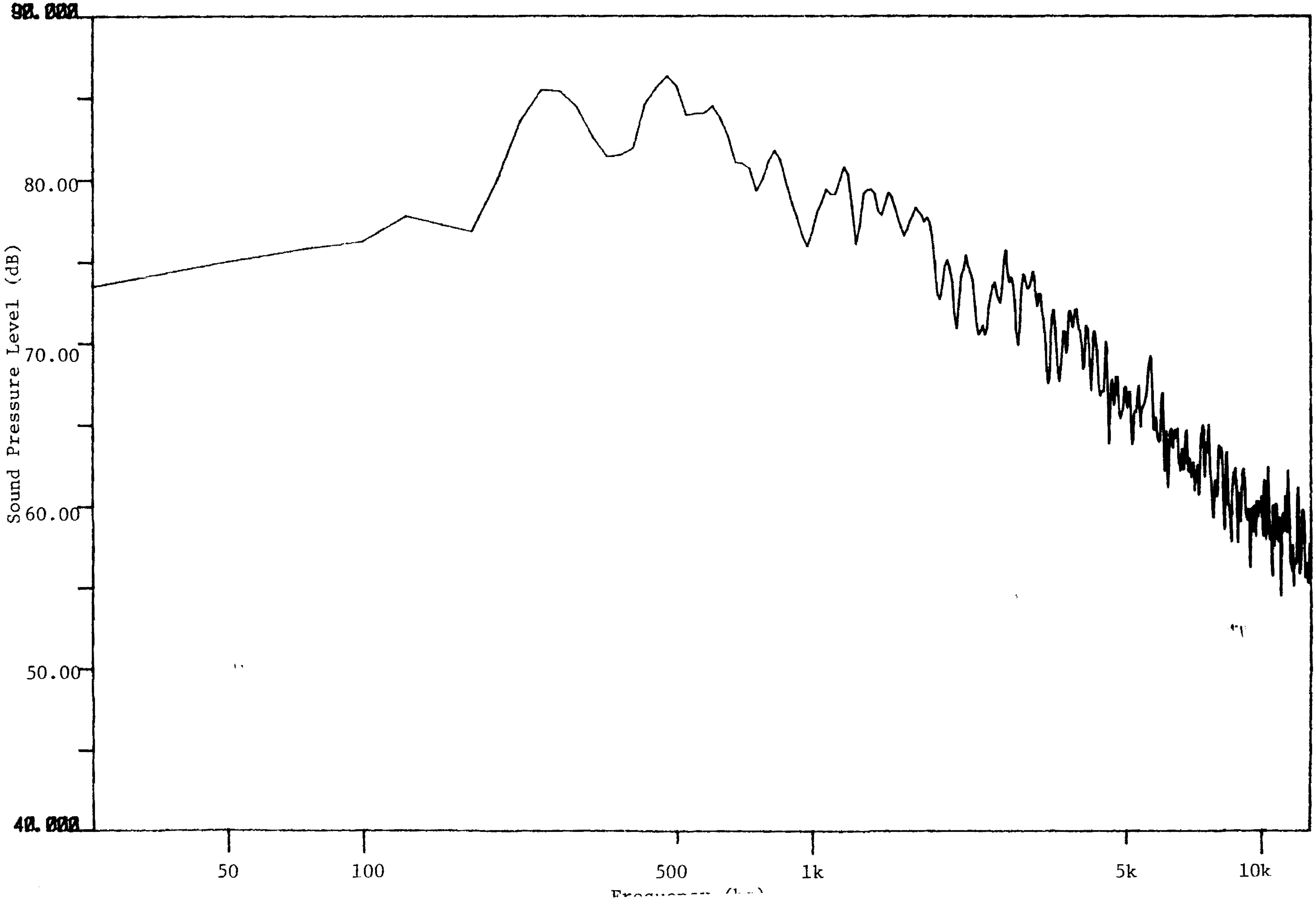


FIGURE 14

POINT C - Linear (no weighting)



F I G U R E   15

POINT C - "A" Weighting

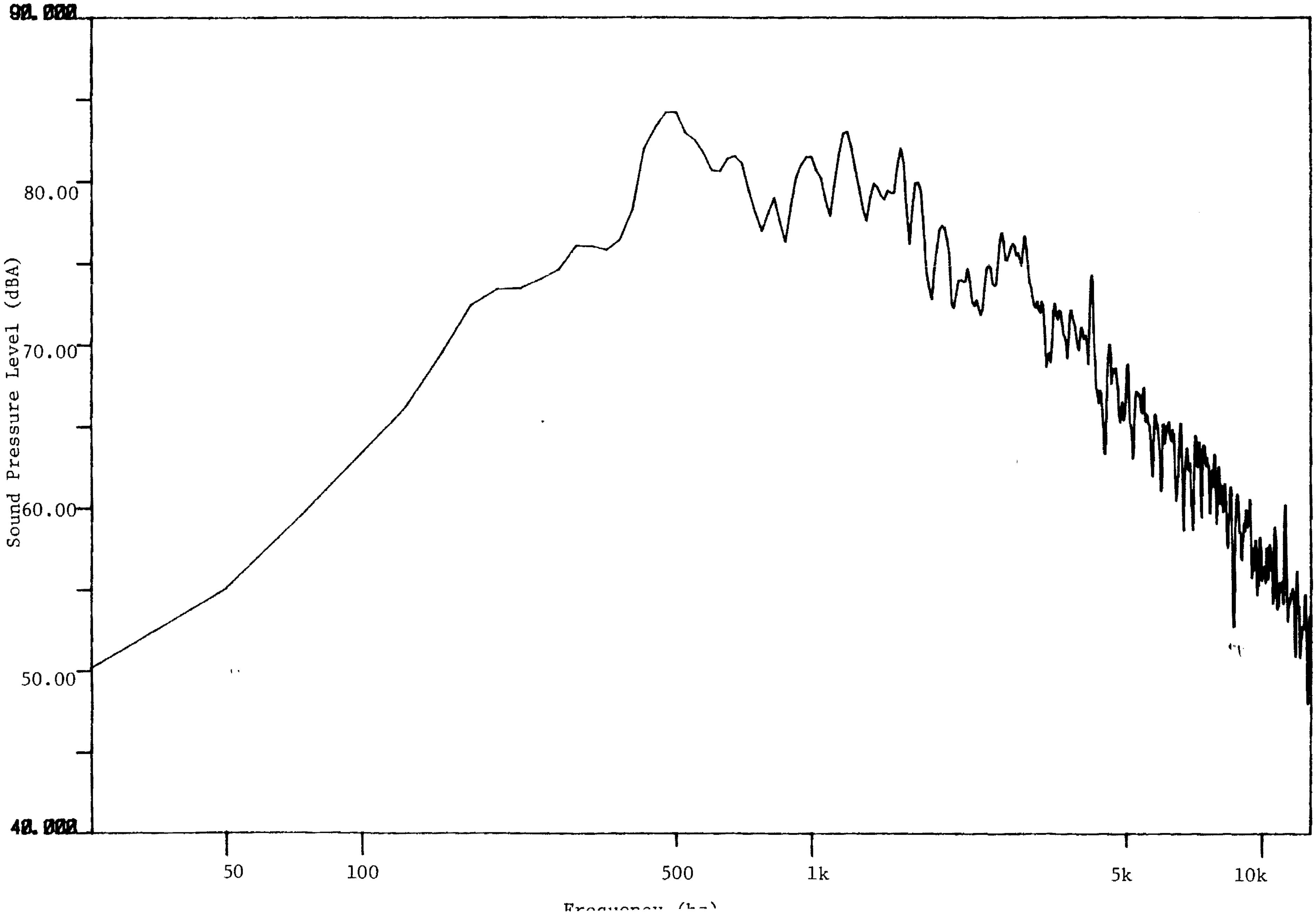


FIGURE 16

POINT D - Linear (no weighting)

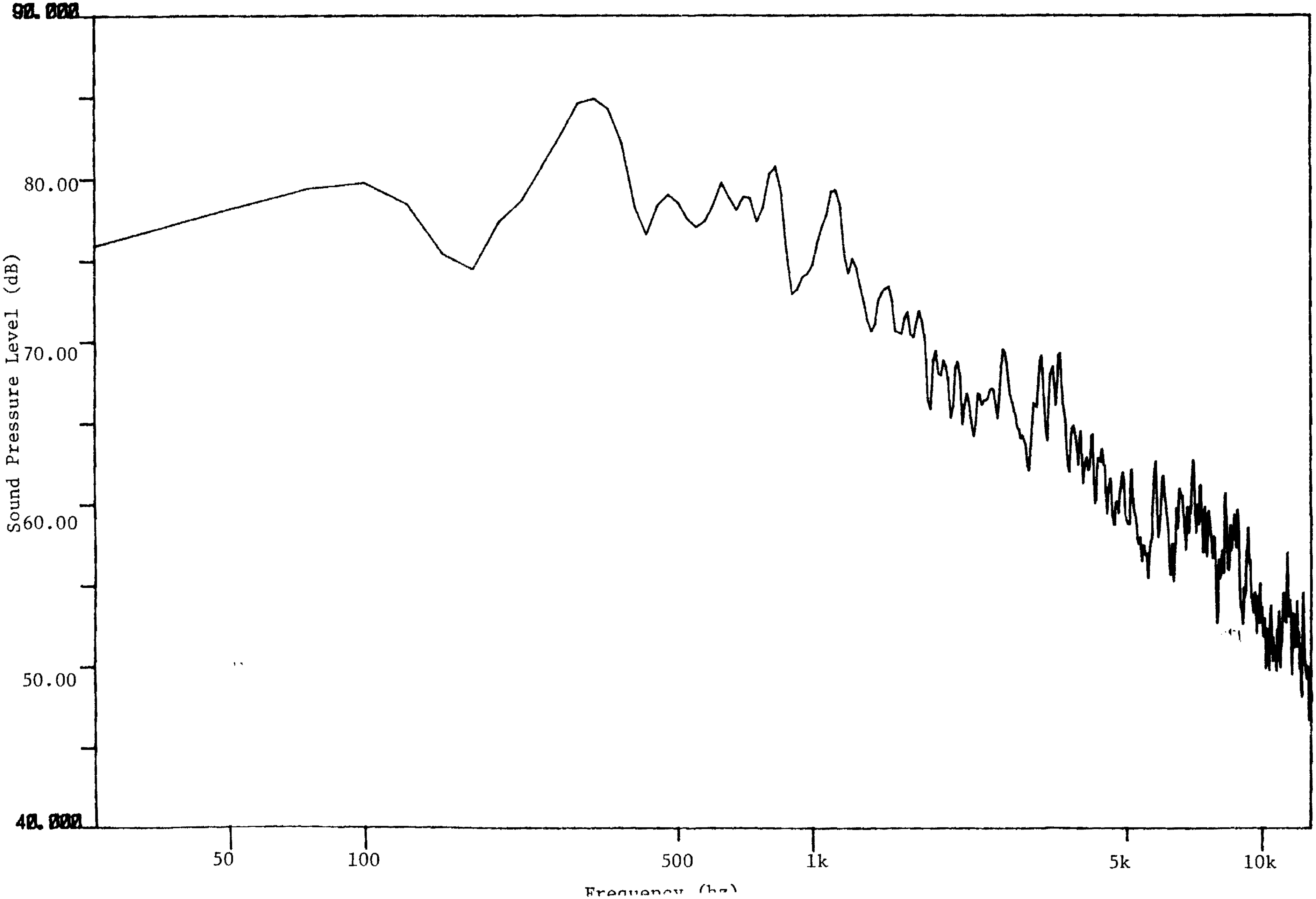


FIGURE 17

POINT D - "A" Weighting

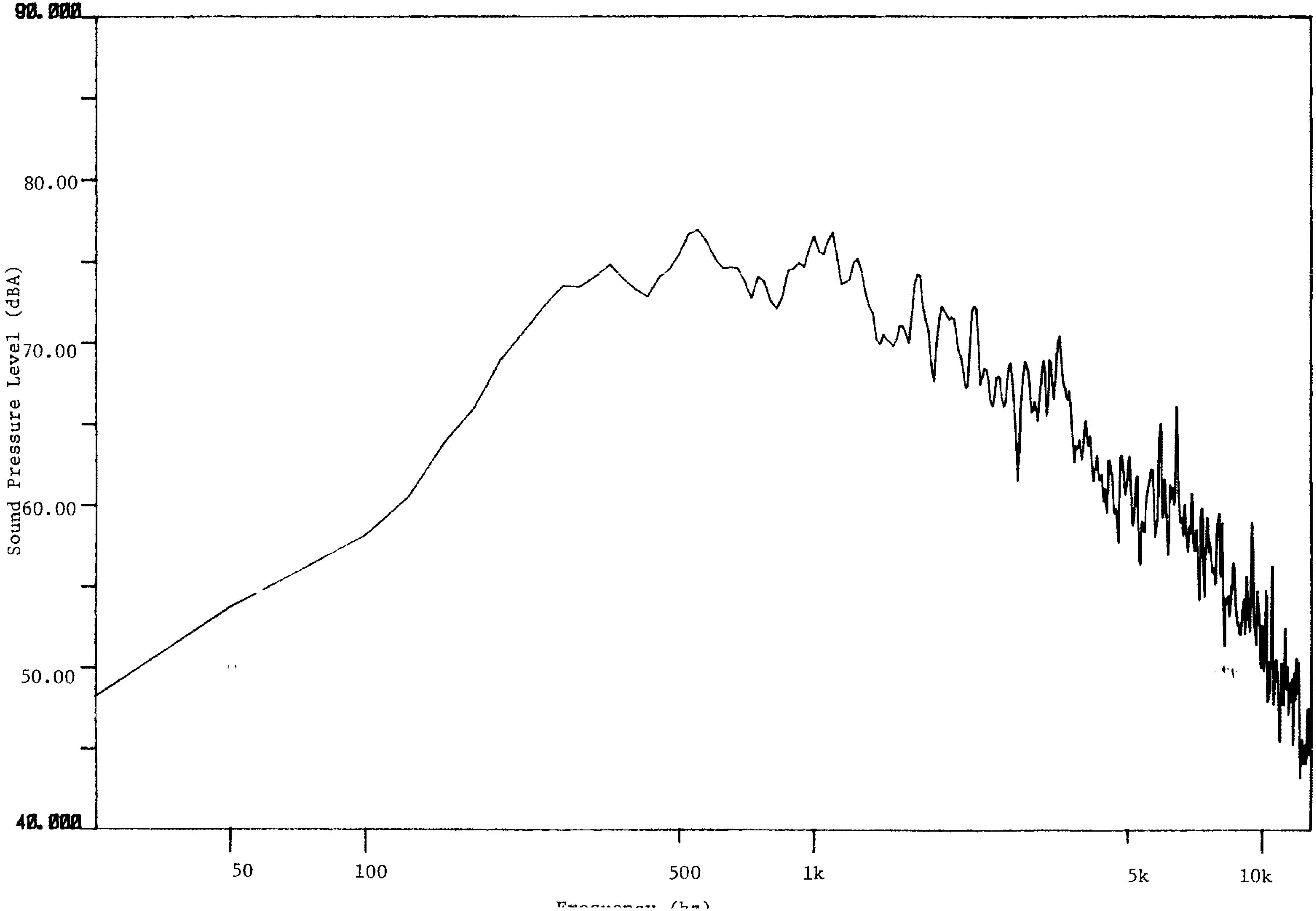
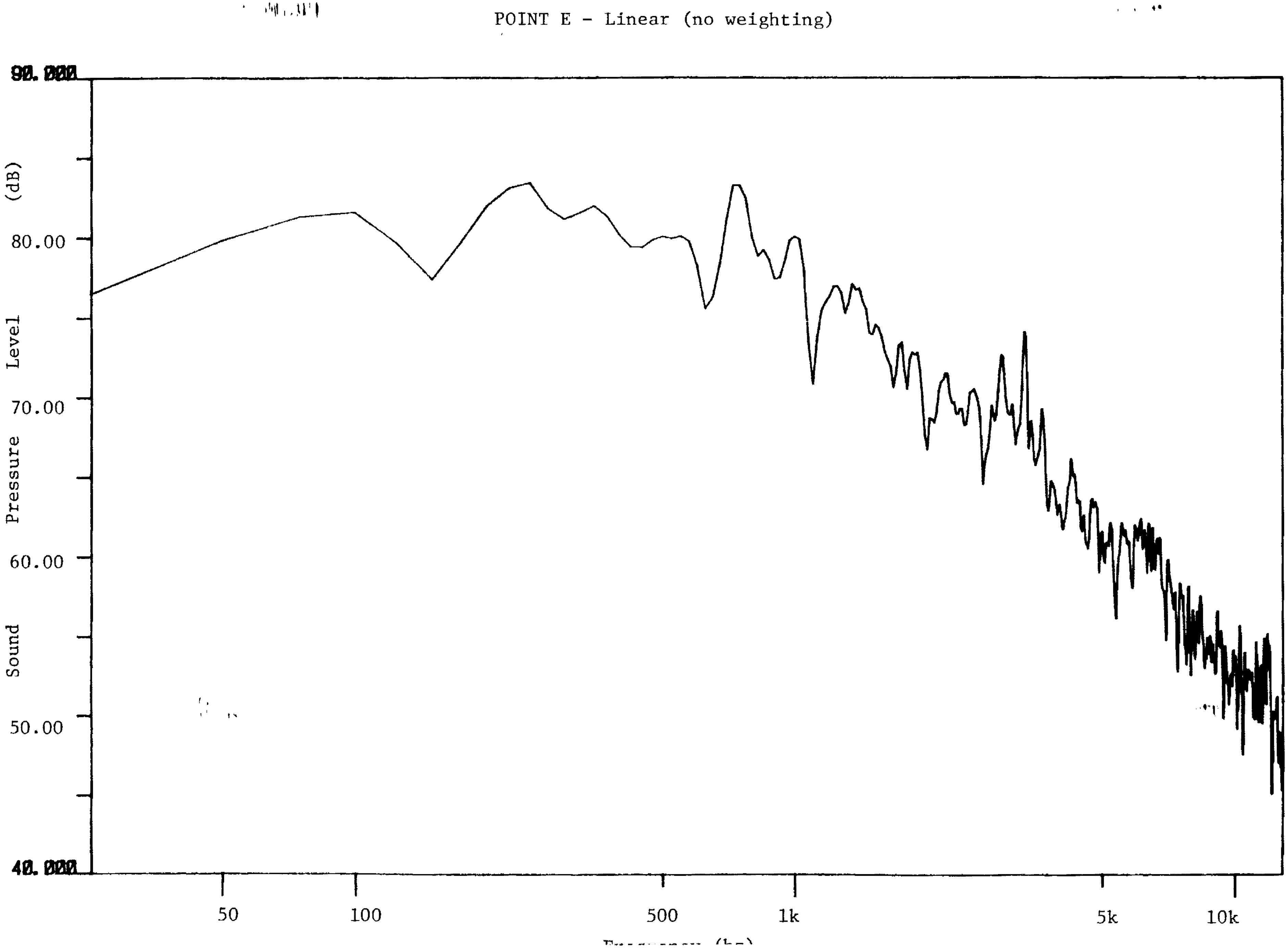


FIGURE 18



POINT E - Linear (no weighting)
90.000
80.00
70.00
60.00
50.00
40.000
Sound Pressure Level (dB)
50
100
500
1k
5k
10k

FIGURE 19

POINT E - "A" Weighting

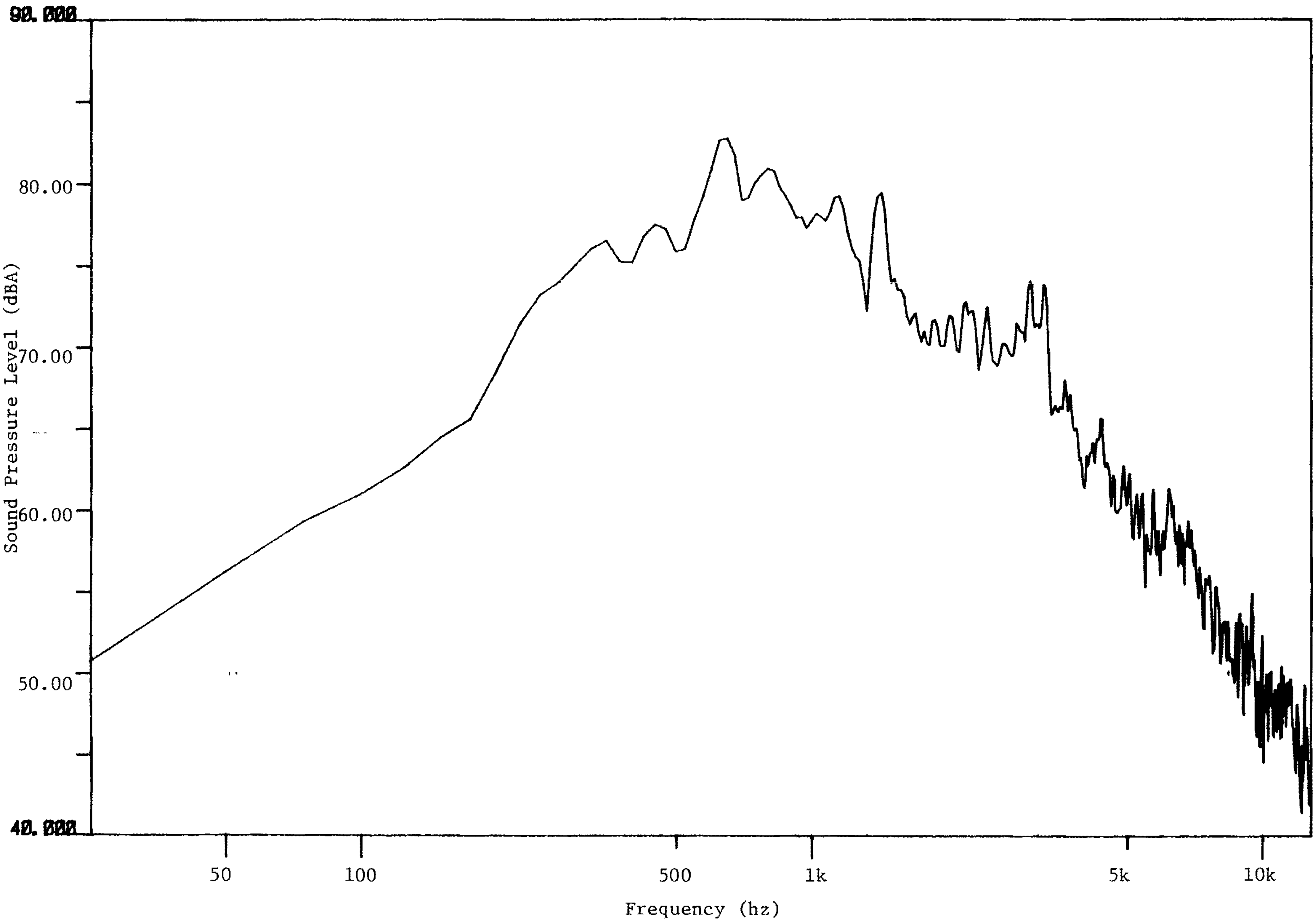


FIGURE 20

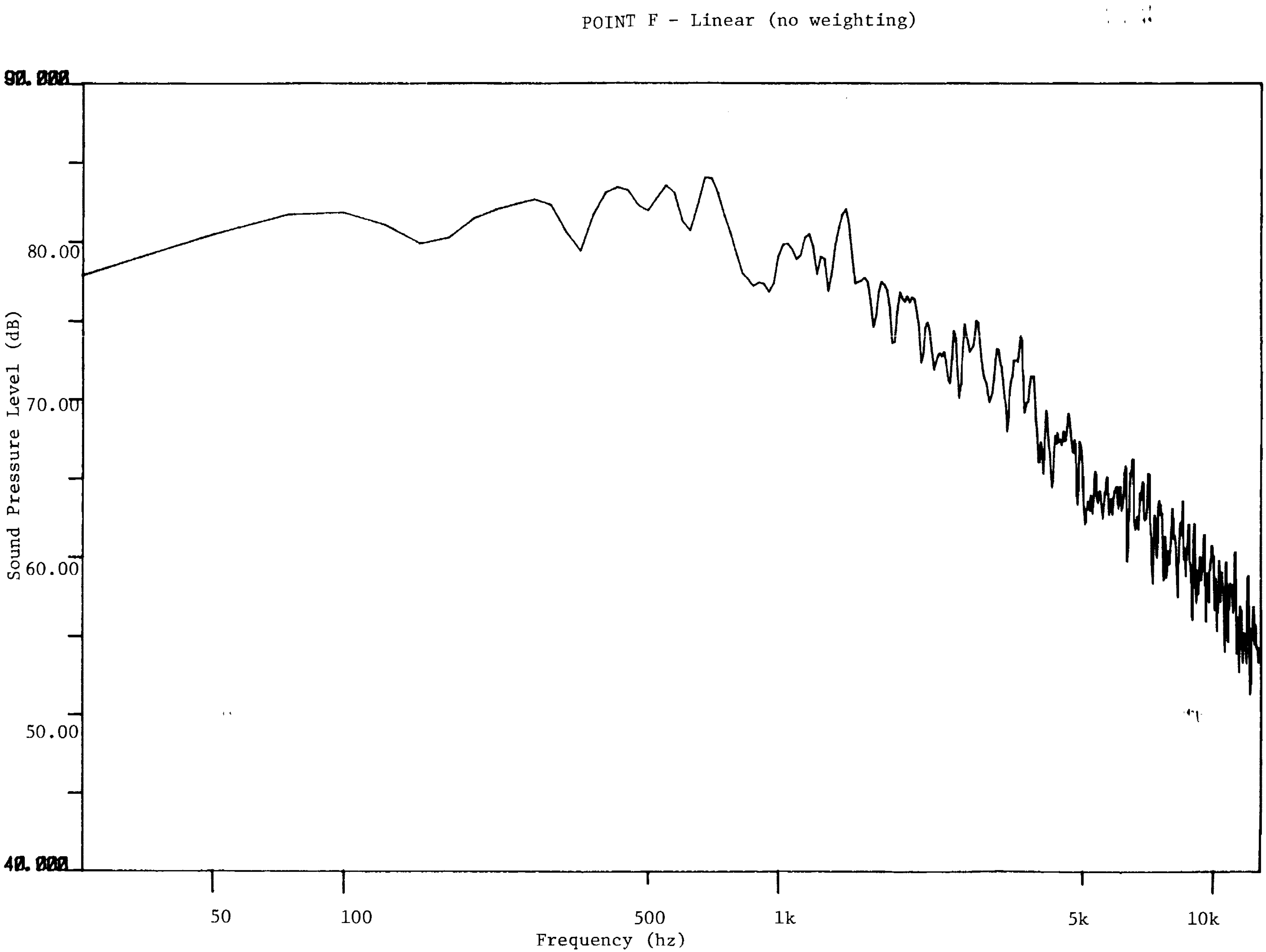
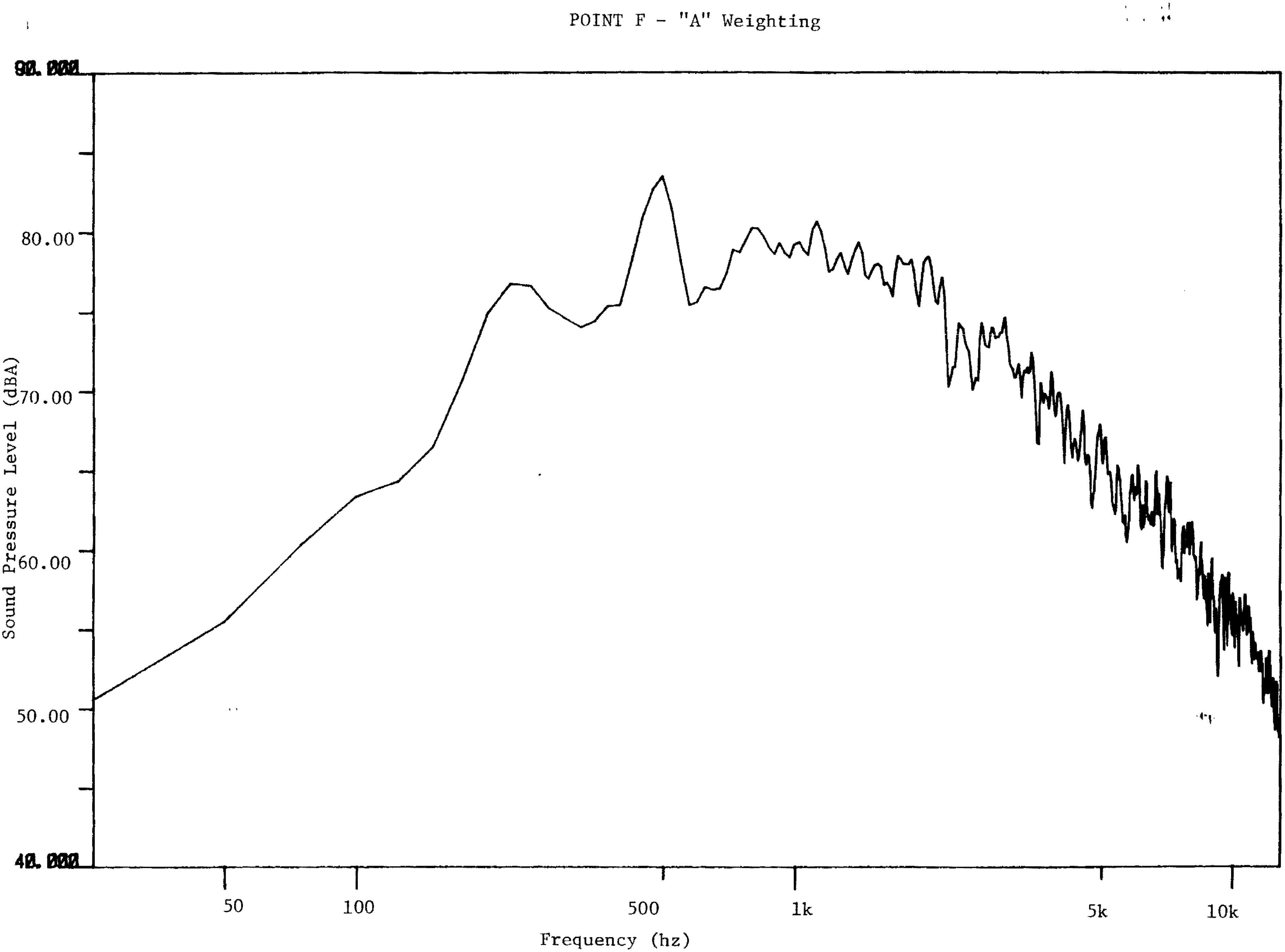

POINT F - Linear (no weighting)
90.000
80.00
70.00
60.00
50.00
40.000
Sound Pressure Level (dB)
50
100
500
1k
5k
10k
Frequency (hz)

FIGURE 21



POINT F - "A" Weighting
90.000
80.00
70.00
60.00
50.00
40.000
Sound Pressure Level (dBA)
50
100
500
1k
5k
10k
Frequency (hz)

IV. FUTURE ACTIVITIES

Project activities to be conducted in the remaining six months of the grant include:

- Detailed noise signature analysis of major noise sources
- Cross-correlation of noise data to determine relative contributions of sources to the overall noise level
- Reverberation time measurements
- Preliminary identification of potentially applicable noise reduction technology

In addition, cursory data will be obtained in a second poultry processing plant for the purpose of quantitizing differences which may exist between the plants.

APPENDIX A

type 2203

Precision Sound Level Meter

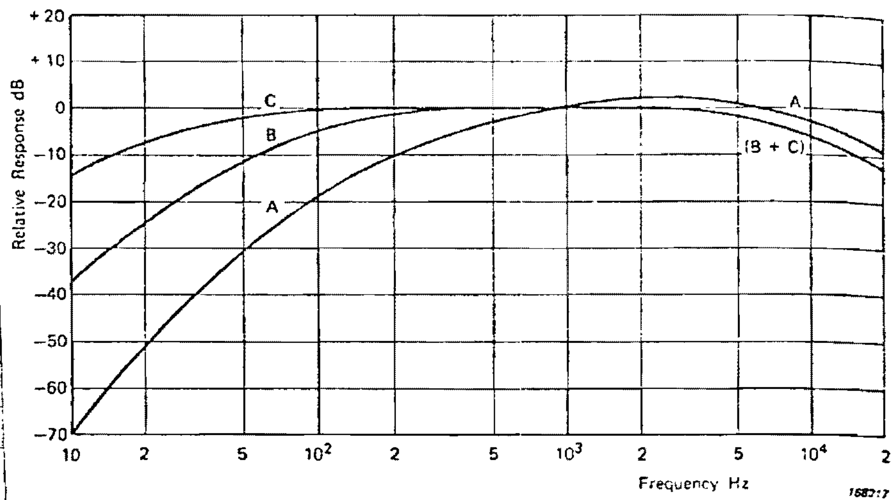
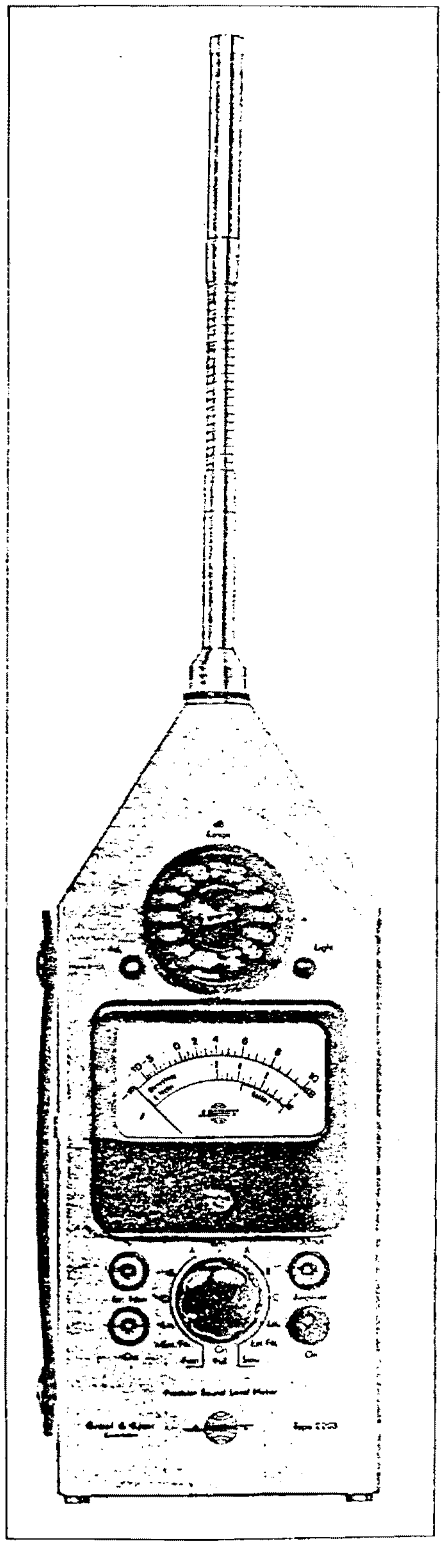


Fig. 1. Frequency response curves of the A, B and C weighting networks



FEATURES:

- Complies with all existing standards for precision sound level meters
- Equipped with individually calibrated, high sensitivity, precision condenser microphone
- Conical shaped front-end for minimum disturbance of sound field
- RMS detector with crest factor capability up to 5
- "A", "B", "C" and "Lin" frequency weighting
- Performs as Octave or Third-Octave analyzer with system-matching filter sets
- AC output for recorders etc.
- Performs as Vibration Meter or Analyzer combined with appropriate accessories
- Wide selection of accessories

USES:

- Acoustics, measurement of sound insulation, sound distribution etc.
- Noise and vibration measurements in industry for quality inspection and development
- Noise and vibration measurements for health protection
- Audiometer calibration

The Precision Sound Level Meter Type 2203 has been designed as a robust compact unit able to perform sound and vibration measurements of almost any kind with the highest degree of accuracy. The instrument fulfils the requirements of IEC 179, DIN 45633 part 1 as well as ANSI S1.4-1971 Type 1 for precision sound level meters and includes the A, B and C weighting networks and the Fast and Slow meter responses. In- and output sockets, for connection of external filters for frequency analysis of the measured signal, as

appendix A

B and K 42...

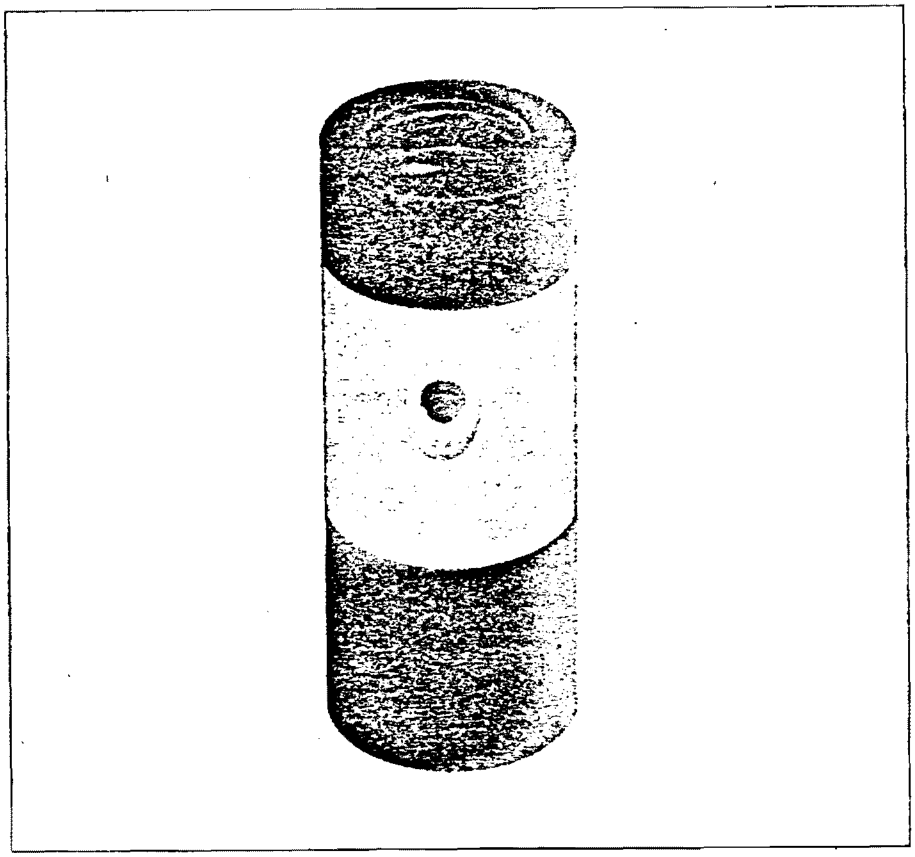
Sound Level Calibrator

FEATURES:

- Pocket size, portable unit
- Calibration accuracy ± 0,25 dB
- Frequency 1000 Hz, gives independence of weighting networks
- Calibration sound pressure level 94 dB = 1 Pascal= 1 N/m² = 10 μbar
- Extremely small influence of static pressure
- Sound pressure independent of microphone equivalent volume
- Fits B & K 1" and 1/2" microphones

USES:

- Calibration of sound measuring equipment



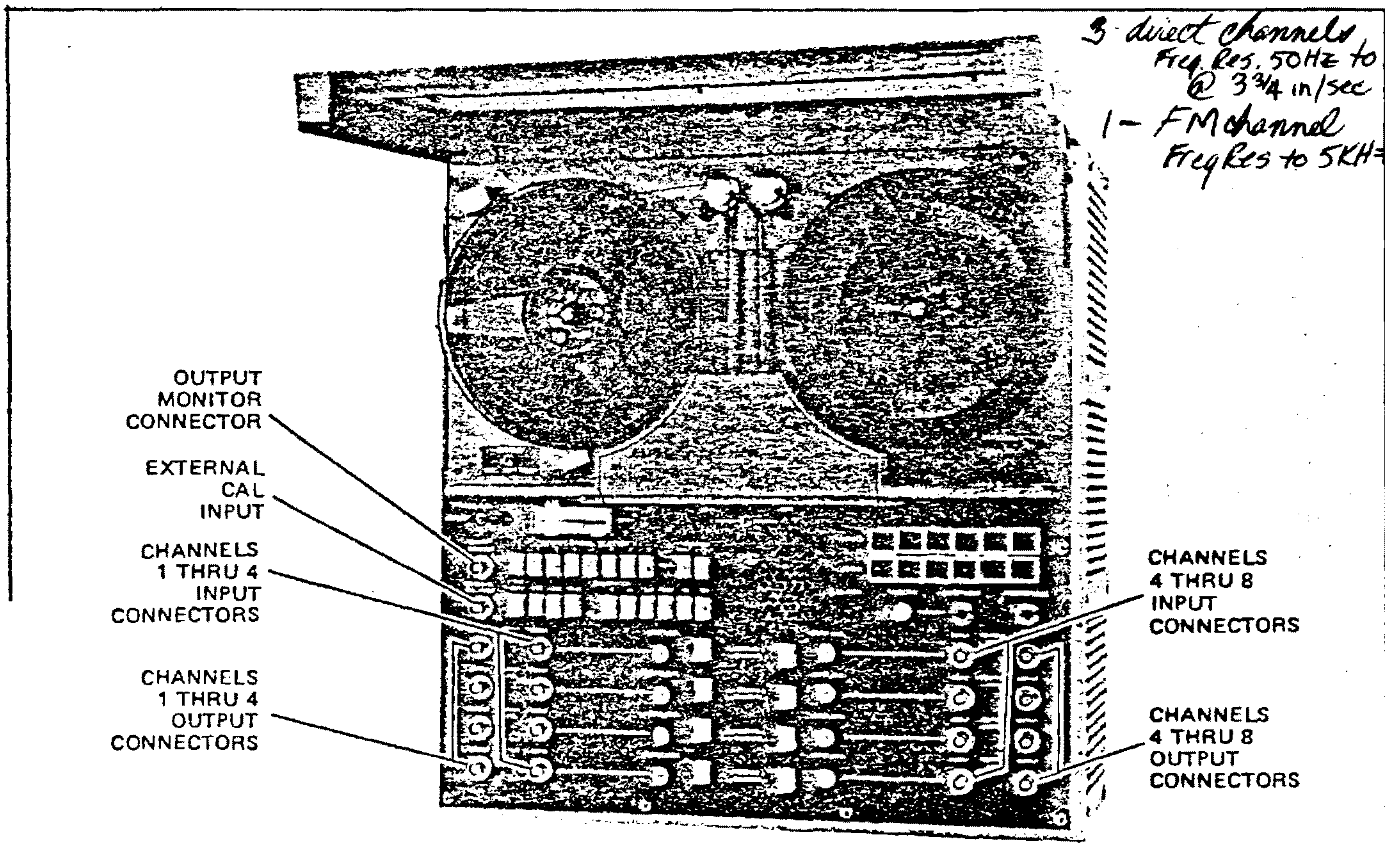
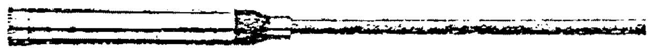


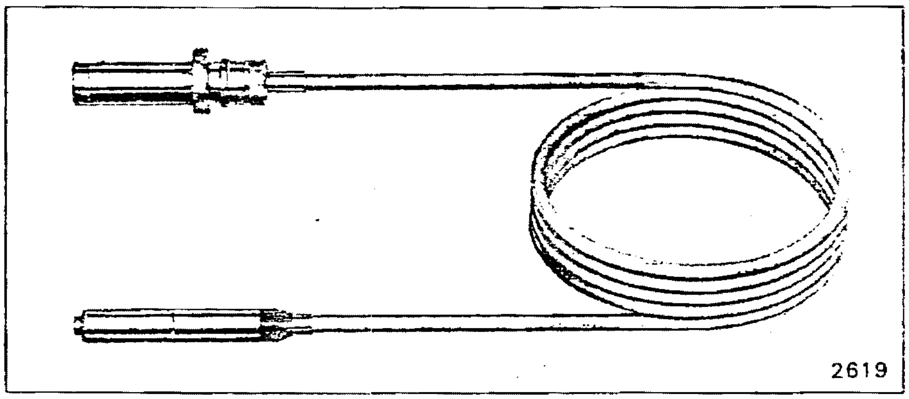
Figure 2-2. Front Panel Connectors

HEWLETT-PACKARD 3964A

appendix A

1/2" Cartridge + 2619





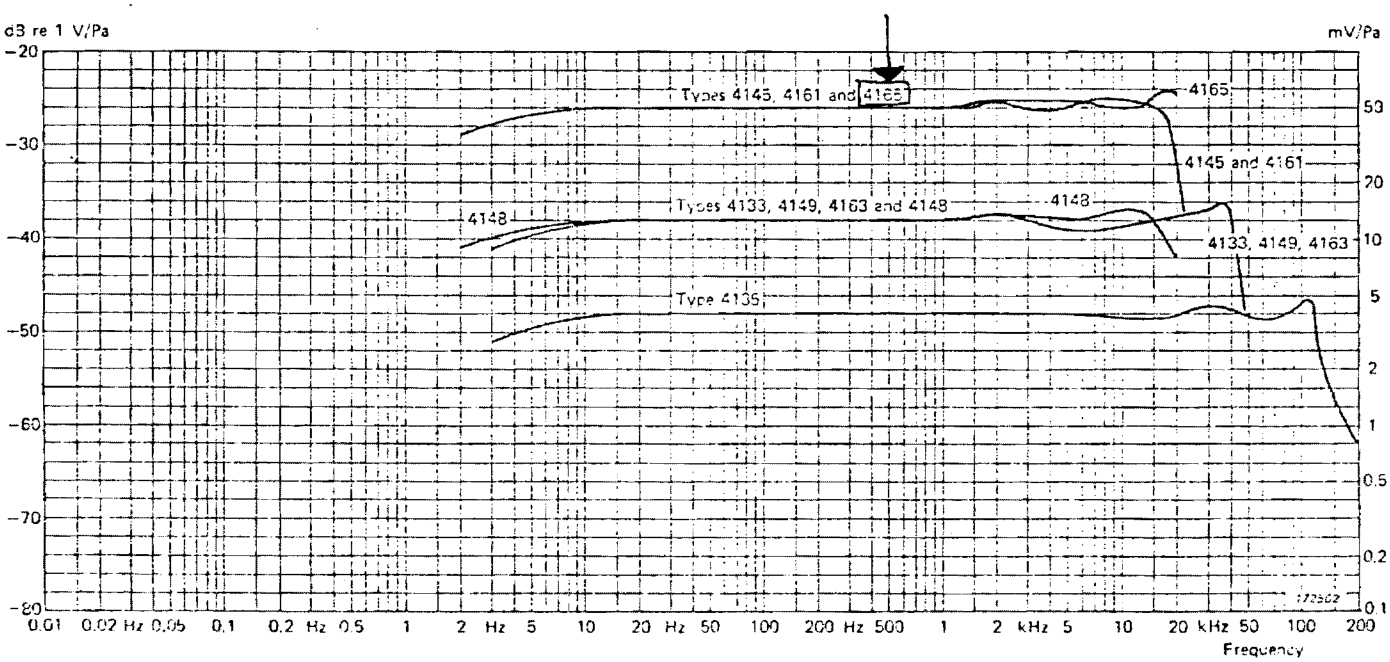


Fig.16. Typical 0° incidence frequency responses of the different free-field microphones recorded by means of the electrostatic actuator method and corrected according to the curves shown in Fig.20

appendix A

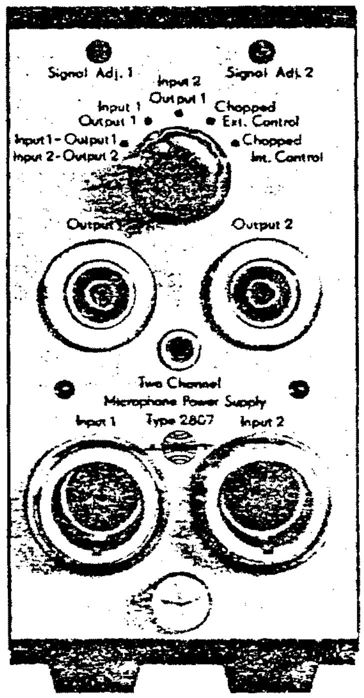
Two-Channel Microphone Power Supply

FEATURES:

- Two signal channels with manual and automatic switching
- Frequency range 2 Hz to 200 kHz
- Croos-talk less than 60 dB
- Channel attenuation adjustable between 0,15 dB and approximately 40 dB
- Allows use of long cables between transducer and measuring instrumentation
- Outputs short circuit protected
- Fits B & K module system for rack mounting

USES:

- Power Supply for two microphone assemblies or preamplifiers
- Sound insulation measurements with B & K condenser microphones
- Vibration measurements with B & K accelerometers
- Remote switching between measuring points
- Simultaneous tape recording of sound and vibration data



The Two-Channel Microphone Power Supply Type 2807 is designed for use in connection with the B & K Microphone Preamplifiers Types 2618, 2619 and 2627. The 2807 supplies the necessary power for the preamplifiers and the polarisation voltage for the condenser microphones. In connection with preamplifiers, the 2807 acts as an impedance transformer with high input impedance and low output impedance allowing the use of long cables between the output of the 2807 and the equipment following it.

The 2807 further enables the use of the two channels simultaneously giving the possibility of measuring, for instance, noise and vibration at the same time, and the choice of manual or automatic switching between the two channels to one set of recording instruments connected to the output. This feature gives the possibility of building up measuring systems having the advantage of simplicity and flexibility.

appendix A

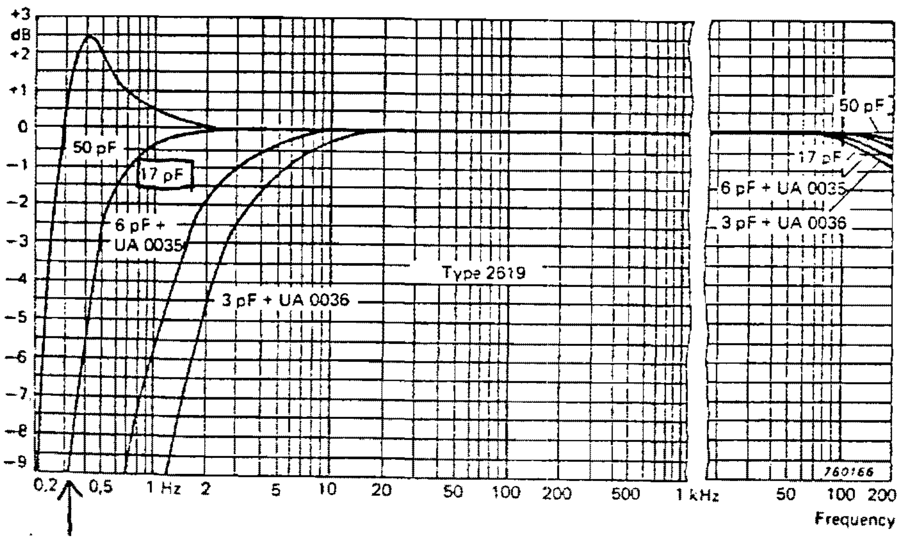


Fig.10. Preamplifier Type 2619. Frequency response curves with different transducer capacitances connected at the input

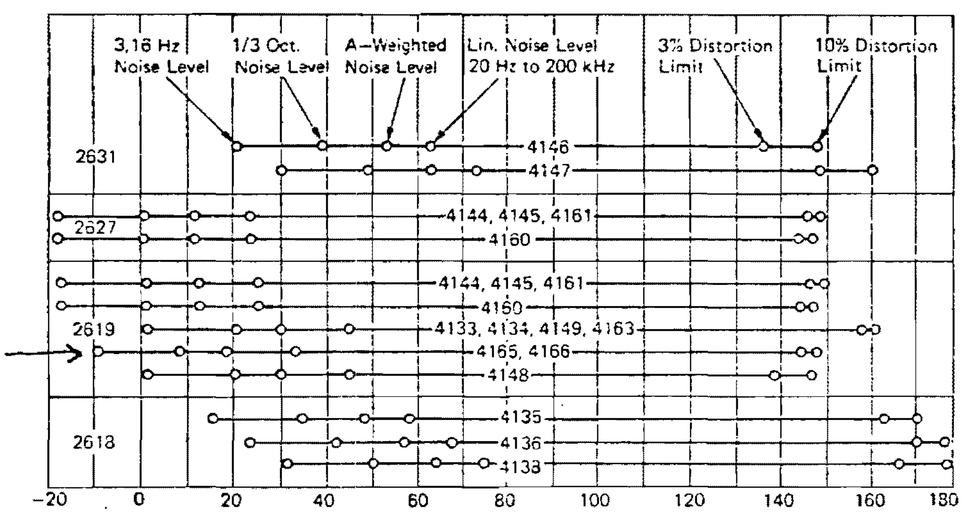
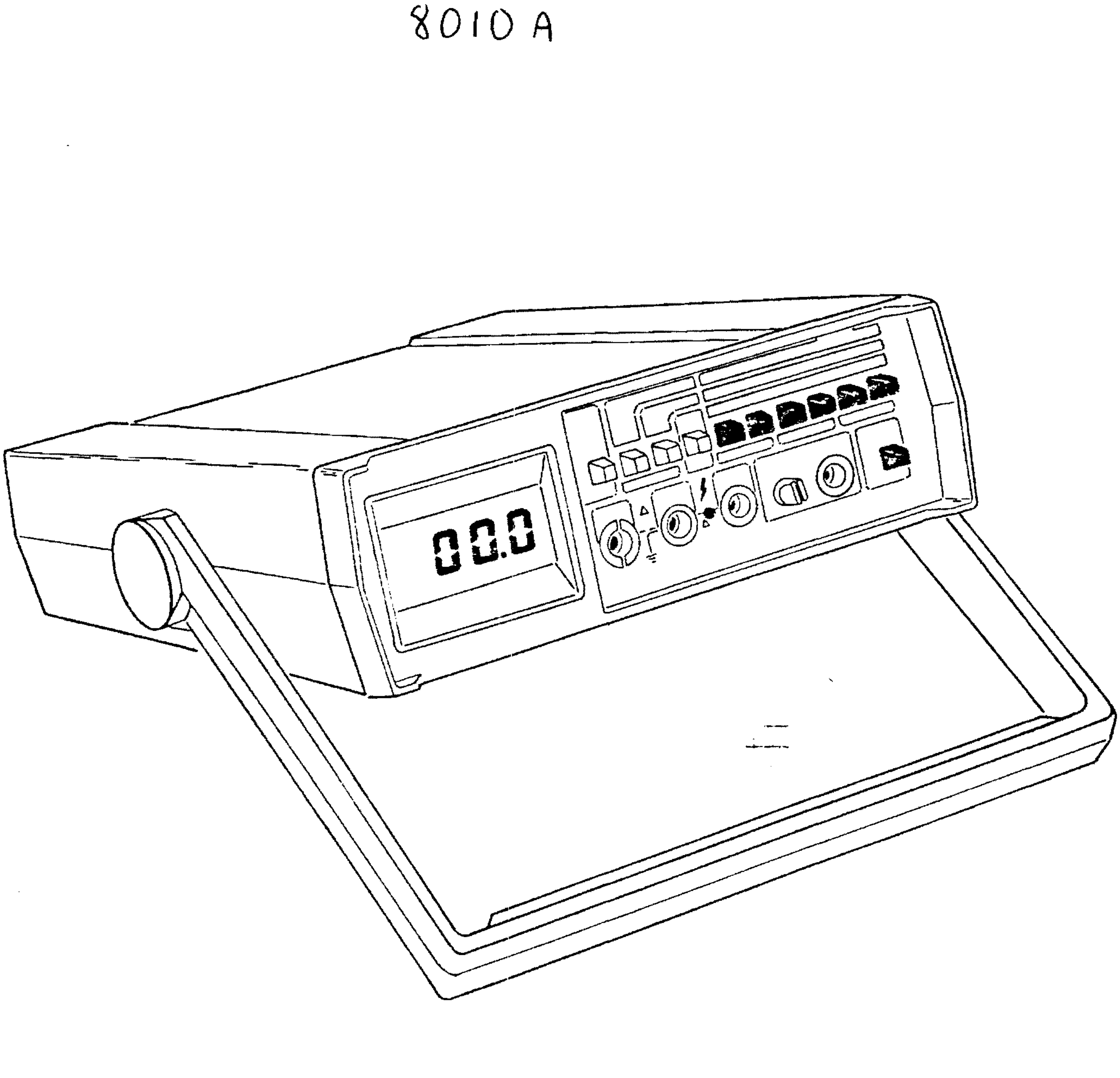


Fig.13. Dynamic ranges of B & K Microphone Preamplifiers (except 2642) and the Carrier System 2631 with different microphones. The upper limit is indicated for two degrees of distortion while the lower limit is given for various bandwidths of the measuring equipment. The limits for 3,16 Hz and 1/3 octave bandwidth is valid at 1000 Hz only

APPENDIX B

Appendix B



8010A/8012A Digital Multimeter

Table 1-1. 8010A/8012A Specifications

ELECTRICAL: The electrical specifications given assume an operating temperature of 18°C to 28°C, humidity up to 90%, and a 1-year calibration cycle.

FUNCTIONS: DC Volts, AC Volts, DC Current, Resistance and Conductance.

DC VOLTS:

RANGE	RESOLUTION	ACCURACY for 1-Year
±200 mV	100 μV	±(0.1% of reading + 1 digit)
±2V	1 mV	±(0.1% of reading + 1 digit)
±20V	10 mV	±(0.1% of reading + 1 digit)
±200V	100 mV	±(0.1% of reading + 1 digit)
±1000V	1V	±(0.1% of reading + 1 digit)

INPUT IMPEDANCE: 10 MΩ, all ranges.

NORMAL MODE REJECTION RATIO: > 60 dB at 60 Hz (at 50 Hz on 50 Hz Option).

Appendix B 3

Table 1-1. [illegible] Specifications [illegible]

COMMON MODE REJECTION RATIO: (1 kΩ unbalance) > 120 dB at dc, 50 Hz and 60 Hz.

OVERVOLTAGE PROTECTION: 1000V dc or peak ac on all ranges.

RESPONSE TIME: 1 second.

AC VOLTS (TRUE RMS RESPONDING):

RANGE	RESOLUTION	ACCURACY for 1-Year			
		45 Hz		10 kHz	20 kHz
		to 1 kHz	to 10 kHz	to 20 kHz	to 50 kHz
200 mV	100 μV	±(0.5% of reading + 2 digits)		±(1.0% of reading +2 digits)	±(5% of reading +3 digits)
2V	1 mV				
20V	10 mV				
200V	0.1V				
750V	1V	±(0.5% of reading +2 digits)			

VOLT-Hz PRODUCT: 10^7 max (200V max @ 50 kHz).

EXTENDED FREQUENCY RESPONSE: Typically ±3 dB at 200 kHz.

COMMON MODE NOISE REJECTION RATIO (1 kΩ unbalance): > 60 dB at 50 Hz and 60 Hz.

CREST FACTOR RANGE: 1.0 to 3.0.

INPUT IMPEDANCE: 10 MΩ in parallel with < 100 pF.

OVERLOAD PROTECTION: 750V rms or 1000V peak continuous not to exceed the volt-hertz product of 10^7 (except 10 seconds maximum on 200 mV, 2V ranges).

RESPONSE TIME: 2 seconds maximum within a range.

DC CURRENT:

RANGE	RESOLUTION	ACCURACY for 1-Year	BURDEN VOLTAGE
20 μA	0.1 μA	±(0.3% of reading +1 digit)	0.3V max.
2 mA	1 μA		
20 mA	10 μA		
200 mA	100 μA		
2000 mA	1 mA		0.9V max.

OVERLOAD PROTECTION: 2A/250V fuse in series with 3A/600V fuse (for high energy sources).

AC CURRENT (TRUE RMS RESPONDING):

RANGE	RESOLUTION	ACCURACY: from 5% of range to full-scale, 1-Year			BURDEN VOLTAGE
		45 Hz to 2 kHz	45 Hz to 10 kHz	10 kHz to 20 kHz	
200 μA	0.1 μA	±(1% of reading +2 digits)	±(1% of reading +2 digits)	±(2% of reading +2 digits)	0.3V rms max.
2 mA	1 μA				
20 mA	10 μA				
200 mA	100 μA				
2000 mA	1 mA	±(1% of reading +2 digits)			0.9V rms max.

OVERLOAD PROTECTION: 2A/250V fuse in series with 3A/600V fuse (for high energy sources).

CREST FACTOR RANGE: 1.0 to 3.0.

HIGH CURRENT-8010A ONLY:

RANGE	RESOLUTION	ACCURACY: for 1-Year	BURDEN VOLTAGE
10A dc	10 mA	±(0.5% of reading + 1 digit)	0.5V max.
10A Trms ac	10 mA	45 Hz to 2 kHz ±(1% of reading + 2 digits)	0.5V rms max.

OVERLOAD: 12A max unfused.

RESISTANCE:

RANGE	RESOLUTION	ACCURACY: for 1-Year	FULL-SCALE VOLTAGE	MAXIMUM TEST CURRENT
200 Ω	0.1Ω	±(0.2% of reading +1 digit)	0.25V	1.3 mA
2 kΩ ▶|	1Ω		1.0V	1.3 mA
20 kΩ	10Ω		<0.25V	10 μA
200 kΩ ▶|	100Ω		1.0V	35 μA
2000 kΩ	1 kΩ	±(0.5% of reading +1 digit)	<0.25V	0.10 μA
20 MΩ ▶|	10 kΩ		1.5V	0.35 μA

OVERLOAD PROTECTION: 300V dc/ac rms on all ranges.

OPEN CIRCUIT VOLTAGE: Less than 3.5V on all ranges.

RESPONSE TIME: 1 second, all ranges except 2000 kΩ and 20 MΩ ranges - 4 seconds these two ranges.

DIODE TEST: ▶|

These three ranges have enough voltage to turn on silicon junctions to check for proper forward-to-back resistance. The 2 kΩ range is preferred and is marked with a larger diode symbol on the front panel of the instrument. The three non-diode test ranges will not turn on silicon junctions so in-circuit resistance measurements can be made with these three ranges.

appendix D
M5

LOW RESISTANCE-8012A ONLY:

RANGE	RESOLUTION	ACCURACY: for 1-Year	FULL-SCALE VOLTAGE	MAXIMUM TEST CURRENT
2Ω (LO Ohms)	1 mΩ	±(1% of reading +2 digits)	0.02V	10 mA
20Ω (LO Ohms)	10 mΩ	±(0.5% of reading +2 digits)	0.2V	10 mA

OVERLOAD PROTECTION: 300V dc/ac rms on all ranges.

RESPONSE TIME: 1 second maximum.

OPEN CIRCUIT VOLTAGE: 16V maximum on both ranges.

CONDUCTANCE:

RANGE	RESOLUTION	ACCURACY: for 1-Year	OPEN CIRCUIT VOLTAGE	MAXIMUM TEST CURRENT
2 mS	1 μS	±(0.2% of reading + 1 digit)	3.5V	1.3 mA
20 μS	10 nS		1V	10 μA
200 nS	.1 nS	±(1% of reading +10 digits)	1V	0.10 μA

OVERLOAD PROTECTION: 300V dc/ac rms on all ranges.

CONDUCTANCE UNITS: We use the international unit of conductance, the Siemen = S = 1/Ω. Another unit of conductance is the mho.

ENVIRONMENTAL:

TEMPERATURE COEFFICIENT: 0.1 times the applicable accuracy specification per °C for 0°C to 18°C and 28°C to 50°C (32°F to 64.4°F and 50.4°F to 122°F).

OPERATING TEMPERATURE: 0°C to 50°C (32°F to 122°F).

STORAGE TEMPERATURE: (without batteries): −40°C to +70°C (−40°F to +158°F).
(with batteries): −40°C to +50°C (−40°F to +122°F).

RELATIVE HUMIDITY: 0 to 80%, 0°C to 35°C (32-95°F) on 2000 kΩ, 20 MΩ and 200 nS ranges.
0 to 90%, 0°C to 35°C (32-95°F) on all other ranges.
0 to 70%, 35°C to 50°C (95-122°F).

GENERAL:

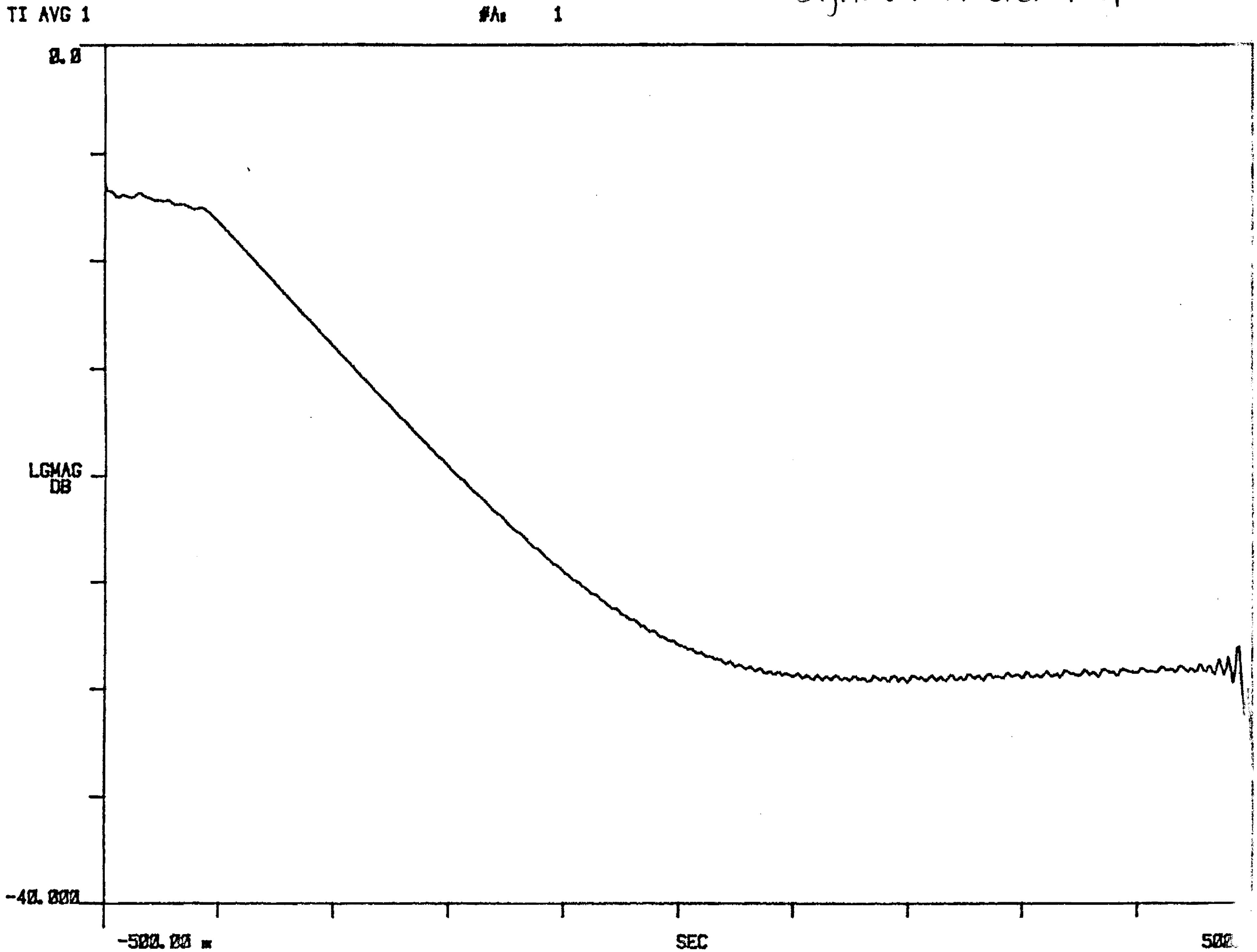
MAXIMUM COMMON MODE VOLTAGE: 500V dc/ac rms.

POWER REQUIREMENTS: 90-132V, 60 Hz.

SIZE: 22 cm X 6 cm X 25 cm (8½" X 2½" X 10").

WEIGHT: 1.08 kg (2 lb., 6 oz.).

Digital voltmeter response curve.



APPENDIX C

APPENDIX C

Analyzer Calibration

For the spectrum mode, a 94 db calibration tone was fed from the tape recorder into a 2203 sound level meter which was connected into the analog input of the analyzer. The spectrum of the 94 db calibration was subsequently plotted (see figure C-1). The calibration constant for the analyzer was divided as follows:

$$SPL = 20 \log \frac{P}{P_r}$$

From figure C-1, SPB = 3.8132 db

Solving for $\frac{P}{P_r}$, $\frac{P}{P_r} = 1.5506$

For SPL = 94 db $\quad 94 = 20 \text{ Log } \frac{P}{P_r} \quad \frac{P}{P_r} = 50118$

$$\frac{50118}{1.5506} = 3.2322 \times 10^4$$

This factor was put into the analyzer and the spectrum was again plotted. The result is figure C-2.

For the time record made, the output of the 2203 was fed into the Fluke 8010A RMS voltmeter. The analog output from the RMS meter was fed into the analyzer.

The recorded calibration tone was plotted with this configuration and the result appears in figure C-3. The calibration constant for this mode was calculated in the same manner as for the spectrum mode. The value was found to be 2.975×10^5. The result of this is shown in figure C-4. Figure C-5 is a time record of sample data that was prepared in this way. In order to further average this data for the determination of the values for the contour plots, the integration capabilities of the analyzer were employed. Figure C-6 is a plot of the 94 db calibration tone from the tape into the 2203 and into the RMS meter and into the analyzer. The power value is the integrated value for the area between the cursors. The average is calculated as follows:

Power = 587.52×10^9 (This value is the square of the average multiplied by the length on the horizontal axis between the cursors)

$$\frac{d_x}{d_t} = \frac{30 \text{ sec}}{125 \text{ Msec}} = 240$$

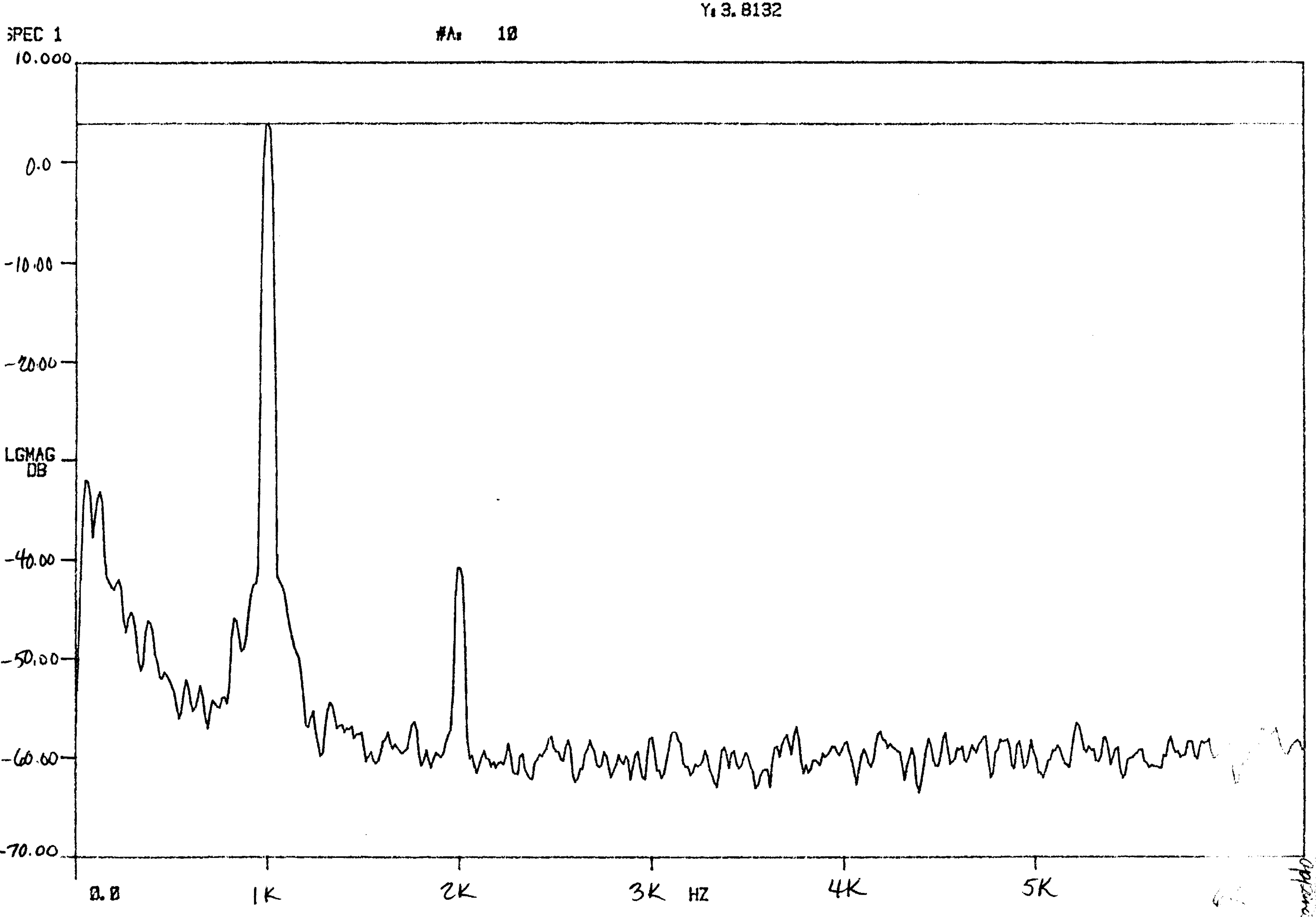
$$[2.448 \times 10^9]^{1/2} = 4.948 \times 10^4 \text{ volts}$$

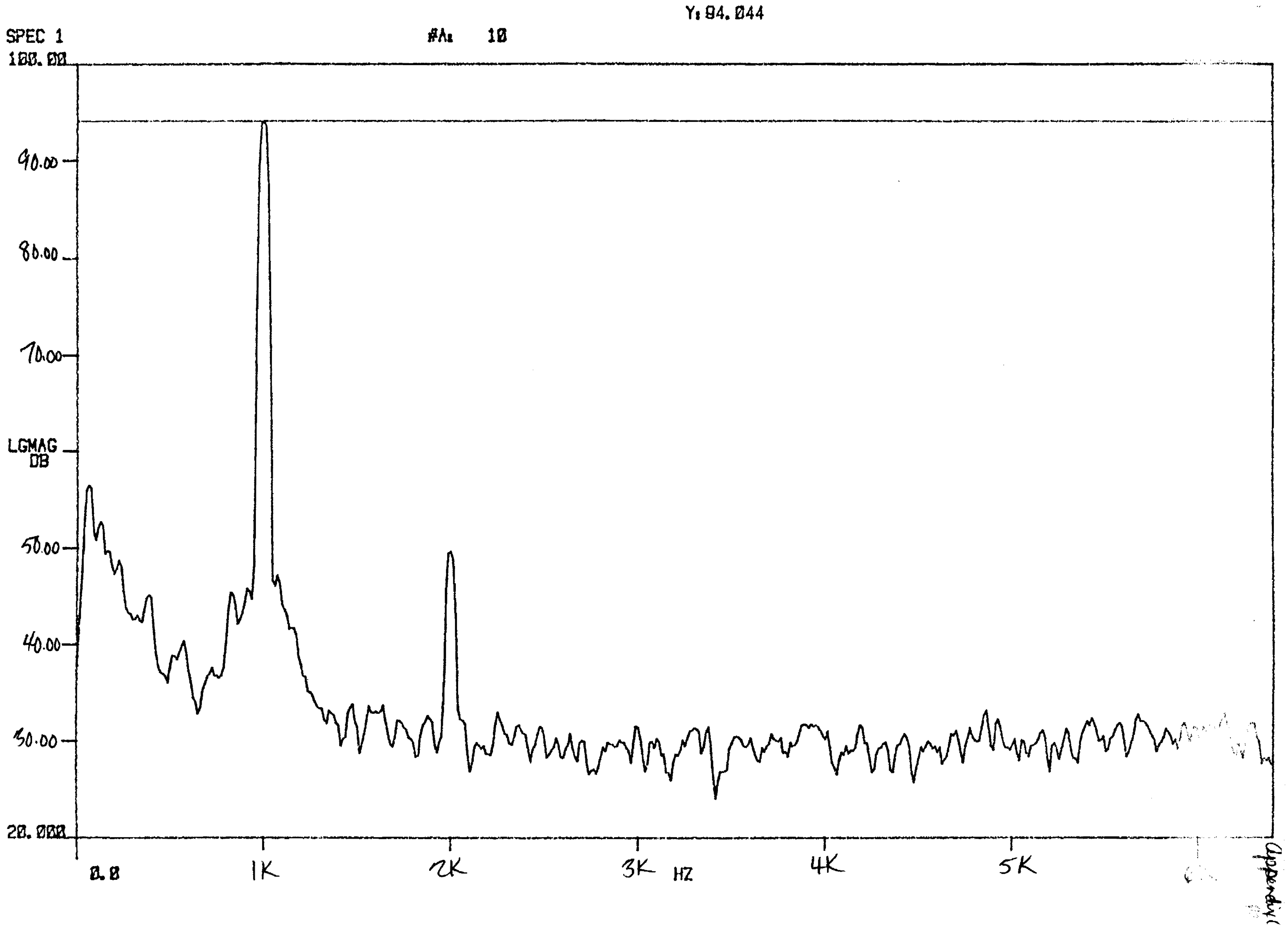
To convert to db: $db = 20 \log \left[\frac{V}{V_r} \right]$

$$94 \text{ db} = 20 \log \left[\frac{4.948 \times 10^4}{V_r}\right] \quad V_r = .987$$

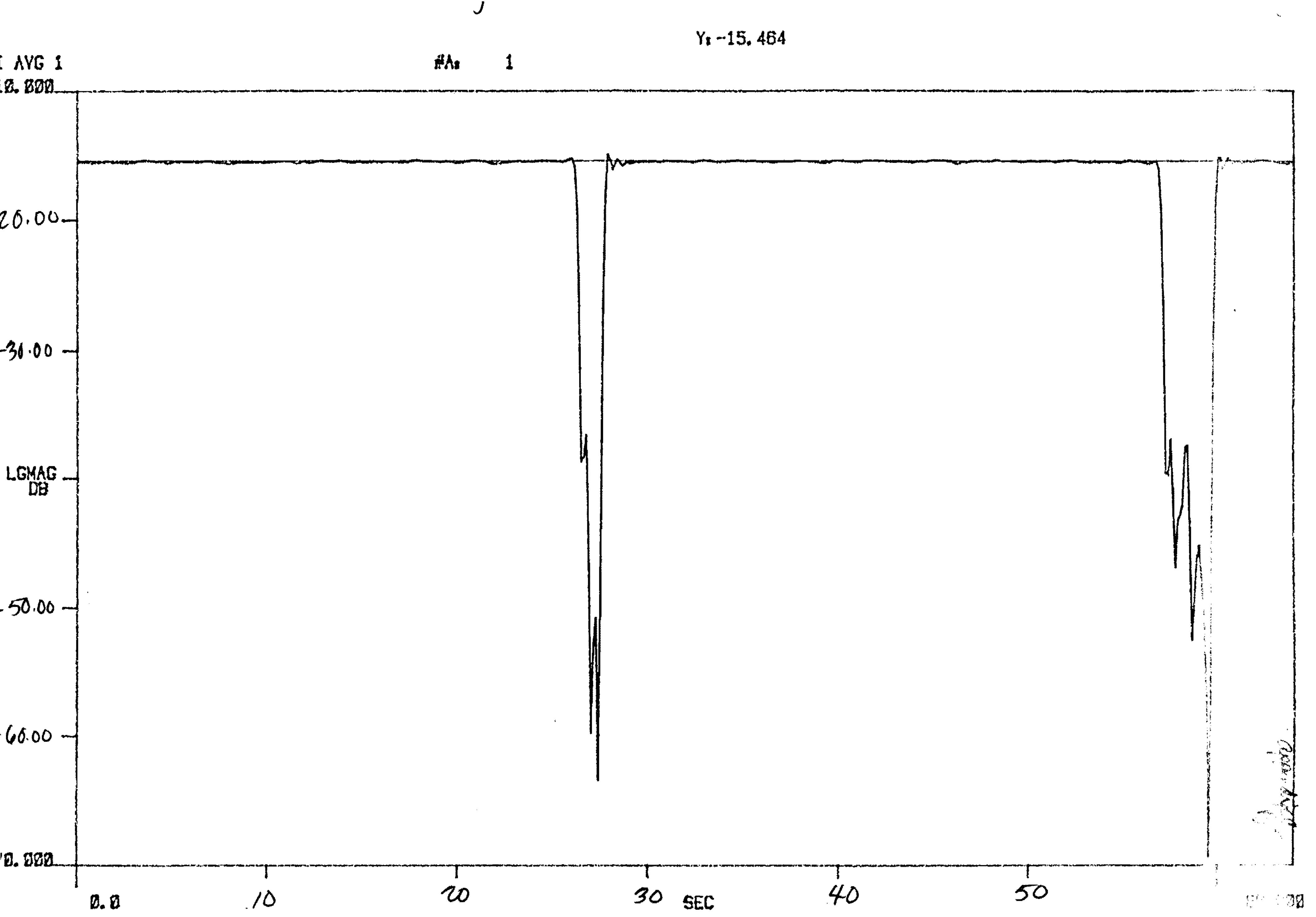
$$db = 20 \log \left[\frac{V}{.987}\right]$$ (This equation was used to calculate the values for the contour plot)

This equation was verified using the attenuator on the 2203. The 94 db dignal from the recorder was boosted and attenuated in 10 db steps over a 40 db dynamic range and the averages were calculated. For levels plus or minus 10 db from the calibration tone, the results were accurate to plus or minus 0.1 db. For levels plus or minus 20 db, the levels were accurate to plus or minus 1.0 db.

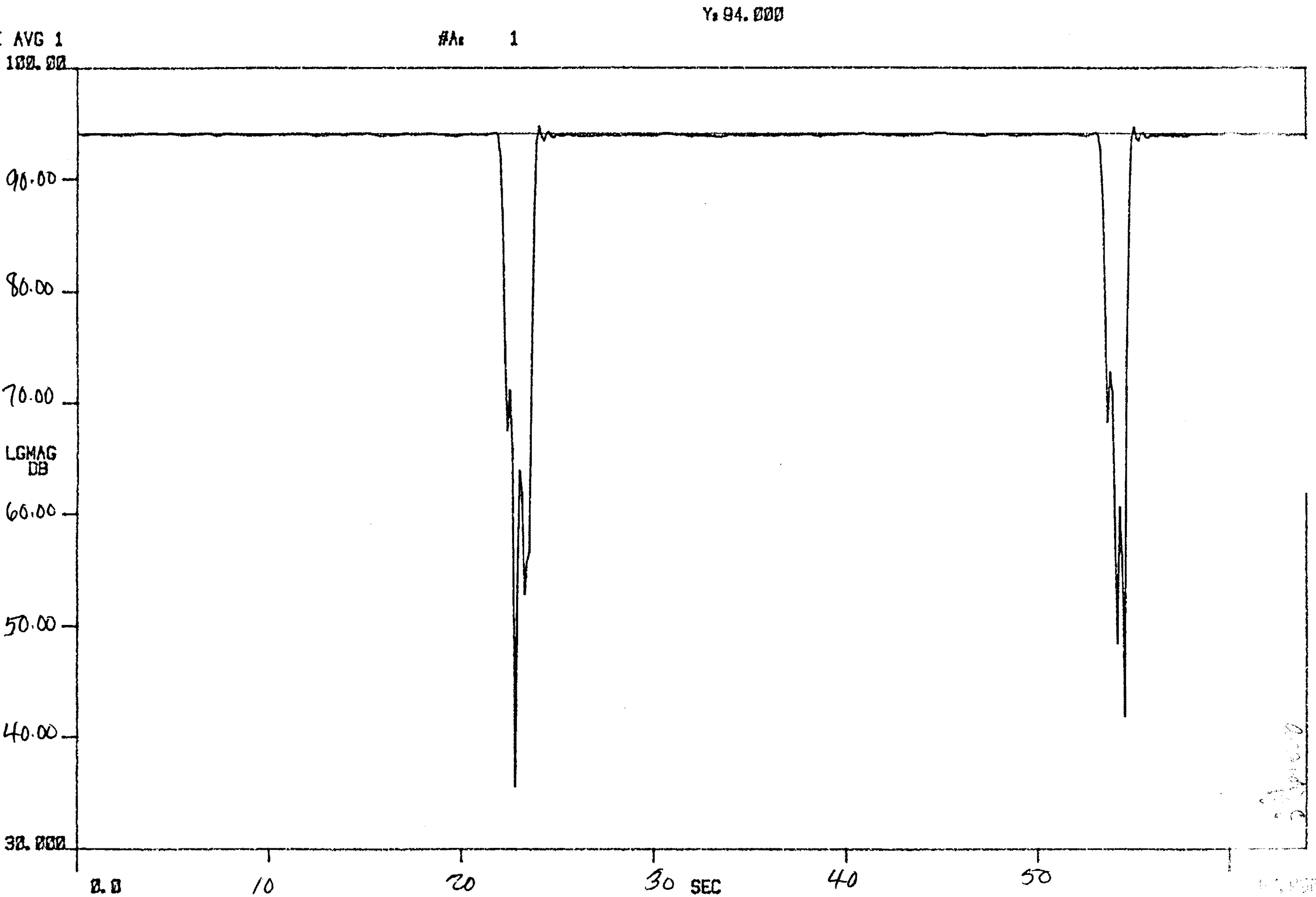

Y: 3.8132
SPEC 1
#A: 10
10.000
0.0
-10.00
-20.00
LGMAG
DB
-40.00
-50.00
-60.00
-70.00
0.0
1K
2K
3K
HZ
4K
5K



Y: 94.044
SPEC 1
#A: 10
100.00
90.00
80.00
70.00
LGMAG
DB
50.00
40.00
30.00
20.000
0.0
1K
2K
3K
HZ
4K
5K

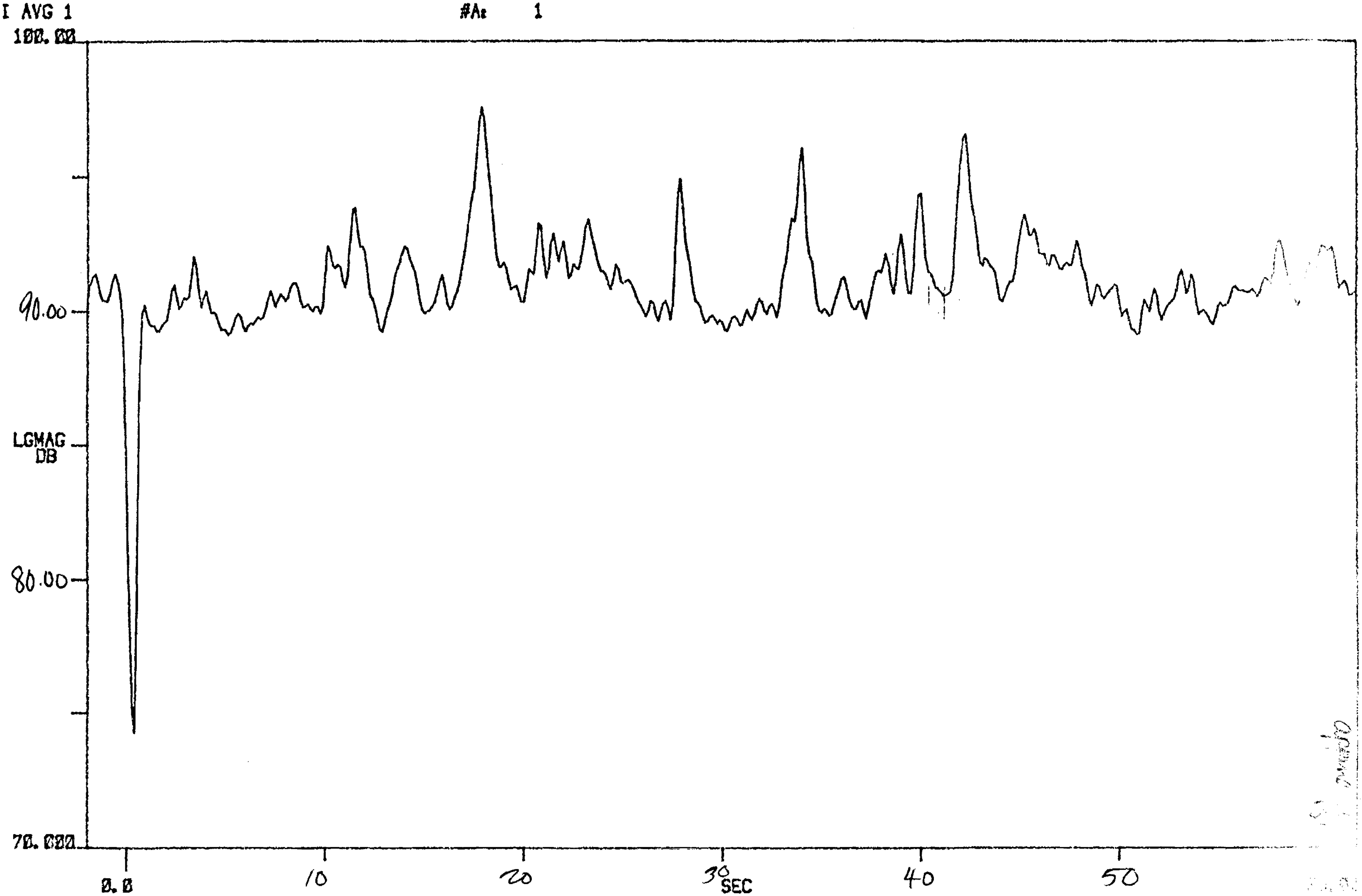


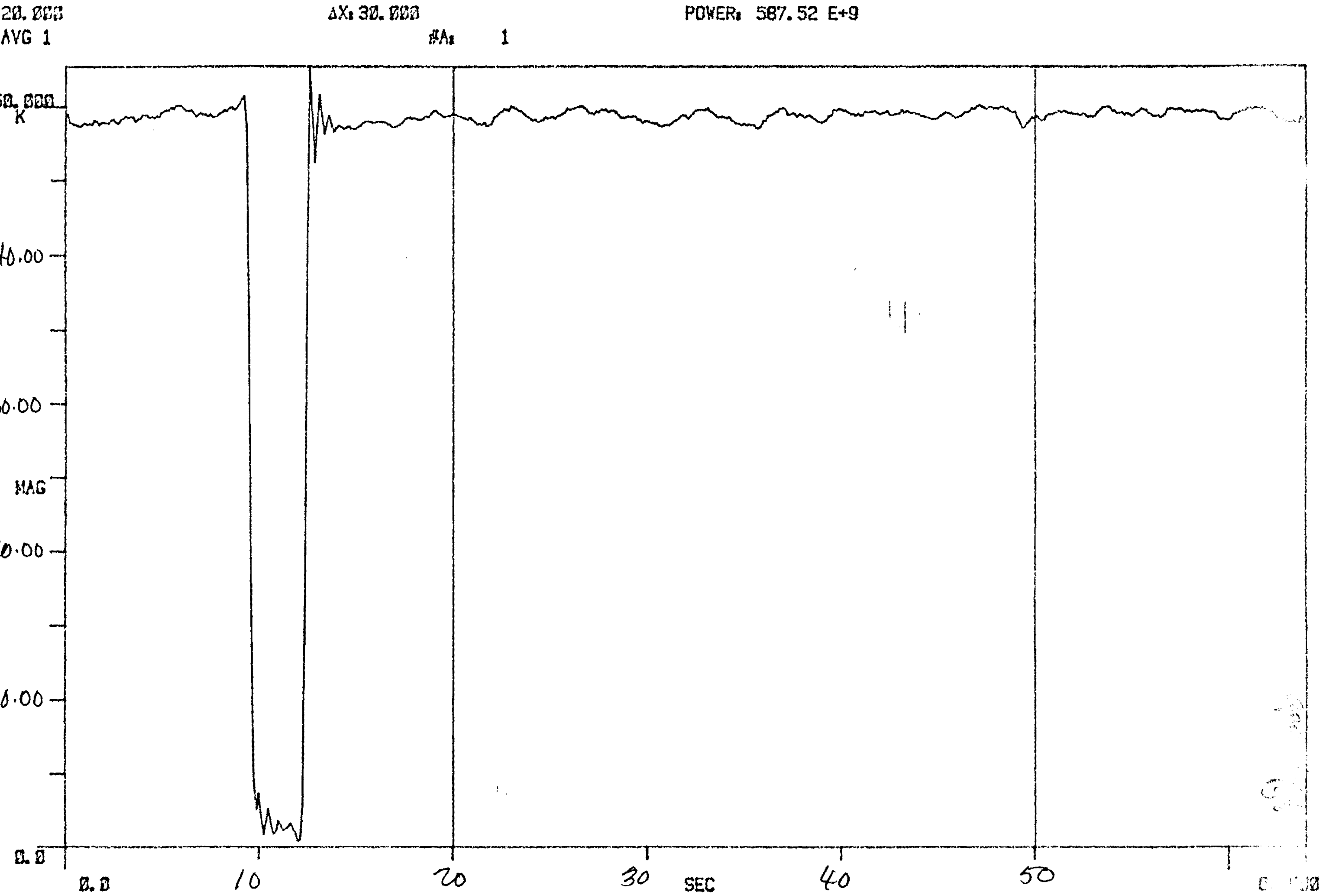
Y: -15.464
AVG 1
#A: 1
0.000
20.00
30.00
LGMAG DB
50.00
60.00
70.000
0.0
10
20
30
SEC
40
50



Y: 94.000
AVG 1
#A: 1
100.00
90.00
80.00
70.00
LGMAG DB
60.00
50.00
40.00
30.000
0.0
10
20
30
SEC
40
50

Figure 2




20.000
AVG 1
ΔX: 30.000
#A: 1
POWER: 587.52 E+9
K
MAG
0.0
0.0
10
20
30
SEC
40
50

A-2231

PROGRESS REPORT

FOR THE PERIOD

JANUARY 1, 1980 to JUNE 30, 1980

A STUDY OF POULTRY PLANT NOISE CONTROL TECHNIQUES UTILIZING NASA TECHNOLOGICAL EXPERIENCE AND EXPERTISE

NASA Research Grant No. NSG 3228

from the

National Aeronautics and Space Administration

Lewis Research Center

21000 Brookpark Rd.

Cleveland, Ohio 44135

by the

Georgia Institute of Technology

Engineering Experiment Station

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J. Craig Wyvill

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Introduction

In our final report entitled "An Analysis of Poultry Processing Plant Noise Characteristics and Potential Noise Control Techniques,"[1] we determined that reverberation was a key factor in the extensiveness of excessive noise levels in poultry processing plants. We further pointed out that acoustical panels for reducing noise had been tried but were encountering durability problems due to the plastic covers tearing under normal use. Plastic covers, it should be pointed out, are necessary to allow conventional absorbing materials to be used and yet allow the panels to be washable thereby meeting USDA cleanability requirements.

This report summarizes our efforts to date to determine the cause of current panel failures and to develop designs of our own which appear capable of enduring the types of abuse typical to the poultry processing environment.

Current Technology

There are only a few companies today who have designed absorbing panels specifically for use as a hanging panel in the food processing industry. Of these designs none have established themselves as capable of withstanding the poultry environment. The standard design is typically a fiberous sheet covered with a plastic film which has been heat sealed and shrunk for a tight fit (see Figure 1).

The absorbing mediums found to date are fiberglass, mineral wool and foam. The plastic covers, all of which are between 0.5 mil and 2 mil in thickness are polyolefin or polyvinyl flouride (PVF). One design, which will be discussed later incorporates a perforated metal cover plate over the plastic covered core.

Two interesting incidents merit noting regarding current technology:

o In 1976, the Virginia Polytechnic Institute in Blacksburg, Virginia installed a urethane foam panel with a PVF cover in a Virginia poultry plant.[2] After installation the plant failed to continue their use. Our contact with the plant uncovered an

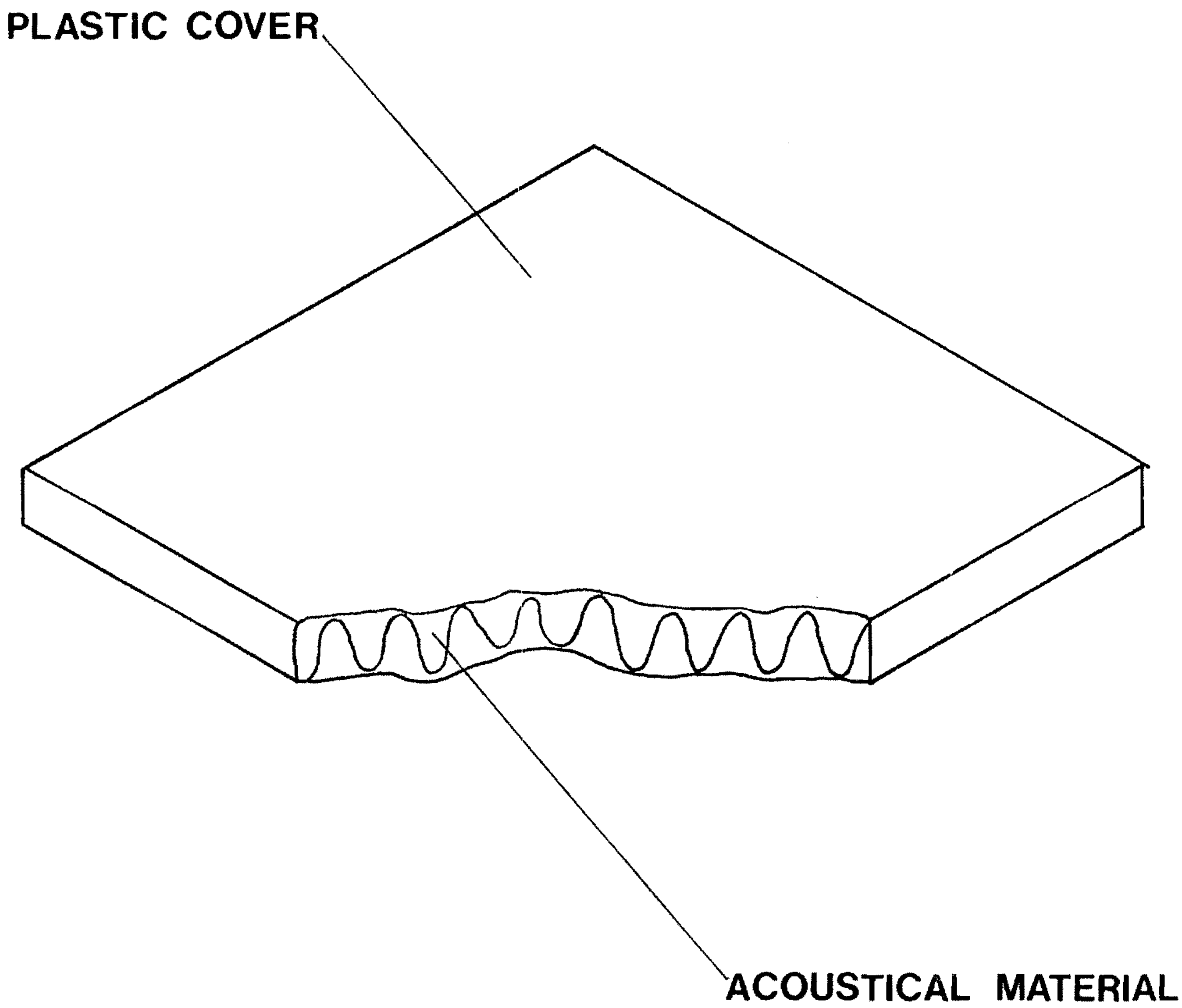


Figure 1 - Typical Design of Current Acoustical Panels

attitude that the panels were too delicate, the covers had failed and therefore had to be removed.

o In 1979, a commercial firm installed a mineral wool panel with a polyolefin cover in a Pennsylvania poultry plant. After a few months of usage, the panel covers began failing at a regular rate. The manufacturer went back to the drawing board and has since come out with a new cover on the same panel, this time using PVF film. The new results are not conclusive at this time.

Clearly, these experiences point out the need for developing a durable absorbing panel. Every time an experiment fails it shakes the confidence of prospective users in any design working. Unfortunately, progress in developing new designs has been slow, and it appears doubtful, at this time, that any breakthroughs are imminent. However, if suitable designs are developed and tested which will work it appears certain producers will try to market such a product if a demand exists and we feel that demand development is merely a matter of instilling confidence in the durability and effectiveness of a panel design.

Design Development

In order to develop a durable design, we set out to evaluate a number of different design options. These options can be categorized as follows:

1) use a tougher film covering
2) reinforce existing film covers
3) eliminate the need for a film cover.

Since a panels acoustical qualities are an essential element of its design, we also had to remodel a room on the Georgia Tech campus to serve as a reverberant room. This room will be used:

1) to test the absorption characteristics of both different panel designs and different panel orientations.

2) to provide a demonstration site for exhibiting our final design and its effectiveness

To date, we have successfully identified several possible designs and are in the process of fabricating them. In addition, we are concluding tests on qualifying our reverberant room in time for its use in testing panels.

Panel Design Considerations

We first focused our attention on developing a more durable panel by considering a tougher covering film. Upon reviewing the list of available plastic films, we found at least two which had superior qualities to those currently being tried.

Table 1 presents pertinent physical properties of the general film categories available today. Since polyvinyl flouride is already in use in current panel designs we began a search for a film with qualities superior to it. Polyester film we found had a superior tensile strength over PVF film while having comparable tear strength. Polyurethane film, on the other hand, has superior tearing strength to PVF film while having comparable shear strength.

Both films, however, have drawbacks which will have to be addressed. Polyurethane, for instance, has a problem in handling sustained temperatures above 190°F and both films have questionable ultraviolet endurance. Nonetheless, we intend considering both films in a panel design since they clearly seem to offer an inexpensive alternative to current designs.

In addition to tougher materials, a tougher film may be achievable through a thicker film. This option will be reviewed for 2, 3, and 5 mil thick cover.

As a side note, we have observed that all current panel designs have a heat shrink cover. Because of this, the film is tightened over the core possibly weakening its ability to withstand an impact. We therefore intend to try PVF film which is only form fitted to a core, to see if this use of the film is more satisfactory.

TABLE I

GENERAL FILM PROPERTIES*

	Polyethylene	PVC	PVF	Polyester	Polyurethane
Tensile Strength, (psi)	1,500-6,100	1,400-16,000	7,000-18,000	20,000-40,000	5,000-12,000
Tearing Strength					
Initial (lb/in)	65-575	110-490	997-1,400	1,000-3,000	350-600
Propagating (g/mil)	50-300	60-1,400	12-100	12-17	220-710
Resistance					
Grease & Oil	poor to good	good	good	good	good

*Source, reference 4

Our next focus was on ways to reinforce the film covering. The three techniques researched were:

1) a protective screen
2) a cloth backing
3) a perforated cover plate

Of the three, using a protective screen seemed the most straightforward. Realizing the screen had to be non corrosive and inexpensive we researched plastics. We found a tough polypropylene screen which appeared suitable for this use. Our only concern now is whether the screen should be adhered to the film or not.

Cloth backed films are perhaps the most novel idea we have researched. There are a number of techniques available today but the two techniques considered most promising are:

1) film adhered to a thin cloth
2) calendered cloth using a waterproof liquid plastic film

Both techniques are novel to this type of application and promise to provide the kind of toughness plastic films alone cannot. Our primary concern is getting the finished product thin enough to transmit significant portions of the pertinent sound frequency bands critical to reducing poultry processing noise.

We have also reviewed a technique called scrimming, whereby a netting is either adhered to the back of a film or sandwiched between two films. This concept is intriguing and will be evaluated. However, our concern is that the film still remains vulnerable in the open areas of the netting to contact with foreign objects and therefore we believe a fine netting will be necessary to minimize the risk of puncture initiation.

The last concept we focused on was covering the plastic film with a perforated plate. As mentioned earlier we are already aware of at least one panel design on the market which utilizes this technique. Unfortunately the design is a variation of panels used to build acoustical enclosures. Hence the firm making these items has not keyed on reducing weight or cost. Nonetheless, their concept is interesting and will be considered.

We lastly focused on a technique which would eliminate the need altogether for plastic covering. In order for this to be realized, a fiberous core must not be used. We are aware of a concept developed by a Lockheed engineer for both aerospace and general application which utilizes an enclosed cavity of air designed with a graduated depth and having a perforated cover plate which appears to be quite effective as a broadband noise absorber. We intend to pursue this design further to discover if it can be designed to reduce low frequency noise which must be dealt with in poultry processing plants.

Constructing a Reverberant Room

Our efforts to convert a storage room on the Georgia Tech campus into a suitable reverberant room were indeed ambitious. We first had to initiate a massive cleanup effort to empty the room of all attached and unattached objects. Since the room had been an instrumentation room, extensive removal of metal boxes and gages was required in addition to patching holes in the walls. Also the ceiling of the room was constructed of a sheetrock material which had become waterlogged from leaks in the roof and therefore we had to use with ½-inch plyboard to cover the ceiling for purposes of providing a suitable reflecting surface.

Our first acoustical qualifying tests showed the room was "hard" enough to be used in reverberation testing. Table 2 provides the surface absorption coefficients calculated from this test. Our guideline for qualifying the room was a standard procedure published by the American Society for Testing Materials.[3]

However, our enthusiasm was dampened slightly when we placed fiberglass panels, with a known absorption value in the room and attempted to reproduce these values. Our calculations provided values substantially below the manifacturer's values and our suspicion was that the absorbing panels lowered the diffuseness of the reverberant field in the room to an unsuitable level. A subsequent check of level variation in the room seemed to confirm this suspicion.

As of this writing, we are introducing diffusion panels into the room. Our hope is that this will result in a test chamber of sufficient quality to do comparative noise analysis.

TABLE 2

SURFACE ABSORPTION COEFFICIENTS FOR REVERBERANT ROOM*

Octave Frequency Band	Average Surface Absorption Coefficient**
250	.0456
500	.0452
1,000	.0503
2,000	.0566
4,000	.0621
8,000	.0618

*As determined by the decay method

**Values include any absorption by air.

Retrospect

Our initial approach to solving the reverberant noise problem in poultry processing plants has been to seek a tough, effective absorbing medium suitable for placement in the poultry processing environment. We have uncovered problems in current absorbing panel cover designs which we feel are solvable. At this time a number of concepts are being considered and will be tested for their suitability both acoustically and structurally. Our hope is that at least one design will be successful.

We also are beginning investigations aimed at quieting the three major sources in Poultry Processing Plants. Our primary thrust in this effort will focus on vibration identification and dampening/isolation techniques.

REFERENCES

1. "A Study of Poultry Processing Plant Noise Characteristics and Potential Noise Control Techniques," J. C. Wyvill, et al., Georgia Tech EES Final Report for NASA research grant NSG 3228 and Georgia Department of Agriculture grant A-2028-006, April 1980

2. "Use of Acoustical Panels in a Poultry Processing Plant," William Mashburn, Virginia Polytechnic Institute and State University, April 1976

3. "Standard Method of Test for Sound Absorption of Acoustical Materials in Reverberation Rooms," ASTM C423-66, American Society for Testing Materials, 1966

4. "Modern Plastics Encyclopedia," McGraw Hill Publication, October 1978.

APPENDIX A

QUALIFYING DATA FOR REVERBERANT TEST ROOM

Qualifying Test of Reverberant Room

These tests were designed to verify the suitability of the test room on the Georgia Tech campus to testing absorption coefficients of material by the reverberant decay method. Using the equipment layout presented in Figure 1-A, we measured decay times at five different positions at the room using three bursts per position (see Figure 1-B)..

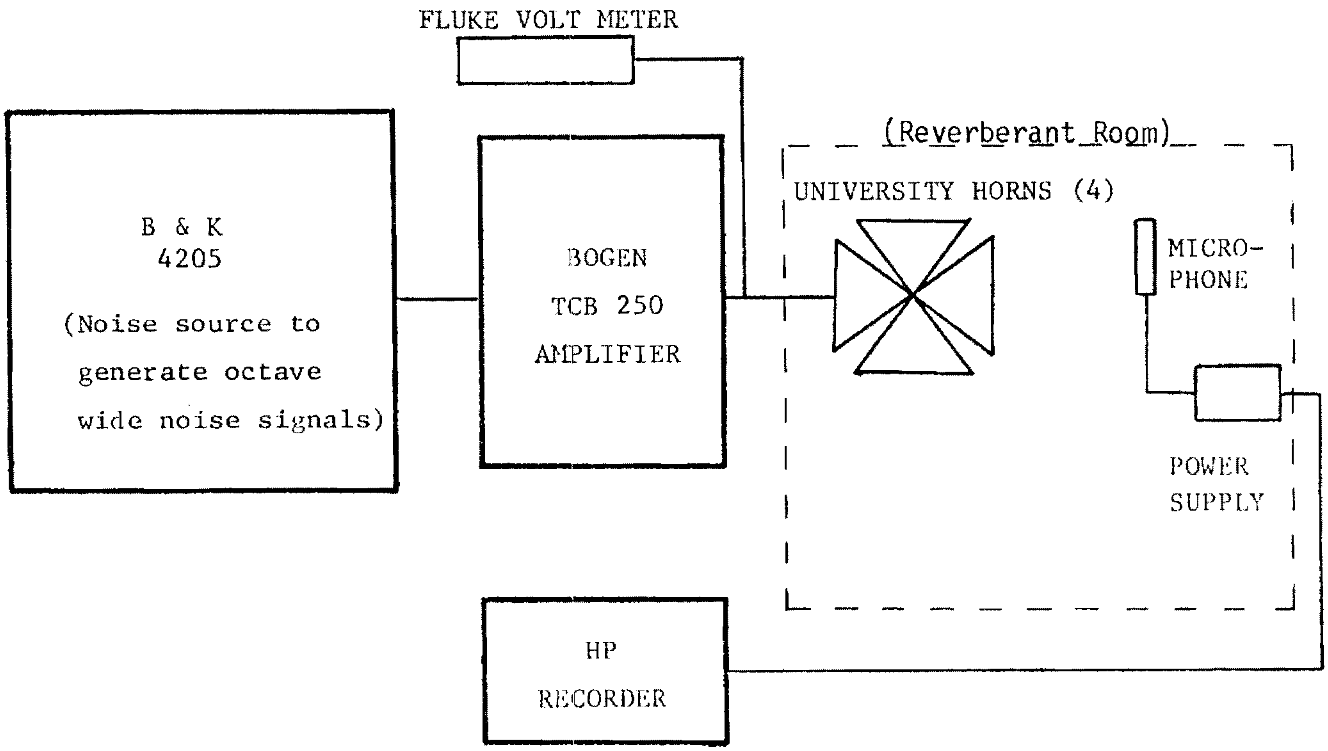
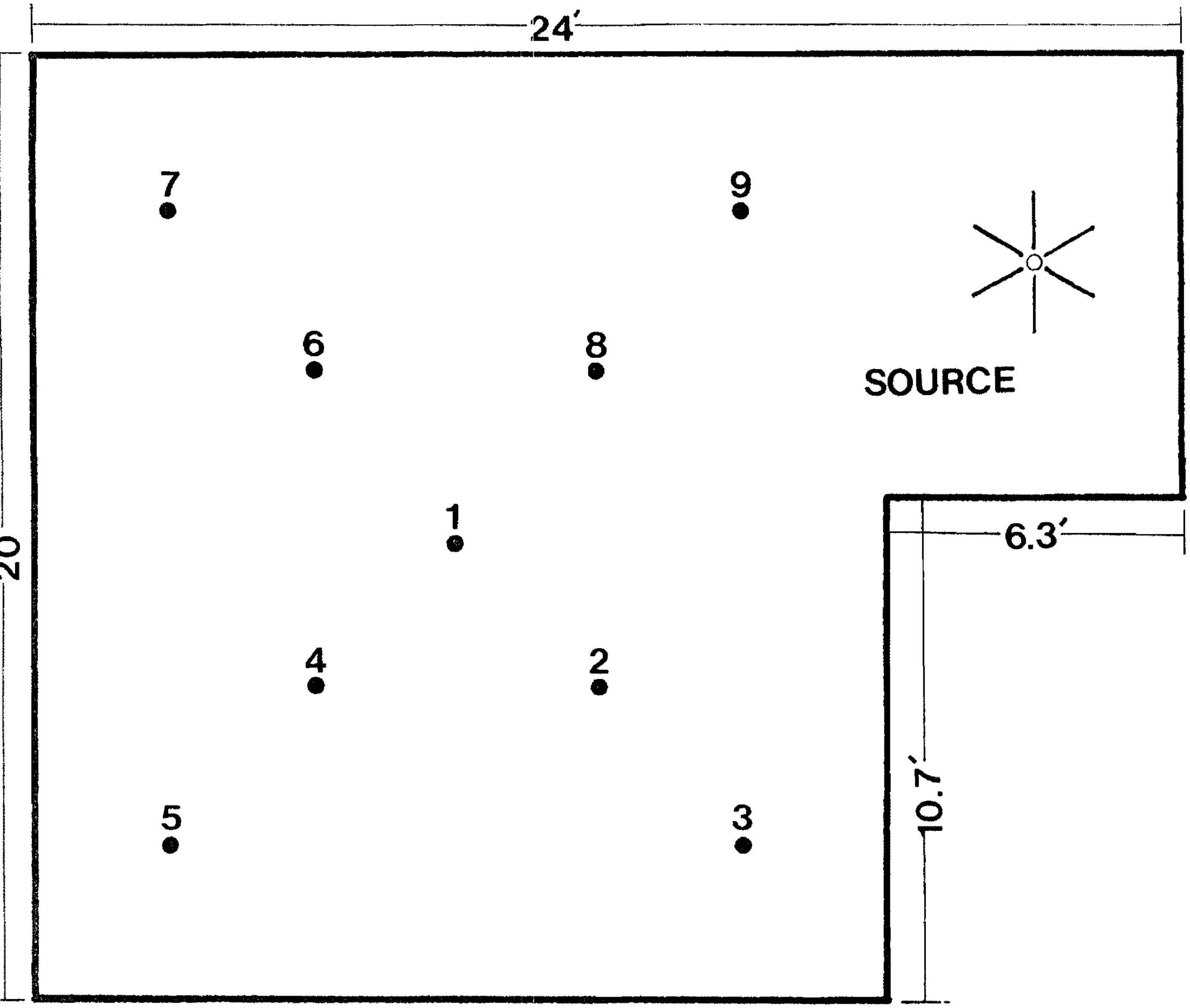


Figure 1-A
Equipment Arrangement

The average surface absorption values obtained are presented in Table 1-A. Tables 1-B through 1-F show the specific decay times observed during our tests which were used in the construction of Table 1-A.



MICROPHONE HEIGHT – 3.4 ft

ROOM HEIGHT – 8.8 ft

Figure 1-B - Measurement Points in Reverberant Room

TABLE I-A

DATA SUMMARY

Average α_{SAB} Determined for Each Microphone Position*

	#1	#2	#4	#6	#8
250	.0460	.0451	.0450	.0462	.0456
500	.0460	.0436	.0456	.0462	.0448
1000	.0514	.0490	.0502	.0517	.0494
2000*	.0555	.0559	.0566	.0580	.0572
4000*	.0622	.0615	.0638	.0623	.0607
8000*	.0618	.0610	.0625	.0627	.0612

Average of All Microphone Positions #1, #2, #4, #6, and #8:

	α_{SAB}	Standard Deviation
250	.0456	.000531
500	.0452	.00106
1000	.0503	.00119
2000*	.0566	.00100
4000*	.0621	.00115
8000*	.0618	.000757

*Values shown do not include correction for air absorption

As shown on the frequency response chart of the University FIID32-T horn loudspeakers used in this experiment, (see Figure 1-C), output in the 250 Hz octave is not adequate for valid results to be expected; likewise, output in the 8 KHz frequency octave is subject to question. It is felt, however, that octaves 500, 1000, 2000, and 4000 are valid. α_{SAB} values for these octaves are below the ASTM requirements. It should be cautioned that our measurements do not use the 1/3 octave wide source spectrum required in the standard but instead use a one octave wide frequency spectrum.

TABLE 1-B

	Microphone Position #1 Delay Time in Seconds Test Number			
	#1	#2	#3	Average α_{SAB}
250 Hz	2.469	2.39	2.403	.0460
500 Hz	2.578	2.25	2.43	.0460
1000 Hz	2.204	2.067	2.225	.0514
2000 Hz	1.99	1.98	1.048	.0555
4000 Hz	1.79	1.751	1.824	.0622
8000 Hz	1.78	1.824	1.80	.0618

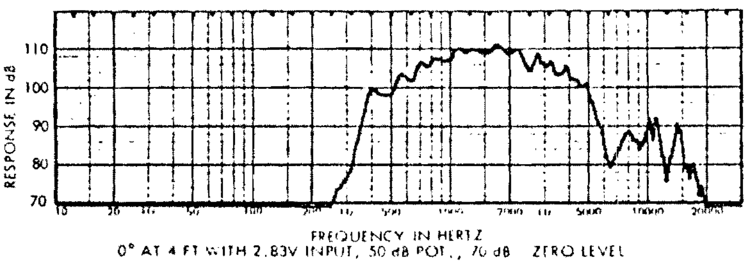


Figure 1. Frequency Response

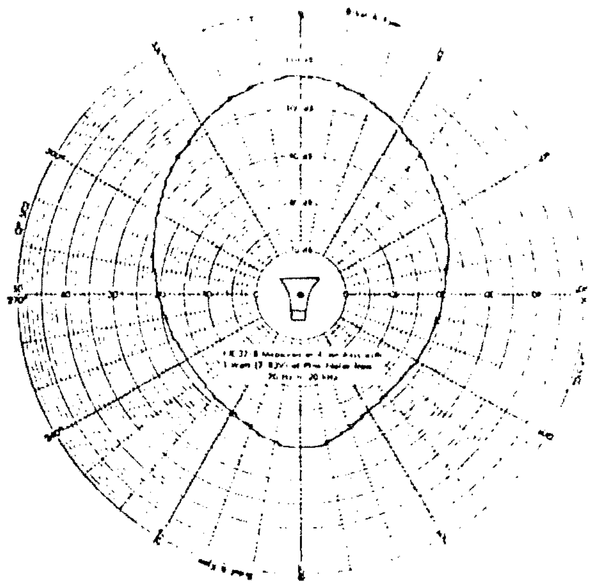


Figure 2. Polar Pattern

Figure 1-C. Frequency Response and Polar Pattern of University FID32-T Horn Loudspeakers

TABLE I-C

	Microphone Position #2 Delay Time in Seconds Test Number			
	#1	#2	#3	Average α_{SAB}
250 Hz	2.460	2.403	2.535	.0451
500 Hz	2.542	2.517	2.596	.0436
1000 Hz	2.262	2.288	2.256	.0490
2000 Hz	1.980	2.029	1.958	.0559
4000 Hz	1.769	1.822	1.8396	.0615
8000 Hz	1.788	1.840	1.840	.0610

TABLE I-D

	Microphone Position #4 Delay Time in Seconds Test Number			
	#1	#2	#3	Average α_{SAB}
250 Hz	2.375	2.627	2.419	.0450
500 Hz	2.508	2.410	2.402	.0456
1000 Hz	2,269	2.220	2.163	.0502
2000 Hz	1.981	1.915	1.995	.0566
4000 Hz	1.742	1.765	1.725	.0638
8000 Hz	1.791	1.778	1.769	.0625

TABLE 1-E

	Microphone Position #6 Delay Time in Seconds Test Number			
	#1	#2	#3	Average α_{SAB}
250 Hz	Null	2.416	2.398	.0462
500 Hz	2.464	2.332	2.420	.0462
1000 Hz	2.134	2.148	2.174	.0517
2000 Hz	1.831	1.976	1.950	.0580
4000 Hz	1.774	1.782	1.804	.0623
8000 Hz	1.778	1.774	1.774	.0627

TABLE 1-F

	Microphone Position #8 Delay Time in Seconds Test Number			
	#1	#2	#3	Average α_{SAB}
250 Hz	2.377	2.556	2.379	.0456
500 Hz	2.410	2.543	2.494	.0448
1000 Hz	2.242	2.269	2.247	.0494
2000 Hz	2.008	1.902	1.919	.0572
4000 Hz	1.818	1.826	1.853	.0607
8000 Hz	1.831	1.813	1.813	.0612

Georgia Institute of Technology
ENGINEERING EXPERIMENT STATION
ATLANTA, GEORGIA 30332

March 25, 1981

Mr. Sanford Tingley
Technology Utilization Office
(Mail Stop 7-3)
NASA Lewis Research Center
21000 Brookpark Road
Cleveland, OH 44135

Subject: Research Grant NSG-3228 Status Report

Dear Sandy:

This letter report is to apprise you of our current status on the subject research grant. As you know, we submitted a draft final report on March 18. This report summarized the technical findings of our research through December 1980. Based on the suggestions received in our December meeting we conducted a series of panel spacing tests in January and February using a four sided plywood box (see picture). However, the test results exhibited conflicting relationships which we felt were the result of the box interfering with the spacing evaluation. Based on subsequent discussions with Ed Rice and with his agreement, we are withholding this data from the final report.

With regard to our contract, I have initiated a request for a two month no cost extension to allow your comments on the draft final report to be received and incorporated into the final report.

Financially, the program is operating well within budget. On February 28, 1981, there was a positive cash balance of approximately $15,000. We currently expect to complete the project below budget.

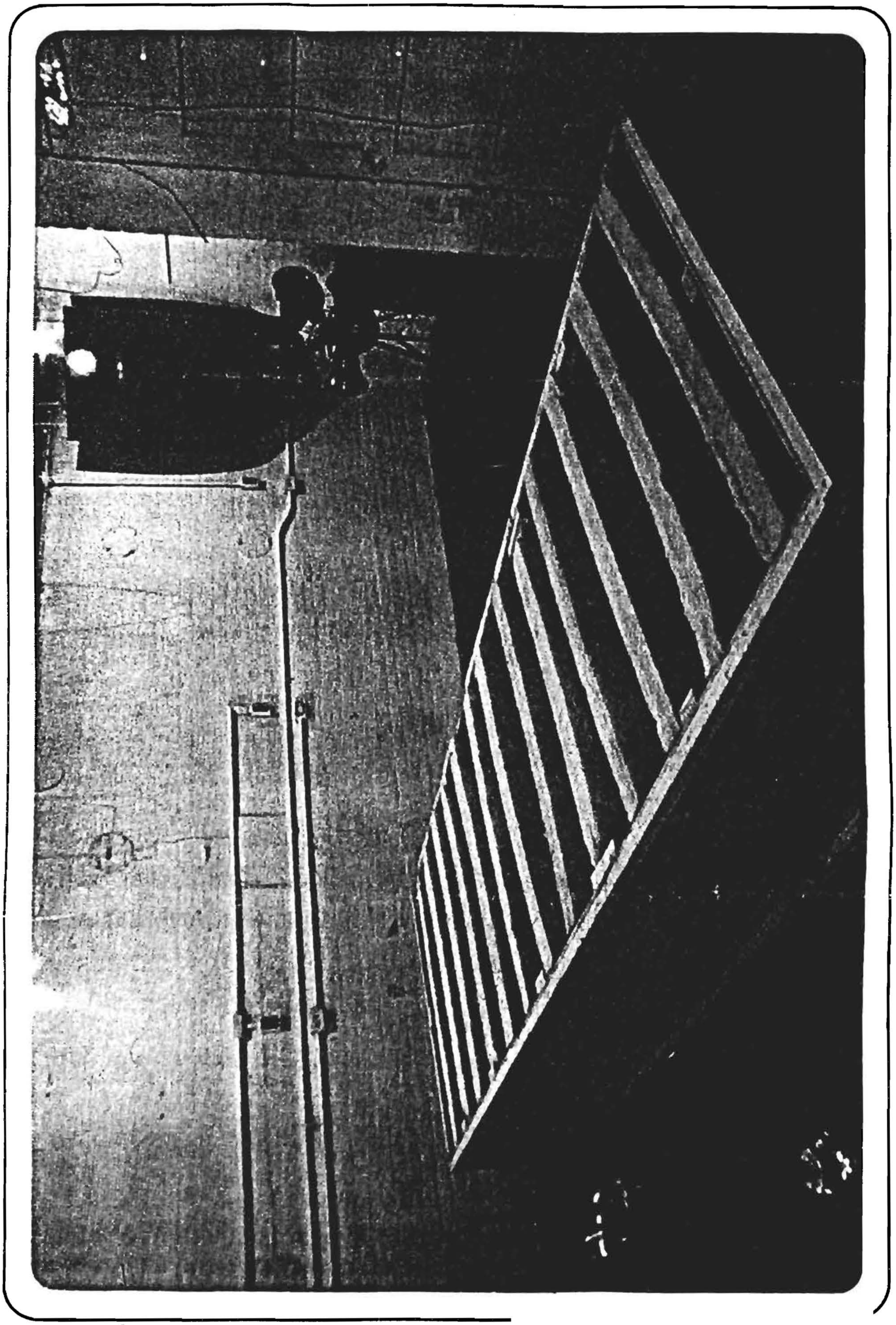
If you have any questions regarding any aspect of the project, please do not hesitate to give me a call.

Very truly yours,

J. Craig Wyvill
Project Director

Enclosure

cc: S. Szabo/NASA Lewis
L. Scott/OCA
O H. Rogers/OCA
C. Feiler
E. Rice



AN ANALYSIS OF NOISE IN POULTRY PROCESSING PLANTS

by

J. Craig Wyvill, P. E.
Georgia Institute of Technology
Engineering Experiment Station
Atlanta, Georgia

ABSTRACT

The poultry processing plant noise problem has been termed unsolvable by some. In order to better understand this problem a study was conducted in two typical processing plants. Noise contours, source sound power outputs and surface absorption coefficients were developed in the study from which it was concluded that only a few "major" sources and an acoustically "hard" environment were responsible for essentially the entire noise problem.

INTRODUCTION

Poultry processing plants currently face a problem with high workplace noise levels which often exceed shift duration exposure limits established by the Occupational Safety and Health Administration (OSHA). Yet efforts to reduce these levels have largely been unsuccessful.

Much of the problem can be traced both to the recent transition of the industry to mechanization (which has increased overall sound power emissions) and to passage of the Poultry Products Inspection Act of 1959 (which set rigid cleanability requirements for all surfaces in a plant restricting the use of many sound absorbing and vibration dampening materials).

The Georgia Institute of Technology Engineering Experiment Station, under the joint sponsorship of the National Aeronautics and Space Administration (NASA) and the Georgia Department of Agriculture, has conducted a study of poultry processing noise in order to determine the extent, cause, and possible solution to the problem.

THE TYPICAL ENVIRONMENT

Upon entering a poultry processing plant, it becomes obvious why plant engineers view the noise problem as overwhelming. The entire plant seems to be filled with noise of indistinguishable origin. Plants typically are composed of tile walls, sheet metal ceilings, and a sealed concrete floor all extremely reflective. In order to determine the nature of the noise problem, measurements were taken throughout the evisceration area of two plants. Measurements were confined to the evisceration area because over half of the processing personnel (typically 60%) are stationed here during a normal work day. Figure 1 and 2 show the noise contours related to each of the plants studied.

Observing figure 1, it appears that plant A has only three areas where noise sources are distinguishable above the general din. In the lung gun area both the lung guns and a component of the chillers are distinguishable. At the other end of the plant a circulating fan is distinguishable as is a source from the picking area which subsequently has been attributed to two hock cutters located immediately on the other side of the wall but open to the evisceration area through two conveyor portals in the wall.

Observing figure 2, plant B appears to have six areas where noise sources are distinguishable above the general din. The hock cutter, lung guns, gizzard peelers, and a component of the chillers are identifiable.

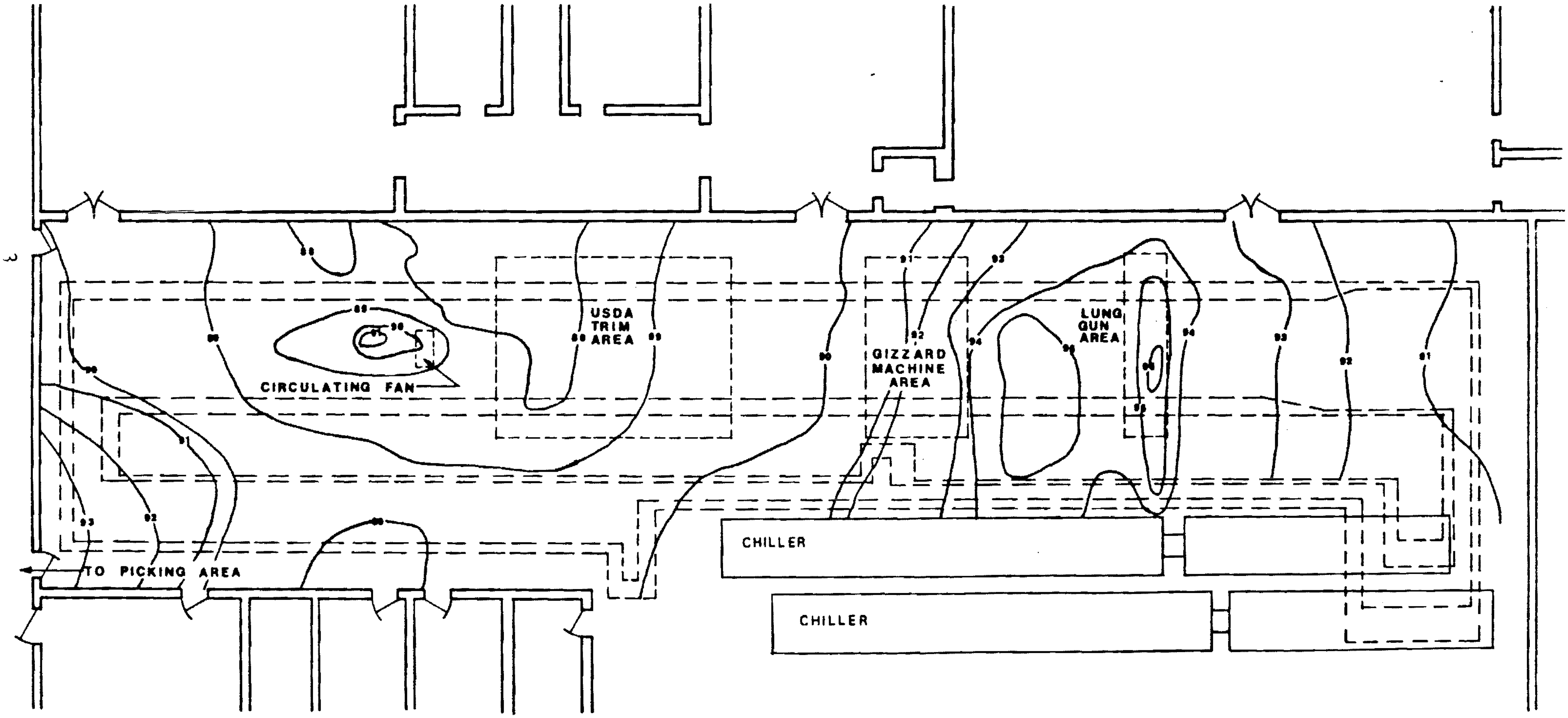


Figure 1
A-Weighted Sound Pressure Level Contours for Plant A

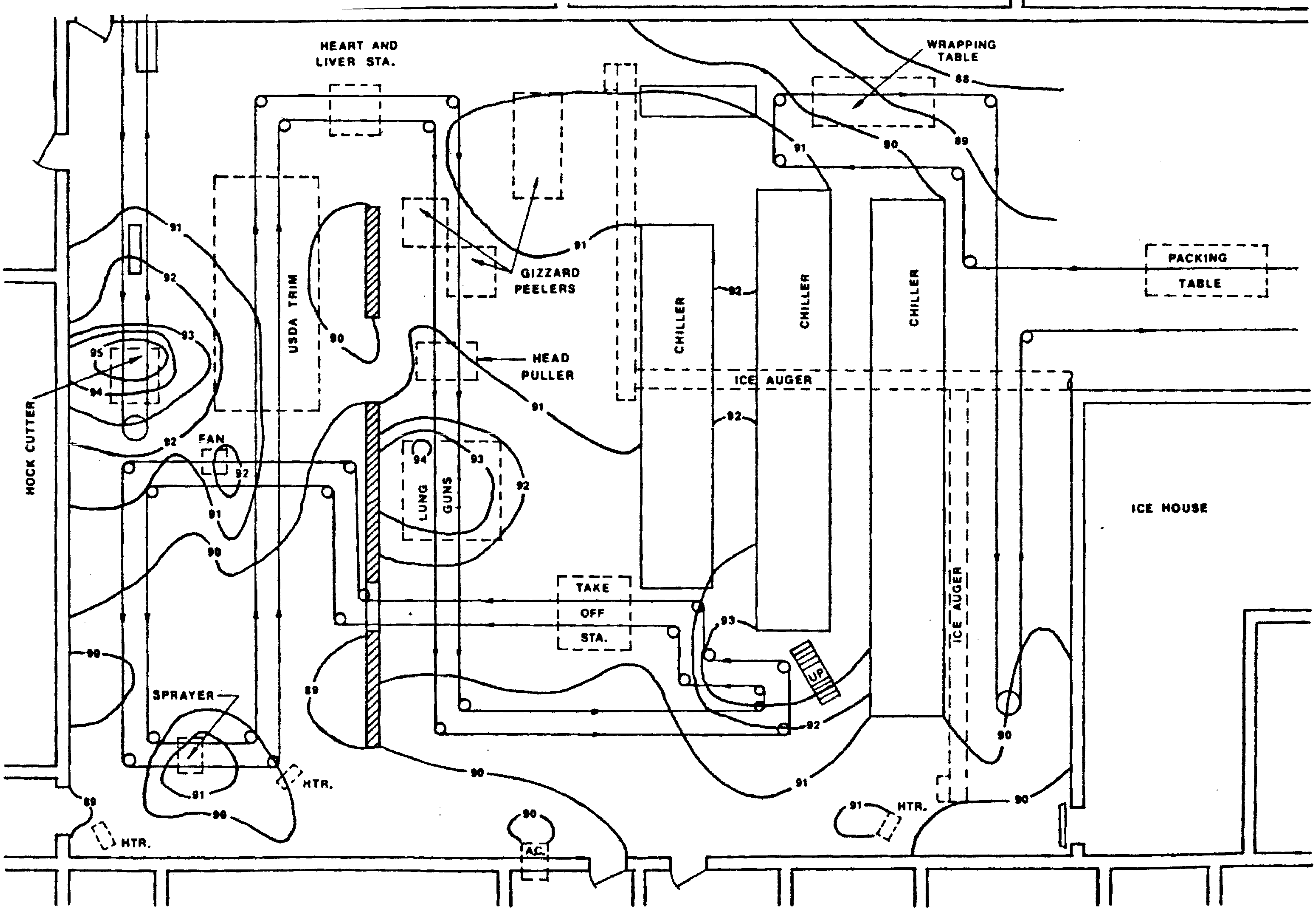


Figure 2
A-Weighted Sound Pressure Level Contours for Plant B

Also identifiable is a source near the spray wash station which has subsequently been attributed to two air hoses used to dry water off of the birds. The source near the exhaust fan remains unclear at this time.

In addition to source identification, these contours provide information on the reverberant noise field in each plant. It appears that plant A has a reverberant noise field which is not uniform in level. This is what should be expected in a long, narrow room with congested source placement. Plant B, on the other hand, appears to have a reverberant noise field which is fairly uniform in level. This also is as would be expected in a square room with even distribution of sources.

THE PROBLEM IDENTIFIED

With the information obtained from the noise contours it would appear only a few "major" sources exist in the evisceration area of a processing plant and that much of the noise observed is predominantly reverberant. In order to determine if these few major sources are capable of powering the reverberant noise field to the levels observed, an analysis of the reverberant environment was performed.

Using a broad band noise source with total acoustic power output of 103 dB, measurements of the direct/reverberant fields were taken in both plants. The results are presented in figure 3 and 4. Note that in both plants, the direct and reverberant noise fields were of equal intensity approximately six feet from the source. Clearly, therefore, reverberation is a serious problem. Using octave band filtering of the data, surface absorption coefficients were calculated for each plant using the following equation for large rooms:

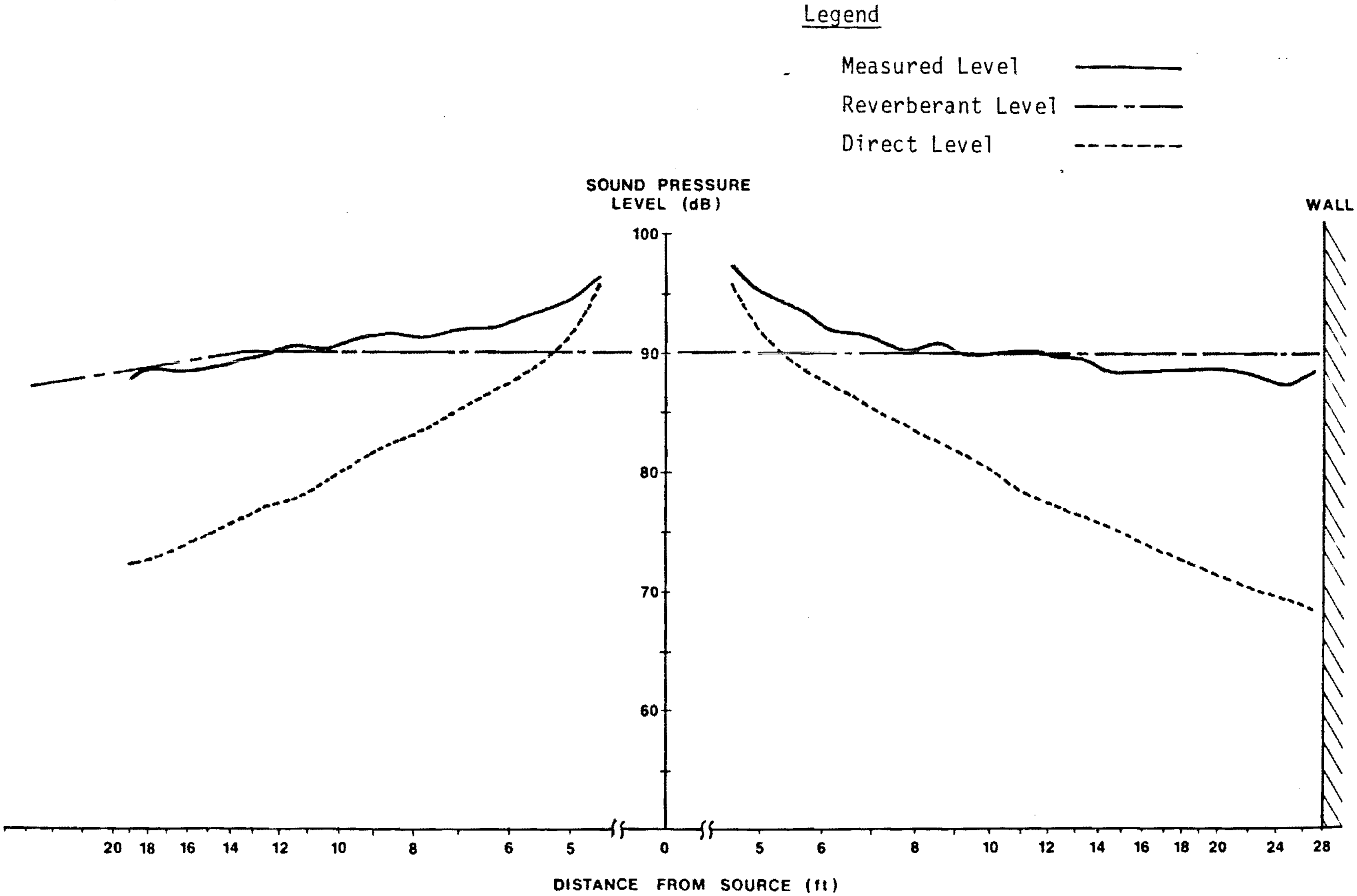


Figure 3
Direct/Reverberant Sound Pressure Levels
Using Known Source in Plant A

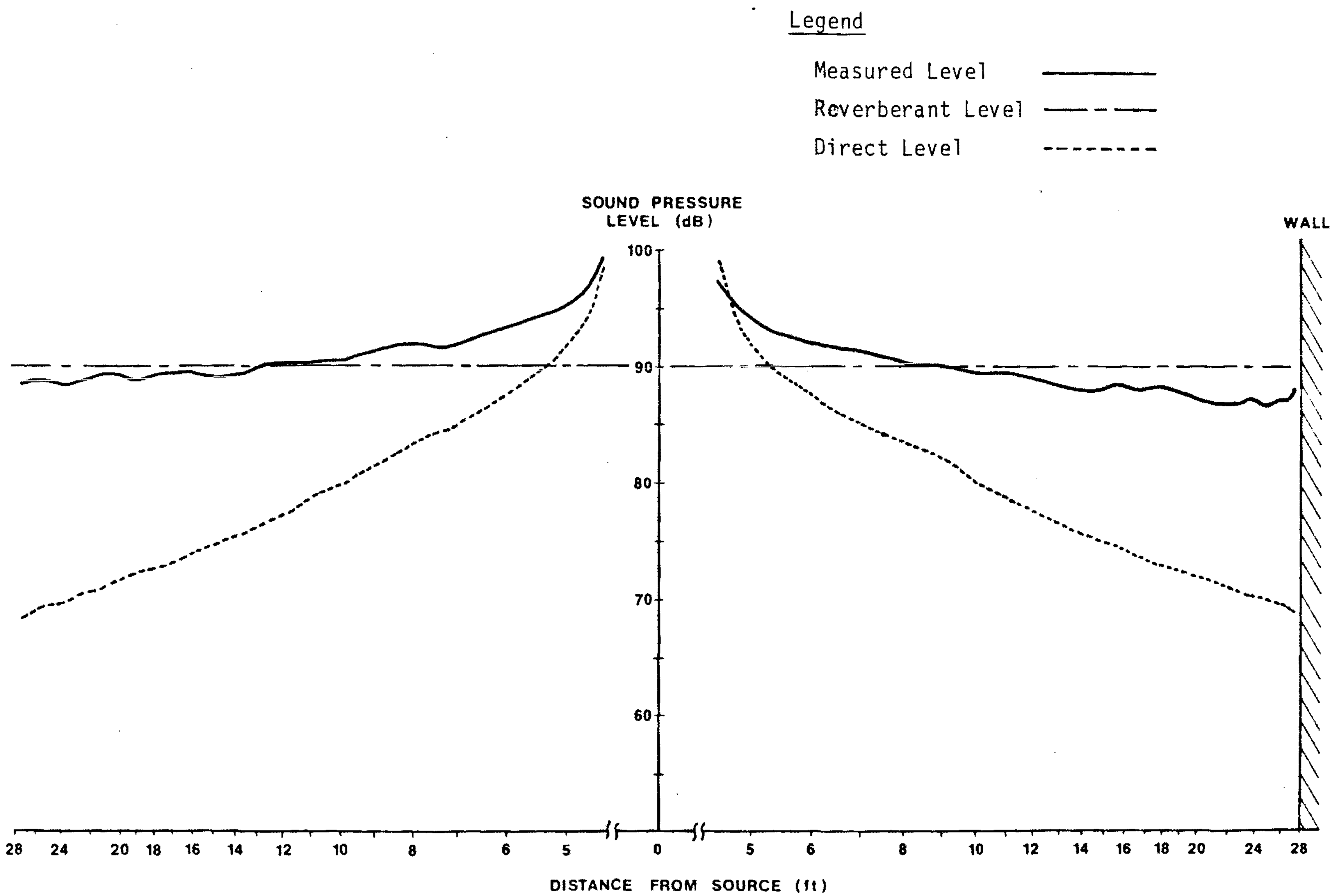


Figure 4
Direct/Reverberant Sound Pressure Levels
Using Known Source in Plant B

EQ(1) $$\alpha\ SAB = \frac{4}{S}\left[\frac{1}{\text{antilog}\ \frac{Lp-Lw}{10} - \frac{Q_\theta}{4\pi r}}\right]$$

Where:

α SAB = sabine absorption coefficient

S = surface area of the room (meters^2)

Lp = sound pressure level measured

Lw = sound power level output

Q_θ = directivity factor of the source

r = distance from the source to the measurement point (meters)

While it should be noted that reverberant field noise levels were not uniform in plant A and that the equation above is designed for use only when the reverberant field noise levels are uniform, the calculations were made for plant A using the space averaged reverberant field sound pressure level for the entire area. Table 1 provides a comparison of the calculated absorption coefficients for octave bands with center frequencies at 250, 500, 1000, 2000, and 4000 Hz and for the broad band test noise. The coefficient for the 4000 Hz band displays some atmospheric absorption due to the large volume of the plants.

TABLE 1

ESTIMATED SURFACE ABSORPTION COEFFICIENTS*

PLANT A		PLANT B	
Octave Band	α SAB	Octave Band	α SAB
250 Hz	.057	250 Hz	.032
500 Hz	.119	500 Hz	.089
1000 Hz	.072	1000 Hz	.053
2000 Hz	.078	2000 Hz	.077
4000 Hz	.228	4000 Hz	.187
Overall	.148	Overall	.104

* values include any contribution from atmospheric absorption as well.

Next the A-weighted noise contour data was used to estimate the A-weighted sound power output of the major sources in each plant. These calculations required an assumption that the sound pressure levels of each source were symmetrical in the vertical plane to those measured in the horizontal plane. They were based on that contour line closest to the apparent acoustical center of the source yet within 2 to 6 feet from that center. If the data indicated the contour level was unduly influenced by other interactions, then the sound power of that source was listed as not determinable. The results are tabulated in Table 2.

TABLE 2

ESTIMATED SOUND POWER OUTPUTS
OF MAJOR SOURCES

PLANT A		PLANT B	
Lung guns	108.2 dBA	Hock Cutter	100.2 dBA
Chillers	not determinable	Lung guns	102.7 dBA
Fan	94.7 dBA	Drying air	94.7 dBA
Hock Cutters	103.9 dBA	Exhaust fan	not determinable
Total	109.7 dBA	Gizzard peeler	not determinable
		Chillers	102.71 dBA
		Total	107.05 dBA

With the sound power output and surface absorption coefficient known, equation (1) was rearranged to calculate the sound pressure level of the reverberant field:

$$\text{EQ(2)} \qquad Lp = Lw + 10 \log \left(\frac{4}{S\alpha SAB}\right)$$

Using the overall absorption coefficient from Table 1 for each plant, the reverberant field sound pressure level attributable to the above estimated sound power inputs and room conditions were calculated and are as follows:

Plant A:	Lp = 91.4 dBA
Plant B:	Lp = 90.9 dBA

The values are reasonably close to the A-weighted sound pressure levels observed in the reverberant field in each plant (per figures 1 and 2):

Plant A: Lp being approximately 91.1 dBA (space averaged)

Plant B: Lp being between 90 and 91 dBA

CONCLUSION

While the above calculations provide only rough approximations, they do indicate that the major noise sources identified from the contour plot for each plant also appear to provide essentially all of the sound power for the reverberant noise field observed. Consequently, the solution to reducing or eliminating the noise problem in poultry processing plants is either to reduce the sound power output of the "major" sources (i.e. the lung guns, chillers, and hock cutters) and/or to treat the surfaces of the room to reduce reverberation.

Reducing the sound power of the major noise sources has been the subject of several studies. From a practical standpoint, redesign must be simple, inexpensive, and not substantially affect the manner in which processing and maintenance are currently done. If it does, the design will encounter difficulty in gaining acceptance. If the sound power output of the major sources can be reduced, there remains a question of whether other sources, currently being masked, will substantially complicate efforts to reduce overall sound power output.

Treating the surfaces of a plant with absorbing panels which are inpervious to water has also been the subject of studies. However difficulties have arisen both with cost and durability. If panels could be designed which were effective, inexpensive, and durable, clearly the extent of excessive noise exposure in poultry processing plants could be greatly reduced. It is here perhaps more than in sound power reduction that efforts should be directed for the benefits should be both long lasting and long reaching.

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2. Noise and Vibration Control, Leo Beranek, McGraw-Hill, 1971
3. A Study of Poultry Processing Plant Noise Characteristics and Potential Noise Control Techniques, J. Craig Wyvill, et al, Final Report for NASA Research Grant No. NSG 3228 and Georgia Department of Agriculture Research Project A-2028-006, 1980.

A STUDY OF POULTRY PROCESSING PLANT NOISE CONTROL TECHNIQUES

Final Report

for

Research Grant No. NSG 3228
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio

Prepared by

J. C. Wyvill
W. G. Morrison, Jr.

GEORGIA INSTITUTE OF TECHNOLOGY
Engineering Experiment Station
Atlanta, Georgia 30332

May 1981

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Industrial Acoustics Company
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Allforce Acoustics

RECOMMENDATIONS

A number of techniques can be used to reduce poultry processing plant noise. While the exact approach to solving a noise problem necessitates some understanding of the specific plant environment to be treated, in general covering the ceiling of a plant with a noise-absorbing medium is a practical first step. Once the reflected noise levels have been abated, then treatment of specific, identifiable noise sources can better take place. The logic behind this recommendation results from our earlier findings that much of the noise observed in a typical plant is caused by the poor acoustic qualities of the plant rather than the presence of numerous, loud noise sources.

In selecting a ceiling treatment, attention must be given to maintenance and replacement costs in addition to purchase and installation costs. Our study revealed a host of potential maintenance problems with noise panels if inadequate attention is given to the demands which will be placed on the covering medium during normal plant operations. Because the cover must remain intact to comply with USDA cleanability requirements, we recommend the use of rugged fiber-reinforced plastic covers to minimize the potential for failure. This results in a potentially higher panel cost but yields one which will last many years. We stress this because the cover is only a portion of the total cost which goes into panel construction, yet its failure renders the entire panel useless.

We also recommend that noise panels be suspended vertically from the ceiling. Our research showed a spacing of 3 feet between panels as a satisfactory spacing. However closer spacings can be chosen if the plant is attempting to bring about a larger amount of noise reduction or trying to improve low-frequency absorption. By our calculations, the 3-foot spacings should provide a 5 to 6 dB reduction in reverberant noise level at the plants presented in our previous report.

With regard to source quieting, a key word must be maintenance. Improperly maintained machinery was one of the leading causes noted for high machinery noise levels. We found that poorly maintained machinery can be located using a portable vibration meter. This provides a means of initiating preventive maintenance which can lower machinery noise levels.

In addition, we recommend isolating drive motors and pumps from large, expansive surface areas, such as those on a chiller. For example, flexible connecting tubes between pumps and chiller bodies could mean a substantial reduction in local sound pressure levels near the chiller. Drive motors also can be placed under hoods filled with absorbing medium to reduce their sound power emission to the plant. Pneumatic devices should always be muffled. A large number of companies design inexpensive mufflers for just such an application.

Lung guns, on the other hand, remain a problem where they must still operate. The most logical solution is to contain lung gun noise through the use of partial plastic barriers between individual operator stations. However, this demands that an absorbing medium be placed over the station to prevent sound pressure buildup to the operator.

Lastly, ice chutes can be insulated for both energy conservation and abatement of noise related to ice transport and discharge. There are a number of good vibration dampening mediums that are efficient thermal insulators as well.

SUMMARY

Industrial noise is a problem with untold potential consequences. High noise levels risk permanent hearing damage, low worker productivity, and poor employee/supervisor relations. It also may be a factor in worker turnover rates. In addition, Federal and State noise statutes require that certain maximum noise ceilings not be exceeded under threat of fine.

For the past two and one half years, the Georgia Tech Engineering Experiment Station has been studying noise generation and control as it relates specifically to poultry processing plants. This research was cosponsored by the National Aeronautics and Space Administration and the Georgia Department of Agriculture. The program took root through the efforts of the Georgia Poultry Federation.

After the first year and a half of investigatory studies, the Tech research team concluded that the poultry processing noise problem was caused by a few major noise sources allowed to permeate throughout the plant by reflecting off of the hard plant walls and ceilings. Subsequent research focused both on designing an absorptive medium to reduce reflections in a plant and on identifying ways to reduce noise at the source.

The absorbing medium found to be most cost effective for the poultry application is a panel composed of a fiberglass core and a tough, rugged, fiber reinforced cover which is impervious to water. By hanging a series of these panels throughout the ceiling of a plant, it is estimated that sound levels can be reduced by as much as 6 decibels in many plant areas. An important consideration in this research effort, however, has been the toughness of

the impervious cover. While a number of potentially acceptible plastic films are available, it was concluded that a reinforced design was critical if the panels were to withstand the abuse typical to normal handling. The study also revealed that using a hanging panel pattern versus placing the panels flat against the ceiling required one-third fewer panels to achieve similar absorption characteristics. This hanging orientation, it was also found, could be varied to allow increased absorption in the lower-frequency octaves.

With regard to source quieting, it was found that there is no substitute for a good maintenance program. Maintenance neglect was one of the most common reasons observed for loud source levels (be it worn bearings, lack of grease, etc). Preventive maintenance procedures utilizing portable vibration monitoring equipment is one method of reducing source levels. Nonetheless, there are machine designs which are either inherently noisy or which require frequent maintenance to keep them quiet. For these machines, isolation from the rest of the plant work area (where practical) and/or vibration isolation and dampening treatments are recommended.

Overall, it was concluded that isolating drive motors and pumps from chiller bodies is an effective method of reducing their noise outputs. Absorptive hoods also can provide relief from inherently loud drive motor arrangements, both on chillers and hock cutters. No practical modification was found for the hand-held lung gun. Installation of barriers between lung removal stations does appear to be a potentially effective containment measure. However, it was determined that the barriers could not serve to isolate operators and an absorptive hood should be positioned over the station to minimize sound buildup. Pneumatic mufflers on hand tools were found to be essential, and energy conservation provided excellent additional justification to insulate ice transport and discharge networks with sound dampening mediums.

PERSPECTIVE

This study was undertaken to identify practical solutions to the noise problem in poultry processing plants. To date, there has been only moderate activity in the area of abating poultry processing noise. Progress has been slow and many quieted designs have not proven satisfactory either in performing the functions they are required to perform or in withstanding the harsh working environment of the plant. Our research sought to remedy some of these problems.

The report is divided into two major parts:

1. Sound Absorption Investigations
2. Source Quieting Investigations

Each section provides a brief overview of selected activities known to be ongoing in that area along with a presentation of our research to find workable solutions to the problem.

There is no single remedy for the noise problem in poultry processing plants, rather a series of remedies, each having its own advantages and disadvantages. Unfortunately, the contribution of each remedy in reducing overall noise in a particular plant will depend on the noise sources in that plant and the plant layout. However, there is little doubt that the solutions described herein can contribute significantly toward reducing the general noise levels in most processing plants. More importantly, these solutions are durable and should not interfere with current operations.

FOCUS ON SOUND ABSORPTION

Introduction

Reverberation plays an important role in the noise problem associated with poultry processing plants.[1] Acoustical panels, to reduce reflected noise, have been used experimentally in plants but have encountered durability problems when the protective covers tear. Protective covers, it should be pointed out, are necessary to bring conventional absorbing materials (such as fiberglass or foam) into compliance with USDA cleanability requirements for use in a plant.

This section summarizes our efforts to determine the reason for current panel problems and to develop designs capable of enduring the types of abuse typical to the poultry processing environment while being effective in reducing plant noise.

Current Technology

We are aware of only a few companies today who are experimenting with absorbing panels specifically for use in poultry processing plants. The typical panel design is a fibrous or porous material covered with a plastic film which has been heat sealed and/or flame bonded (see Figure 1).

The absorbing media typically are fiberglass, mineral wool, or foam. The plastic covers, all of which are thin (between 0.5 mil and 2 mil in thickness), are either polyolefin or polyvinyl fluoride (PVF). This latter film (typically Tedlar®) has grown in popularity among panel builders.

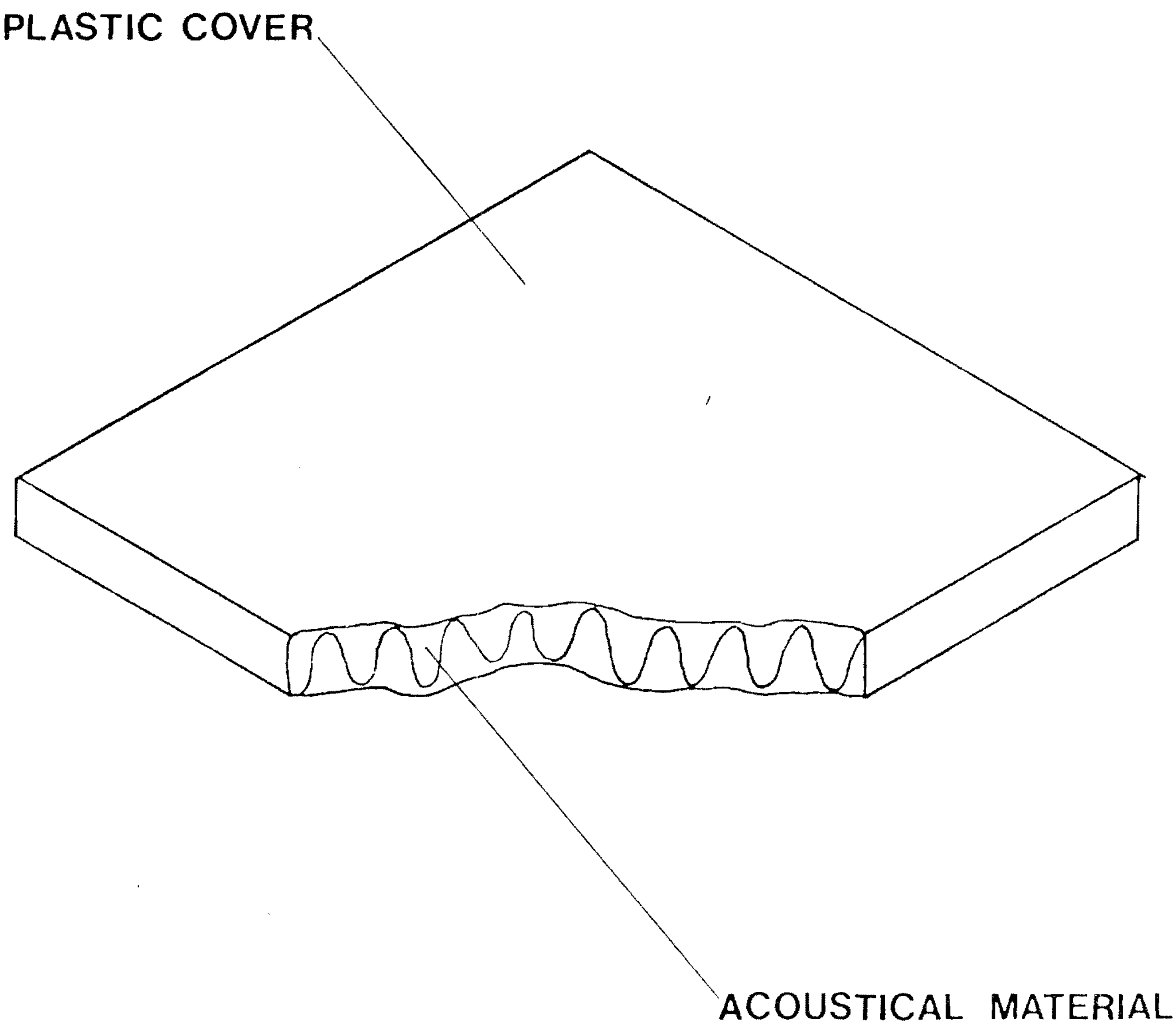


Figure 1 - Typical Design of Current Acoustical Panels

Because the cover of a panel is critical to its survival in the poultry processing environment, we devoted a great deal of attention to seeking materials and designs which lower the risk of cover failure from abusive handling. It was not our goal to design a panel which could not be destroyed through abuse, but rather to evaluate alternatives in cover ruggedness to determine which, if any, panel design might work given both USDA cleanability constraints and the very nature of the harsh cleaning and maintenance procedures typical to the poultry processing industry.

In evaluating the experimental panels currently being developed specifically for poultry processing plants, we discovered, through a series of informal tests, that one panel had a major weakness in its heat-sealed seam. When the panel was sprayed with high-pressure water during routine cleanup, the seam separated and the cover sheared off (see Figure 2). A similar panel design, by another firm, had a reversed seam which was capable of withstanding high-pressure water contact. We must add that tape has since been added to the seam of the first design, which does seem to offer the necessary reinforcement required.

Another common flaw we found in these experimental panels, however, was actually a characteristic of the PVF film itself. PVF film is strong, yet unusually susceptible to perforation (see Figure 3). This typically results in a total failure of the cover once perforated, due to the poor tear strength of the film.

These two findings served as a starting point for our research into a better panel design.

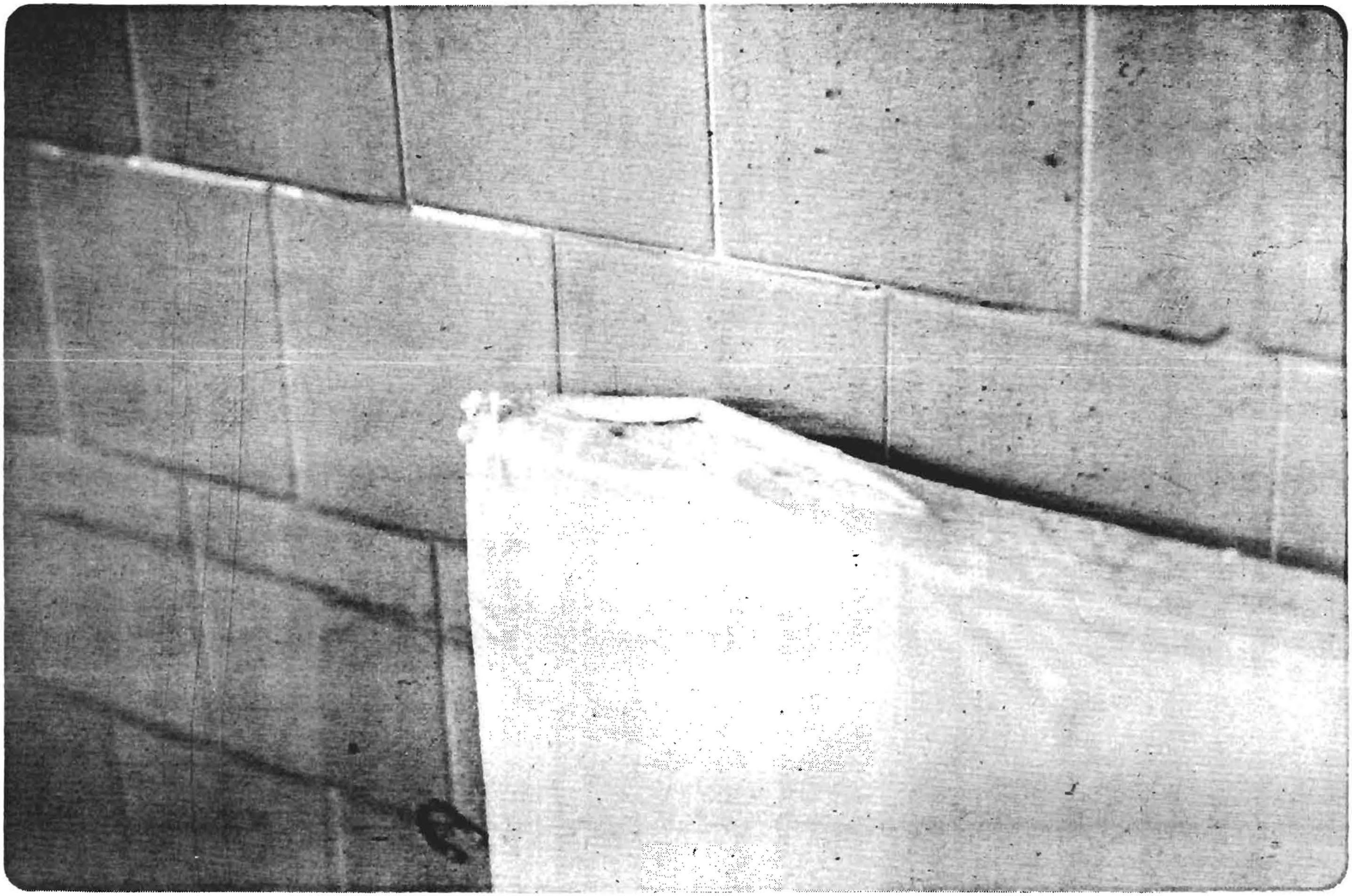


Figure 2 - Separated Seam After Washing

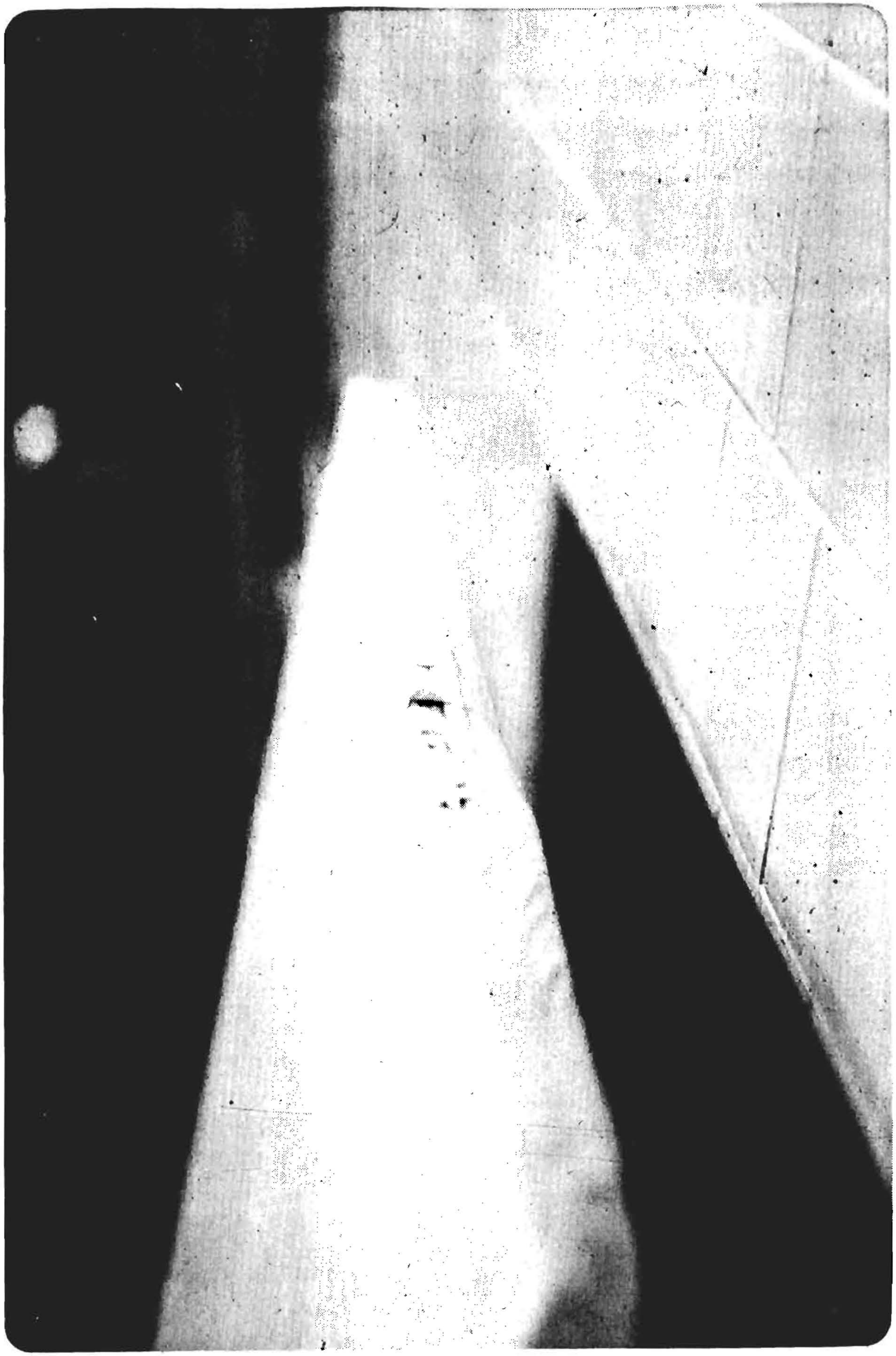


Figure 3 - Perforated Cover

Panel Design Research

In evaluating methods of providing a better panel, we looked at the strength of different covering materials, the absorption characteristics of the panel core, and the effect the cover had on the panel core absorption. Panel costs were also considered from the standpoint of overall cost minimization.

Covering Material Studies

There are a number of ways to improve the strength of a panel cover:

1. Use a stronger material.
2. Protect the cover with a shield.
3. Use thicker material.

We first focused our attention on stronger materials. Realizing that PVF film was the most commonly used covering material, we looked for materials with superior qualities to it. Table 1 presents the pertinent physical properties of several general film catagories. From this table we observed that polyester film offered superior tensile strength to PVF film while having comparable tear strength. Polyurethane film, on the other hand, offered superior general tearing strength to PVF film while having comparable shear strength.

In order to evaluate these properties in a commercial product, we acquired 1 mil samples of Du Pont Tedlar® (a PVF film), Du Pont Mylar® (a polyester film), and B. F. Goodrich Tuftane® (a polyurethane film). The tests conducted were:

1. Tensile Strength - This test (comparable to ASTM D882-79) provided a general measure of the overall strength of the film. It involved

Table 1

GENERAL FILM PROPERTIES

	Polyethylene	PVC	PVF	Polyester	Polyurethane
Tensile Strength (psi)	1,500-6,100	1,400-16,000	7,000-18,000	20,000-40,000	5,000-12,000
Tearing Strength					
Initial (lb/in)	65-575	110-490	997-1,400	1,000-3,000	350-600
Propagating (g/mil)	50-300	60-1,400	12-100	12-17	220-710
Resistance					
Grease & Oil	poor to good	good	good	good	good

Source: <u>Modern Plastics Encyclopedia, 1978-79</u>

taking 1-inch strips of each sample, placing them in the jaws of a gripping apparatus, and applying a pulling force until the sample failed.

2. Tear Strength - This test provided a measure of the strength of the film to shearing once a tear was initiated. It involved mechanically initiating a slit in a sample and then applying a pulling force to continue the tear through failure.

3. Burst Strength - This test measured the strength of the material to concentrated forces. It involved mounting the test sample on a small port on the side of a water-filled cylinder and gradually increasing the water pressure in the cylinder until the sample began to release water.

Table 2 presents the test results on the three samples.

Table 2

RESULTS OF FILM TESTS

	Tensile Test (lbs force)	Tear Test (lbs resistance)	Burst Test (psi)
1 mil Tedlar®	10.1	0.1	41
1 mil Mylar®	15.3	0.1	58
1 mil Tuftane®	2.6	0.8	43

From these tests, we observed that the urethane film we had selected did not exhibit tensile strength comparable to the PVF film, but that the other relationships were indeed similar to the general properties in Table 1. We concluded, therefore, that of the three products selected, Mylar® was the best covering medium candidate, from a strength standpoint, although we were disappointed in its tear strength.

We next looked at reinforced films to determine if suitable strength and tear characteristics could be found. The most commonly used technique for reinforcement is adhering a film to a thin cloth. Such a composite is used for vapor shields by the aerospace industry.[2] Initially we were unable to find such a product commercially available, so we constructed our own (calling it the EES composite). The materials we selected were Du Pont Mylar® film (1 mil) and dacron cloth (40 denier). We bonded the two together using a latex glue. No sooner had we fabricated this composite, however, than we found a sailcloth manufacturer who had an experimental composite product called Temperkote® which was made of Du Pont Mylar® and rip stop nylon. Their product was similar to our test sample, yet offered the additional advantage of being commercially available.

We began an immediate evaluation of both samples. Utilizing the test procedures referenced above, we generated information on the strength characteristics of each sample. In order to maintain a comparison basis for these dissimilar test samples, we weighed each one. The reinforced films, we found, had a weight per square foot similar to that for a 2 mil unreinforced film. To provide a comparative link for our strength studies, we therefore tested a 2 mil sample of ICI Americas Melenax® (a polyester film), since at this point we felt polyester film represented the best unreinforced covering material. Table 3 presents the test results.

Table 3

COMPARISON OF REINFORCED AND UNREINFORCED MATERIALS

	Tensile Test (lbs force)	Tear Test (lbs resistance)	Burst Test (psi)
2 mil Melinex®	36.7	0.1	102
EES Composite	35.8	1.6	120
1.9 oz Temperkote®*	38.3	5.8	125

*Approximate weight per square yard of material.

From these data we concluded that both of the composites had tensile strength similar to the 2 mil polyester film but significant tear strength advantages. We were satisfied that this represented a significant advantage and therefore were inclined to list the composite as the better cover medium from a strength standpoint.

We also looked at one additional method of reinforcing films, namely, scrimming. Scrimming is a term used to define the bonding of a netting material either to the back of a film or between two films. Most of the scrims we studied were experimental in nature, usually employing nylon or fiberglass netting and a PVF film. While the concept offered tear strength advantages over unscrimmed films, we found that the film in unsupported areas of the netting remained vulnerable to failure and hence offered no significant advantage over the composite.

Before leaving the subject of material properties, we conducted one additional test on the six test samples studied. As mentioned earlier, PVF film exhibits a low resistance to perforation if scrapped, and any contact with the film can result in a scratch or slight perforation. Once perforated, as our own tests proved, the film has little resistance to tear propagation. We therefore conducted an abrasion test on each of the six samples in Tables 1 and 2 to determine their relative strength. The test involved rotating a sample 1,000 times against a rough surface. Our tests resulted in none of the samples, other than Tedlar®, failing. The Tedlar® sample failed after only about 10 revolutions. Therefore, we were satisfied that all of the films being studied offered a significant advantage over PVF film in this regard.

We next focused on methods of protecting the cover with a shield. We investigated two methods of providing this protection:

1. a screen
2. a perforated plate

Neither of these methods was particularly attractive because of the problems associated with cleaning them.

The screen we selected was made of polypropylene, which is both non-corrosive and inexpensive. We selected a 6 x 8 strand per inch pattern for evaluation. Conferring with the United States Department of Agriculture (USDA) field office in Atlanta, we gained approval to test a panel with a PVF film cover and a screen on the outside in a plant. After nearly six months of exposure, the sample remained relatively clean. However, we must point out that had the panel come in contact with blood, feathers, etc., cleaning would have been difficult without removing the screen.

We also evaluated a perforated plate design. Unfortunately, the only commercial design we could acquire was a box design made of steel. The difficulties we encountered in sealing the box so as to eliminate food entrapment forced us to consider putting the film on the outside of the box, thereby defeating the box's protective qualities. Since the perforated plate designs we reviewed were also expensive, we chose to eliminate it from further consideration.

We lastly focused on utilizing a thicker film material to increase cover strength. In order to learn if there was a minimum satisfactory thickness for use in the poultry environment, we decided to experiment with different panel covers in an actual poultry application. We constructed panels with covers of four different thicknesses (1, 2, 3, and 5 mil). Based on our earlier work, we also decided to evaluate simultaneously the different materials studied: Tedlar®, Mylar®, Melineax®, Tuftane®, the EES cloth/film composite and, as mentioned above, a polypropylene screen covered PVF film. The four thicknesses were evaluated on polyester film only.

The samples (eight in all) were made using a fiberglass core material, and the panels were hung at Tip Top Poultry in Marietta, Georgia. The mounting arrangement placed the panels low enough to the floor (approximately 10 feet) to insure their being washed daily (see Figures 4 and 5). The test lasted six months. Test results are presented in Table 4.

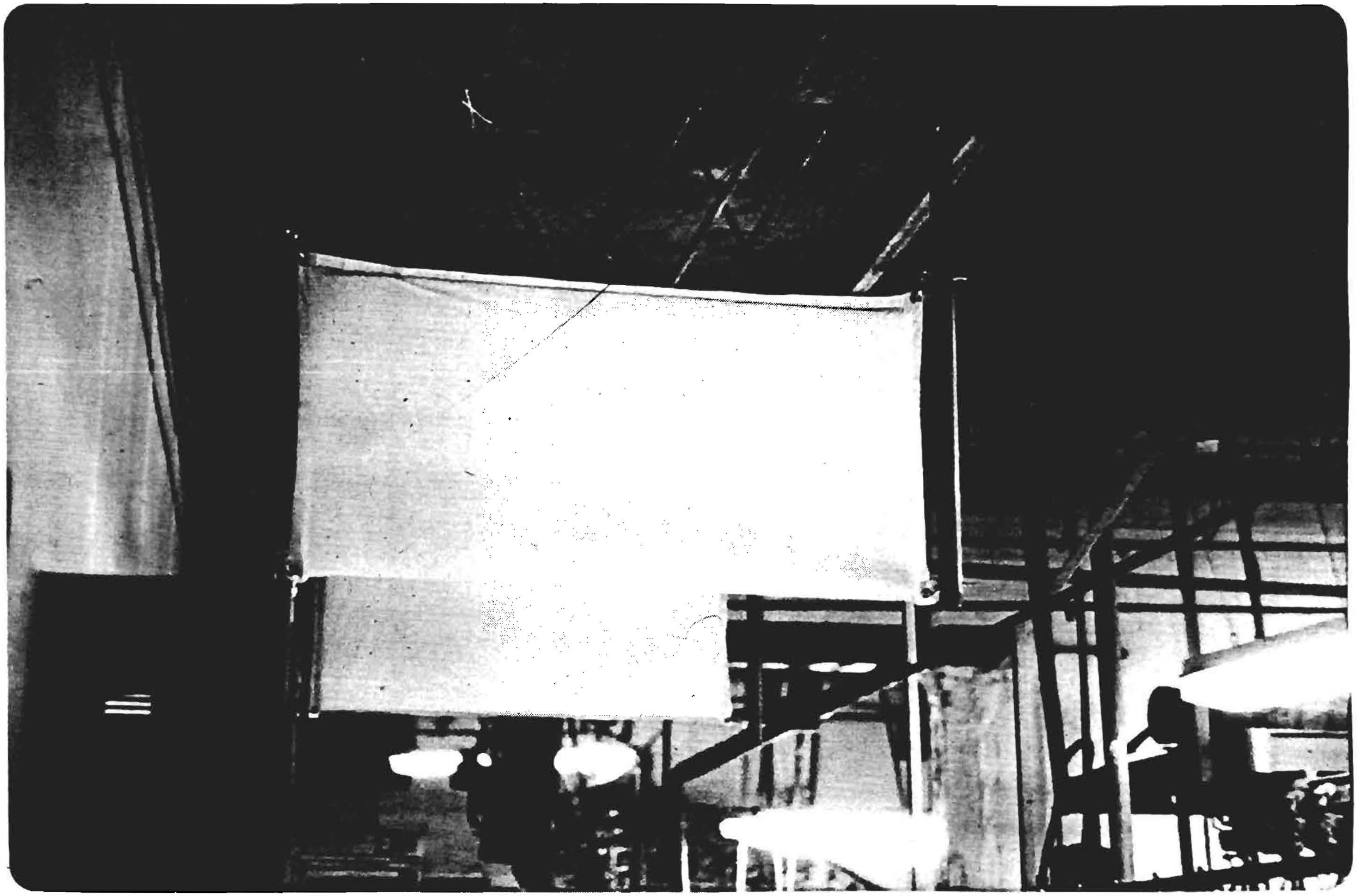


Figure 4 - Test Panels in Plant

Figure 5 - Test Panels Being Sprayed in Plant

Table 4

SIX MONTH PANEL ENDURANCE TEST

Panel Cover Material	Test Result	Explanation
1 mil Tedlar®	failed (after 4 mo.)	corner pulled away
1 mil Tuftane®	failed (after 1 day)	corner pulled away
1 mil Mylar®	failed (after 1 mo.)	corner pulled away
2 mil Melinex®	survived test	
3 mil Melinex®	survived test	
5 mil Melinex®	survived test	
Polypropylene screen over 1 mil Tedlar®	failed (after 5 mo.)	a rip in the film developed on the panel's bottom edge
EES film/dacron composite	survived test	

From this test, we concluded that a 1 mil cover is simply too thin for long-term exposure in a poultry processing plant. We made this conclusion realizing that the conditions of abuse were highly variable between panels and hence did not provide precise failure information regarding the four panels that did fail. We therefore ruled out attempting to draw conclusions regarding individual material-to-material strength characteristics for the various 1 mil panel covers.

Based on the investigations described above, we concluded that a fiber reinforced cover (such as Temperkote®) was the best all-around covering material of those studied. We sought, at this point, to answer two additional questions regarding the Temperkote® material in particular:

1. Could it endure continuous exposure to industrial lighting?
2. Was it a fire hazard?

The first question evolved from documented evidence that neither untreated Mylar® nor nylon can resist degrading if exposed for prolong periods to sunlight. However, when the light spectra of typical industrial fluorescent lighting are viewed, there is a noticable absence of ultraviolet energy (see Figure 6), which is absorbed by both Mylar® and nylon (see Figure 7) and eventually leads to their degradation.[3] While considerable debate was found regarding the possible life of the composite under continuous lighting exposure, nearly every expert contacted agreed the product should last several years at worst.

In order to reinforce these opinions, we conducted a light exposure test. Placing a sample of the Temperkote® around a 25-watt fluorescent light fixture, we subjected it to over 4,100 hours of continuous light exposure. Due to the direct placement of the film on the bulb, the light energy concentration was several orders of magnitude greater than that typical for panels in an actual plant situation. This high-energy-intensity exposure was expected to accelerate any degradation that might occur, thereby compensating for the short time duration of the test. The test ended with no noticeable change in material property strength. We concluded, therefore, that the material was suitable for long-term exposure to industrial lighting without displaying significant degradation.

The second question came from a concern for placing large quantities of this material in a plant without having any information on its burning characteristics. Tests were conducted on the composite film using the ASTM E 84-80 test for developing surface burning characteristics of building material. This test is recognized nationally as a means of classifying materials for use in industrial building applications. Appendix A provides more specific information on the test. The test results are shown in Table 5.

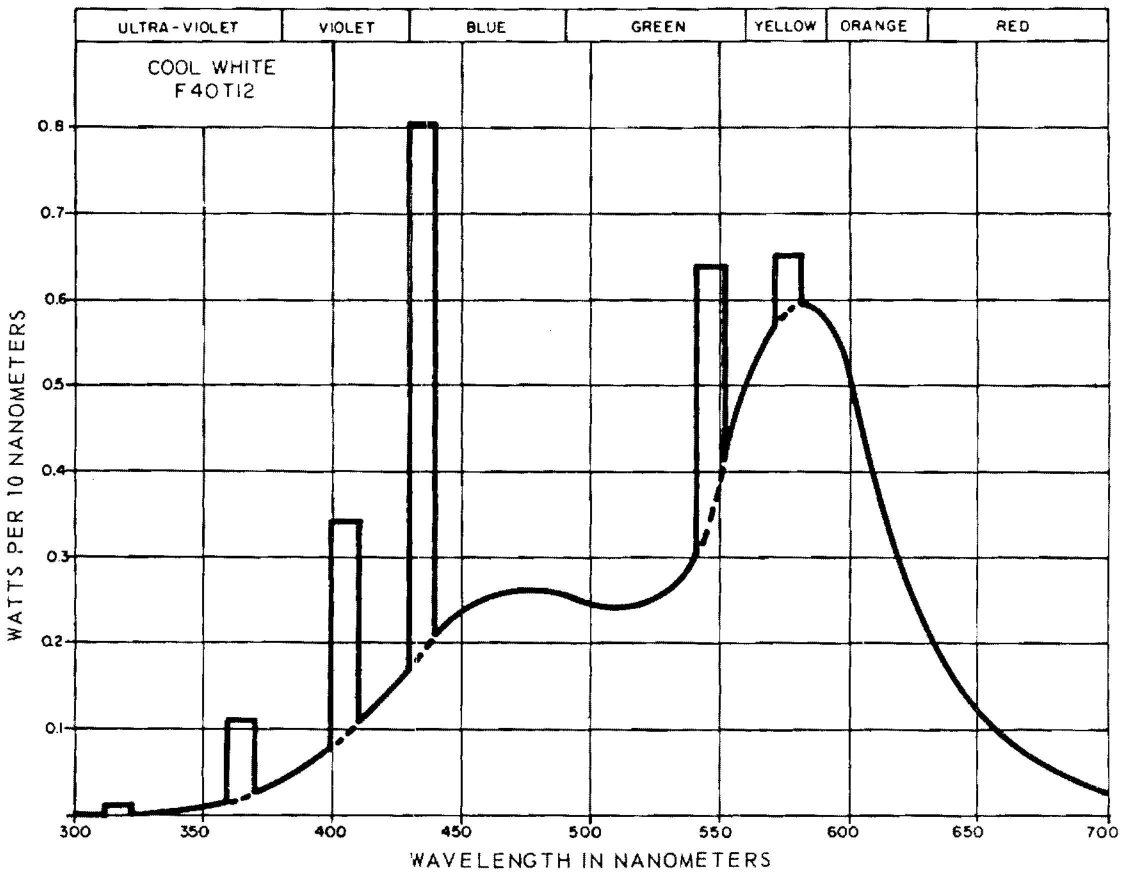


Figure 6 - Energy Spectrum for a 40-watt Fluorescent Light Fixture

Source: GTE Sylvania Engineering Bulletin 0-283

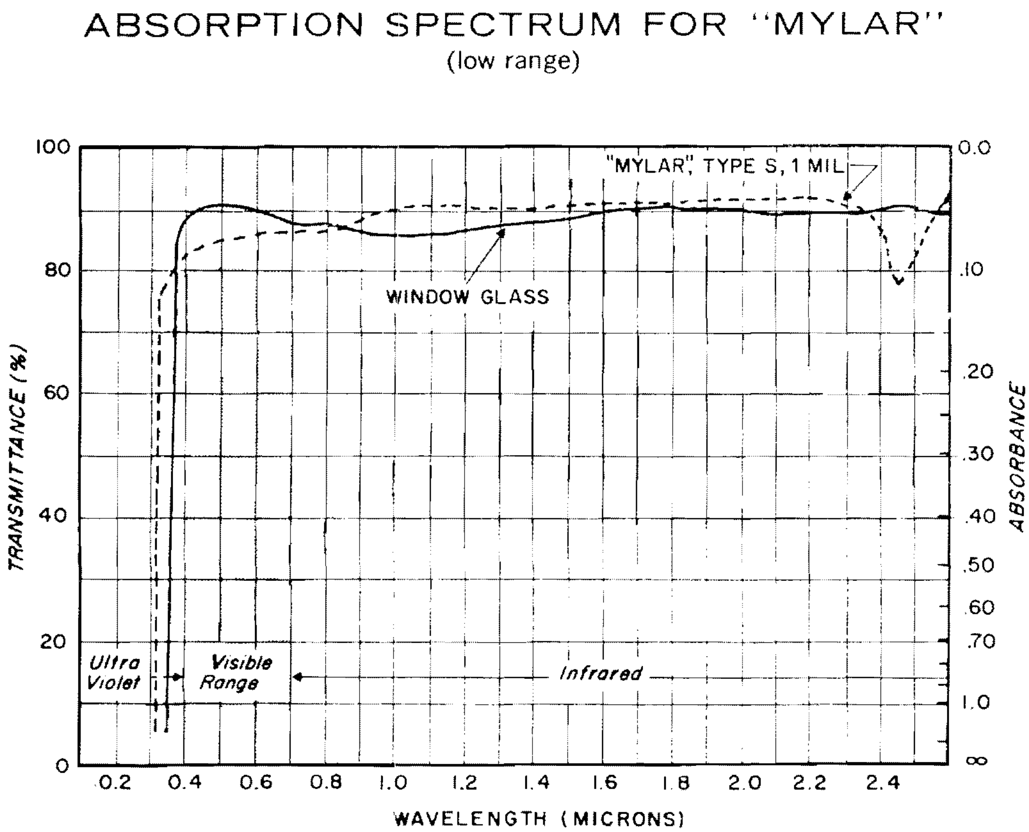


Figure 7 - Absorption Spectrum for Mylar

Source: <u>DuPont Bulletin M-5C</u>

Table 5

FIRE CHARACTERISTICS OF 1.9 OZ TEMPERKOTE®

Flame Spread Index	10
Smoke Development Index	25

These results indicate that the composite does not offer any unusual fire risk. In fact, the test rating for the product is class A (the best obtainable). Nonetheless, these tests do not check the material in an orientation similar to that in which it will be used. Consequently, we still feel that caution should be exhibited in drawing conclusions about the flammability of the film in use. What this test does do is compare the film to other materials commonly used in the construction of industrial plants, and does so under controlled, consistent test conditions.

Just as there were differences in the strength of the cover materials studied, so also there were differences in the price of each. Table 6 presents prices obtained for three of the six samples studied.

Table 6

QUOTED PRICES FOR SELECTED COVER Materials (1980 prices)

1 mil Tedlar®	7¢/sq. ft.
1 mil Mylar®	3¢/sq. ft.
1.9 oz Temperkote®	37¢/sq. ft.

Panel Core Consideration

In selecting a core material to use in a panel, we evaluated three candidates:

1. fiberglass
2. urethane foam
3. felt

Our criteria for selection were cost, absorption characteristics, and acceptability by USDA for use, realizing that if and when a cover tears, the core is at least momentarily exposed.

Focusing first on cost, we compared three products currently available on the market:

1. Owens-Corning semi-rigid fiberglass
2. Allforce polyurethane foam
3. Scott polyurethane felt

Other companies were checked to assure that the price figures for these products were representative. The cost figures appear in Table 7.

Table 7

QUOTED PRICES FOR ABSORBING CORE MATERIAL (1980 PRICES)

Owens Corning Fiberglas® 703 semi-rigid board (2" thick - 2' x 4')	$0.56/sq.ft.
Korfund Noiseguard (foam absorber F-500) (2" thick roll 54' wide x 50' long)	$1.45/sq. ft.
Scott Industrial Foam Scottfelt (2½-900) (2" thick - 2' x 4')	$9.00/sq. ft.

Next, we reviewed the absorption characteristics of these three products

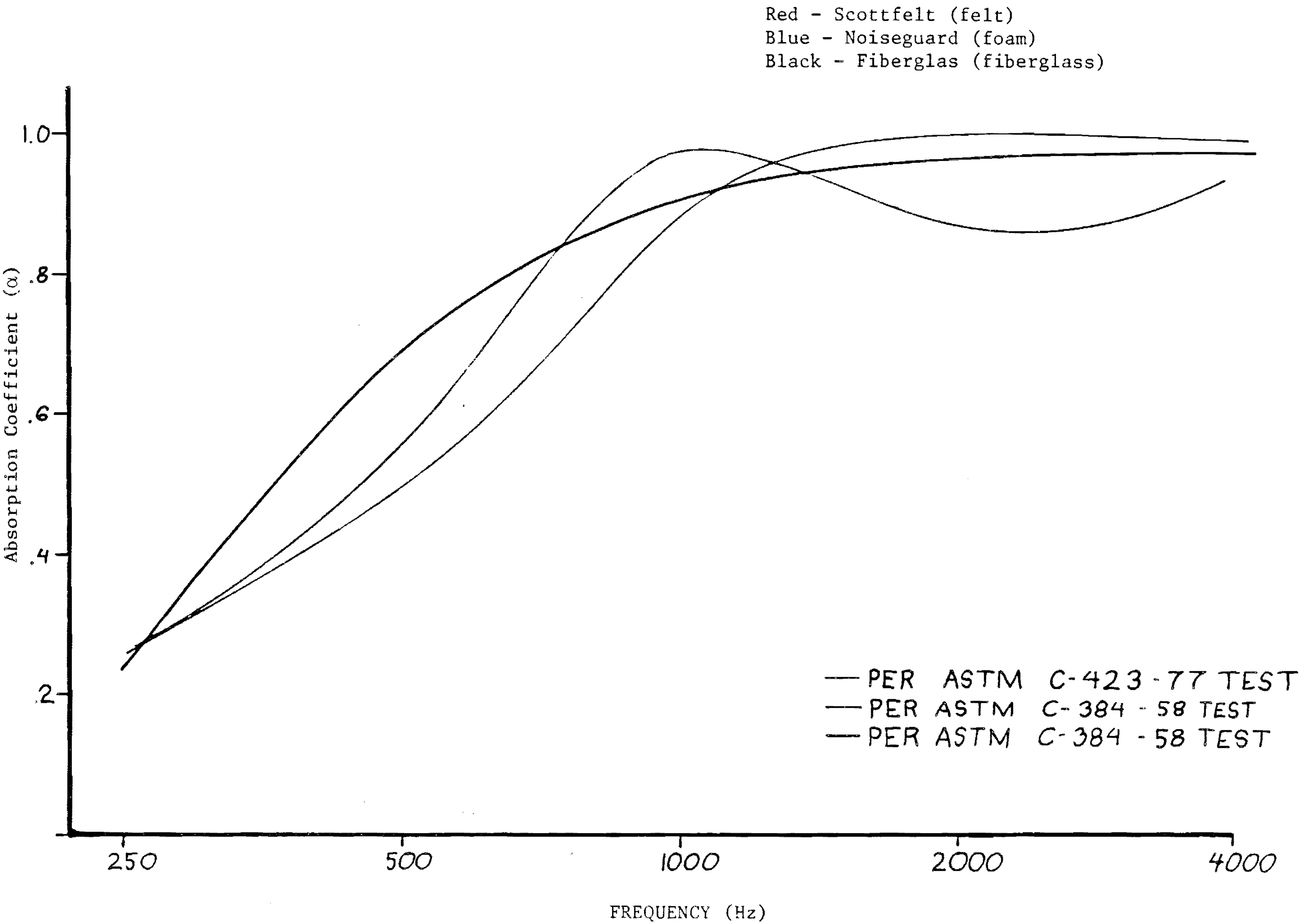


Figure 8 - Absorption Characteristics of Potential Panel Cores (1" Thickness)

based on published data. In order to utilize published data and still make an equal comparison, we had to look at a 1" thick product. The data are shown in Figure 8. It must be pointed out that this information was developed by the manufacturer. Two of the manufacturers used the impedance tube method, which measures normal incidence performance. They then corrected the data for random incidence performance. The third manufacturer used the reverberant room test method, which measures random incidence performance.

Because of this difference in test procedure, a precise comparison of product performance is difficult. What the data do show, however, is that none of the products offer a clear absorptive advantage over the others. Granted, the specific data do show octave band absorption differences, but given the differences in procedure that were used, it is doubtful these could be called significant.

Lastly, we reviewed the acceptability of these products by USDA. The Washington office of the USDA told us that none of the generic products contained toxic components and therefore could be used in a poultry application. Discussions with local inspectors confirmed this position stressing however that any panel with a ripped cover would have to be removed immediately.

Based on the three aspects mentioned above, we selected fiberglass as the most cost-effective absorbing medium of the three evaluated.

The Impact of Placing a Cover Over the Absorbent Core in Terms of Acoustical Performance

We conducted a series of tests designed to determine the acoustic effect of covering a sound-absorbing panel with various protective coverings. The

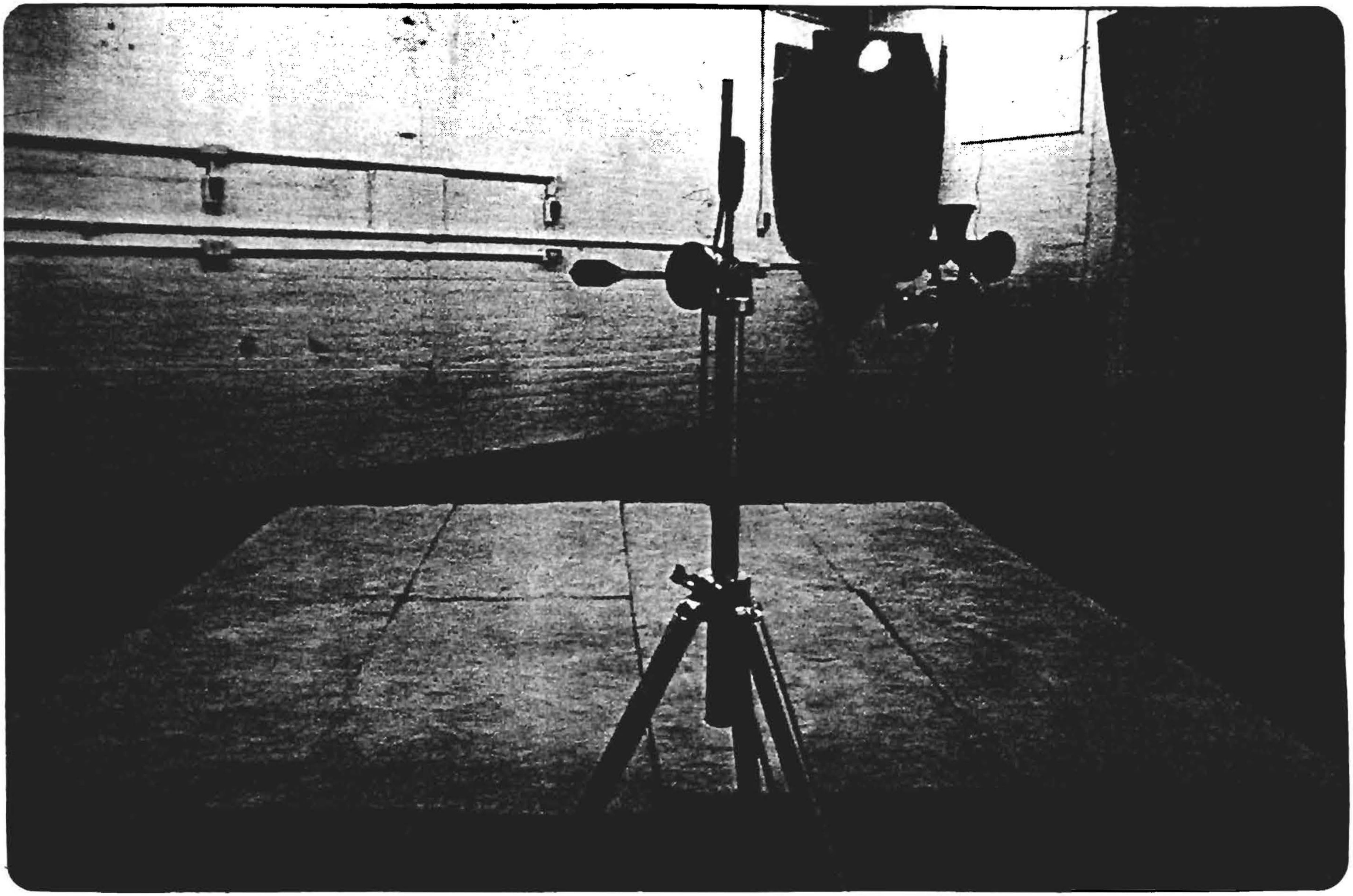


Figure 9 - Uncovered Panels Being Tested in Reverberant Chamber

test procedure involved using eight 2' x 4' x 2" series 703 semi-rigid Fiberglas® insulation boards from Owens-Corning. The panels were placed on the floor of a reverberant test chamber (see Figure 9). The sound absorption coefficients for the uncovered panels were determined, and a comparison was made of the change in the sound absorption coefficient when the panels were covered with different films.

The test methodology utilized in evaluating the effect of protective covers on the acoustical absorption of a fiberglass core was the reverberant decay method (ASTM C423-77). Prior to conducting the test, we modified a building on the Georgia Tech campus to serve as our test chamber. (See Appendix B, which discusses our qualification tests.)

To determine the absorption coefficient of the test panels, a loudspeaker system was fed by a B&K 4205 octave wide noise source to develop a steady-state diffuse field in the chamber. Reverberant levels reached were 110 dB for the 250 and 4000 Hz octave and 125 dB for the 500, 1000 and 2000 Hz octaves. The noise signal was left on for about two seconds to insure that a diffuse steady-state field had been set up in the chamber. The signal was then abruptly cut off, and the decay of the sound field picked up by a microphone and recorded onto magnetic tape. The analysis of the decay rate was performed by playing the tape-recorded signal back through an additional octave filter and into a true RMS detector. A Hewlett-Packard 5420-A spectrum analyzer, in the time record mode, displayed the decay as sound pressure level versus time. This display was transferred to paper by a chart reader. A best-fit line was then drawn through the decay portion of the curve starting 5 dB down from the beginning of the decay and extending to a point that was 15 dB above the noise floor of the measurement. This usually resulted in 25 dB to 30 dB of range, which was then extrapolated to the time required for 60 dB of decay to occur. Six decays at each octave band were charted and an average decay time calculated. The standard deviation of the measurement samples was also calculated to indicate the measurement uncertainty.

The absorption of the test chamber empty and with panels was calculated using the following formula:[4]

$$A = \frac{.9210 \times V \times 60}{C} \times \frac{1}{T}$$

where
- V = volume of the chamber (ft^3)
- C = speed of sound (ft/sec)
- T = average decay time for a 60 dB drop in sound pressure level in the room (sec)

The absorption of the test specimen alone (A_T) was then calcualted using the following formula:[5]

$$A_T = A_1 - A_2$$

where
- A_1 = The absorption of the test room with panels
- A_2 = The absorption of the test room empty

The absorption coefficient (α_T) of the test specimen was next calculated using the following formula:[6]

$$\alpha_T = (A_T)S_T + \alpha_1$$

where
- A_T = The absorption of the test sample (Sabines)
- S_T = Surface area of the test specimen (ft^2)
- $\alpha 1$ = Absorption coefficient of the floor area covered by the test specimen

The absorption coefficients obtained on the uncovered fiberglass panels are shown in Table 8.

Table 8

ABSORPTION COEFFICIENT FOR UNCOVERED FIBERGLASS PANELS (2" THICK, SERIES 703)

Octave (Hz)	250	500	1000	2000	4000
α	.661	.798	.824	.766	.605

We hasten to note that our values are significantly lower than those reported by the manufacturer, but a difference in measured values is not uncommon between two different reverberant rooms. Our goal in reporting these absolute values is to allow subsequent evaluations. We make no challenge of the manufacturer's reported values.

Listed in Table 9 are the percentage changes in absorption coefficient observed for the fiberglass panels using different covering materials.

Table 9

PERCENTAGE CHANGE IN PANEL ABSORPTION COEFFICIENT CAUSED BY COVERING THE PANEL

	Octave				
Material	250	500	1000	2000	4000
Mylar® (1 mil)	13.5	0.88	0.0	-2.87	-10.2
Mylar® w/screen (1 mil)	-8.17	4.14	-2.67	-9.01	-11.2
Melinex® (2 mil)	-2.12	2.25	-9.71	-18.4	-42.5
EES composite	8.58	0.80	-9.91	-23.9	-44.7
Temperkote® (1.9 oz)	17.2	2.8	-4.38	-9.68	-38.8
Melinex® (3 mil)	3.03	4.64	-11.77	-29.5	-57.0
Melinex® (5 mil)	17.2	-2.13	-29.4	-43.7	-71.9

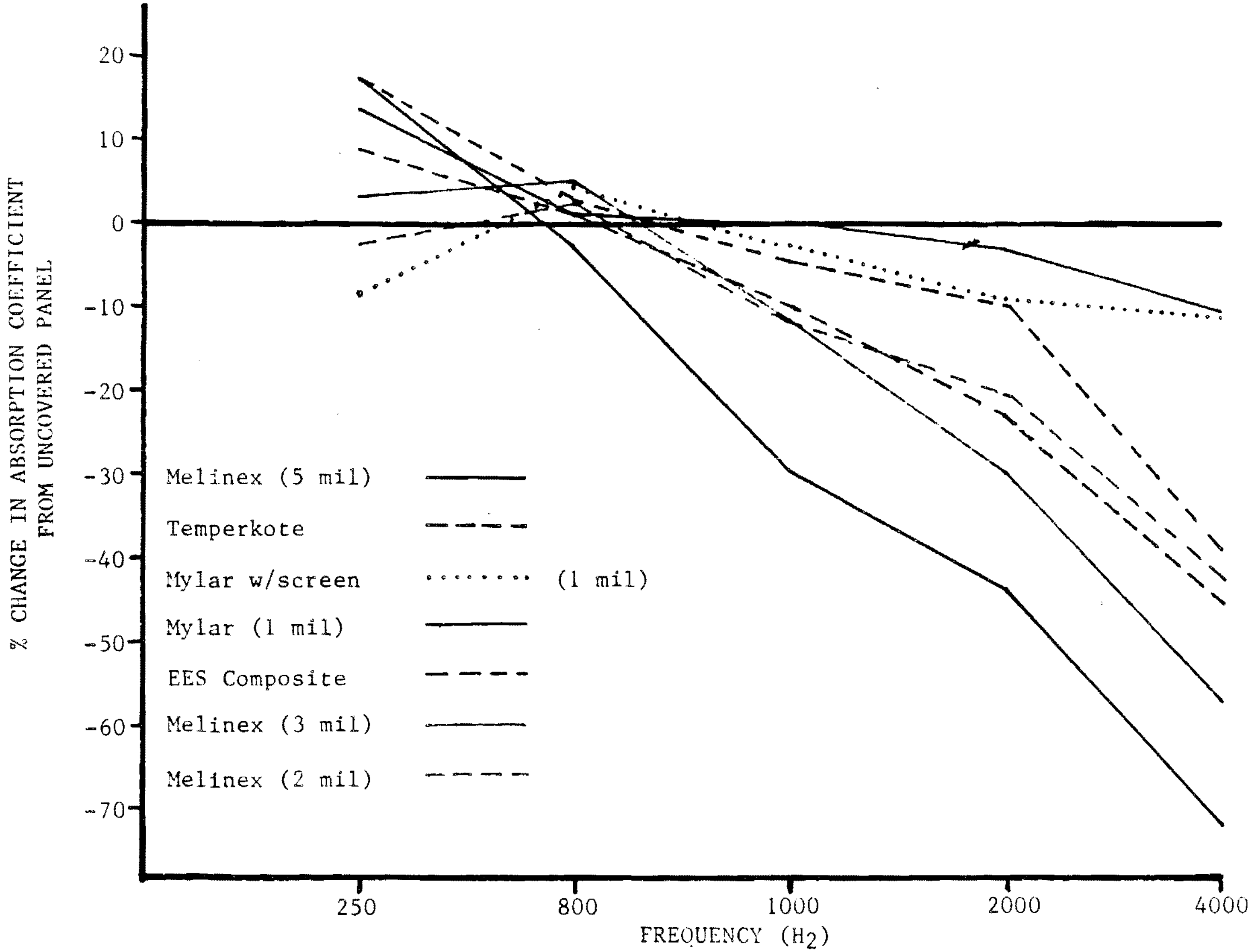


Figure 10 - Percentage Change in Panel Absorption Coefficient Caused by Covering the Panel

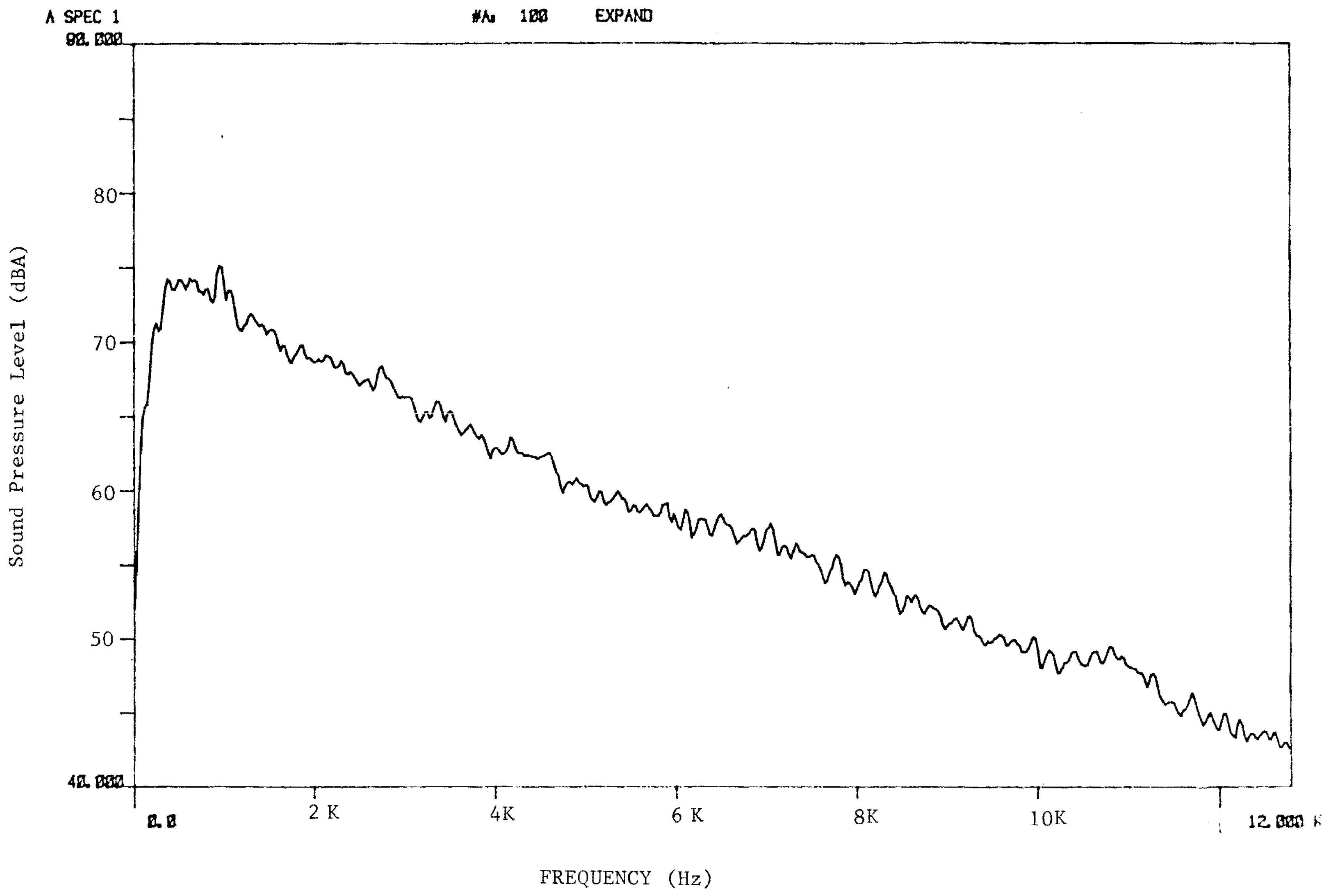


Figure 11 - Typical A-Weighted Spectrum for Reverberant Noise Field in a Poultry Processing Plant

These values are graphically plotted in Figure 10. They exhibit an interesting phenomenon. At lower frequencies, a thicker cover enhances panel absorption, while at higher frequencies a thicker cover diminishes panel absorption. This phenomenon is explained when the covering material is viewed as a driven oscillator transmitting sound energy to the core. The amplitude of oscillation is controlled by stiffness, resistance and mass. The degree of stiffness enhances the amplitude of oscillations at lower frequencies because of the inverse relationship of frequency and mechanical impedance. This overrides the mass damping impact. At higher frequencies, this is not true, and therefore, mass diminishes the amplitude of the oscillation.[8]

Because these panels must absorb noise specific to the reverberant sound field of a poultry processing plant, we reviewed the sound pressure frequency spectrum typical for such plants. Figure 11 presents a typical A-weighted spectrum. From this figure, it is obvious that the panels must be optimally effective between 250 and 2500 Hz. From Table 9 and Figure 9 it appears that a cover of greater thickness than 3 mils substantially diminishes absorption for all frequencies over 1000 Hz. Below 3 mils, however, it does appear as though the panels retain suitable absorption to allow sensible tradeoffs for strength. Based on our earlier strength research, we continue to remain convinced that a fiber reinforced cover, such as Temperkote®, is the best overall covering medium.

Other Panel Considerations

In addition to cover strength, we became concerned with the strength of the panel seam as well. As mentioned in our discussion of current technology, the seam can prove to be the weak link in a panel design. This became particularly obvious to us during our endurance test at Tip Top Poultry. The panels installed in that plant were taped with two-sided Mylar® tape (3M brand #415). After one week of exposure in the plant, we observed that the tape was beginning to separate from the inside of the seam (interior of the panel) out. After the full six months of exposure, none of the seams had failed, but the early separation forced us to consider different seaming techniques. The three techniques considered were:

1. two sided tape
2. heat seal
3. stitching

We made up 1" wide samples of 1.9-oz Temperkote® using each seaming technique. Two taped seams were studied, one employing the Mylar® double-stick tape already mentioned and the other a special high-adhesive double-sided tape (3M brand ISOTAC #Y-9460). The heat seal was a ¼" wide seal produced by a Vertrod® heat sealer. The stitching employed polyester thread. Using the tensile test described earlier, we applied a pulling force evenly distributed along the entire seam and measured the failure point of the seam. Table 10 presents the test results.

Table 10

SEAM STRENGTH TESTS USING
1.9-oz TEMPERKOTE®

	Failure Load
Mylar® tape	1.77 lbs
ISOTAC tape	1.38 lbs
Vertrod® heat seal	6.60 lbs
Stitch	4.19 lbs

It should be added that the heat seal exhibited some brittleness near the edges of the seam. Also the thread actually failed in the test, thereby exhibiting the strength characteristics of the thread used in the stitching.

While we make no recommendations on seaming technique, we feel it is important to note the effect different techniques can have on cover strength. Obviously, when the maximum load a seam will be asked to take is unknowing seam selection should tend towards the highest strength obtainable, limited only by the strength ceiling of the material being bonded.

Panel Spacing Studies

When placing panels in the ceiling of a plant, a number of possible mounting techniques may be used. Perhaps the most straightforward is to mount the panels flat against the ceiling. However, if the panels are hung vertically from the ceiling, more surface area is exposed, thus increasing the total sound absorbing potential of a single panel.

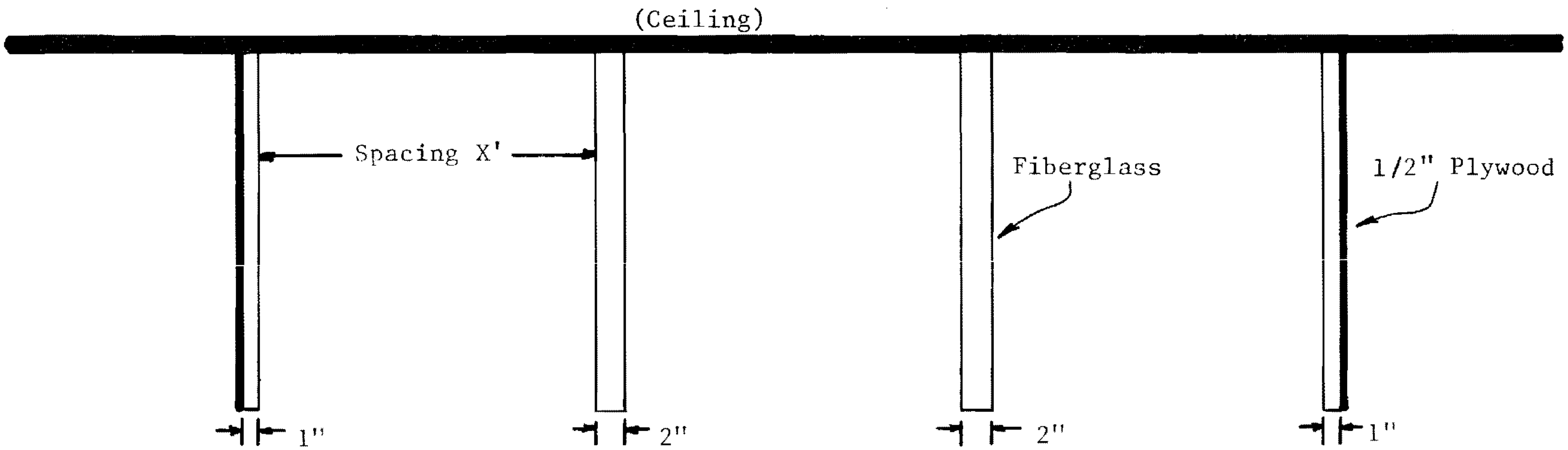
There have been several studies on hanging configurations of panels.[9] Our tests were designed to measure the relationship between spacing and absorption when covering is added to a panel. It was hoped that by varying the distance of the spacing between hanging panels, improved low frequency absorption could be obtained.

The tests were conducted in the reverberant test chamber described in Appendix B. The test procedure involved hanging 2" thick, series 703, 2' x 4' Fiberglas® panels and two 1" thick, series 703, 2' x 4' Fiberglas® panels from the ceiling in the center of the test chamber. The panels were positioned face to face and were spaced at equal intervals. The 1" thick panels were placed on the ends of the spaced arrangement and ½" thick plywood sheets were placed against the outside face of each 1" panel (see Figure 12). This prevented the outside face of these exterior panels from distorting the measured differences in total absorption resulting from spacing variations. The 1" thickness of the outside panels when placed against the plywood gave them an effective thickness of 2" on the interior, complementing the other 2" interior panels.

The panels were tested at 6", 1', 2', 3', and 4' spacings. Because of limitations in room geometry, one of the three interior panels had to be removed when going from a 3' to 4' spacing to keep the length of the arrangement at 12'.

The tests were conducted on both uncovered Fiberglas® panels and on panels covered with 3 mil Melinex®. The use of 3 mil Melinex® as the cover material was based on our earlier assessment that it represented the maximum cover thickness that could be used without significantly impairing panel acoustic performance in the 250 to 2500 Hz bandwidth.

The test results were evaluated from two angles. First, the total change in absorption (sabins) noted during the tests was divided by the area of ceiling displaced by the hanging panels. This evaluation provided a measure of the improvement in the reflecting surface absorption when it was covered with the hanging panel arrangement. These values are presented in Table 11. Next, the total change in absorption (sabins) was divided by the number of panels used in the hanging pattern. This evaluation provided a measure of the effectiveness of each panel in contributing to the total absorption observed. These values are presented in Figures 13 and 14.



NOTE: Spacing Values X = 1/2, 1, 2, 3, 4

Figure 12 - Schematic of Panel Spacing Test

The test results from Table 11 demonstrate that the hanging arrangement had absorption characteristics similar to those of panels lying flat on the floor when spaced as shown in Figure 15. Since a 2' spacing arrangement utilizes the same amount of material per square foot of area covered as panels placed flat on the floor, less total absorbing medium was needed using the hanging arrangement to achieve results similar to placing panels flat against the ceiling.

The test results observed from Figures 13 and 14 demonstrate that as the spacing between panels is increased, the unit absorption per panel increases. The values in Figure 13 for the uncovered panels resemble results reported by Owens-Corning.[10] As they noted, there is an optimal panel spacing beyond which no additional increase in unit panel performance will be achieved. This spacing, however, is frequency dependent. Figure 14 shows that the relationship between panel spacing and unit panel absorption is changed by the addition of a cover, particularly in the lower frequency octaves, where the increase in unit panel absorption bears an exponentially increasing rather than decreasing relationship with larger spacing. This suggests a possible shift in the optimal panel spacing point if covering materials are used.

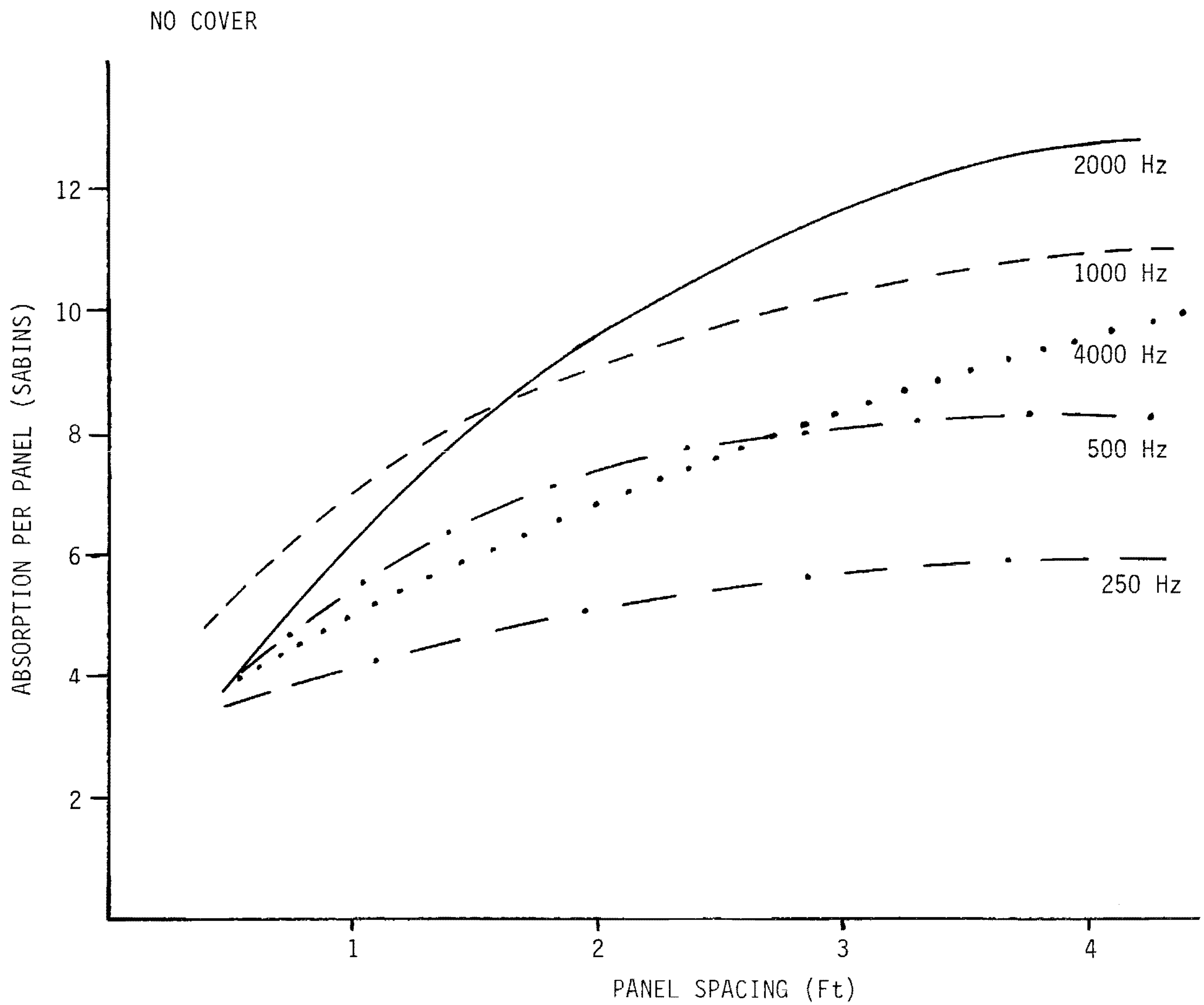


Figure 13 - Panel Absorption as a Function of Spacing
(Uncovered Fiberglass)

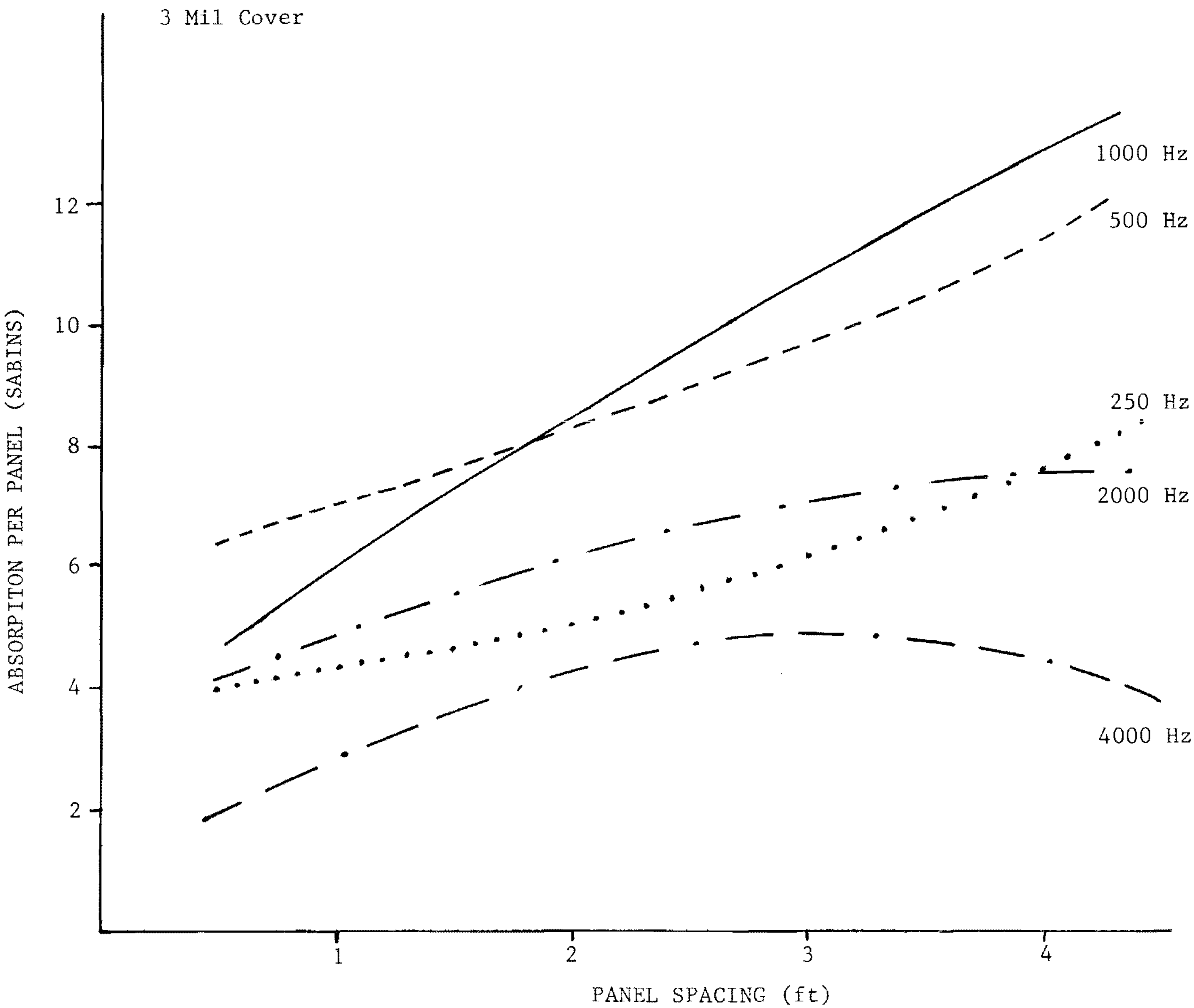


Figure 14 - Panel Absorption as a Function of Spacing
(Fiberglass covered with 3 mil Melanex)

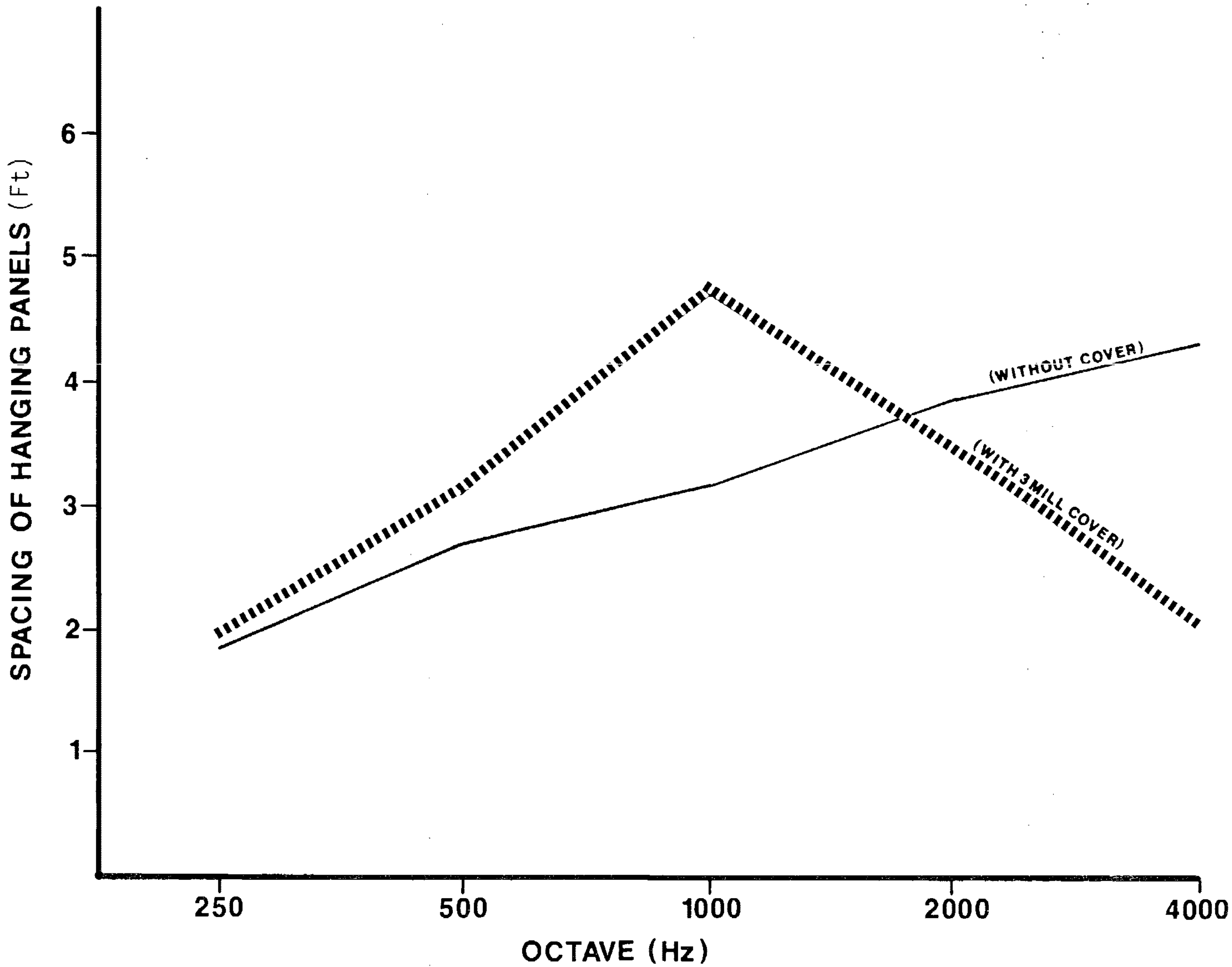


Figure 15 - Spacing at which Hanging Panels have Comparable Absorptive Qualities per Square Foot of Surface Area Covered as Panels placed Flat on Surface

Table 11

ABSORPTION COEFFICIENT OF A CEILING SURFACE COVERED WITH HANGING PANELS

Panel Spacing	250 Hz	500Hz	1000Hz	2000Hz	4000Hz
Uncovered @ 6"	1.000*	1.000	2.000	1.000	1.000
Uncovered @ 1'	.988	1.000	1.000	1.000	1.000
Uncovered @ 2'	.671	.842	1.000	1.000	.919
Uncovered @ 3'	.502	.709	.862	1.000	.660
Uncovered @ 4'	.386	.444	.688	.741	.619
Covered @ 6"	1.000	1.000	1.000	1.000	1.000
Covered @ 1'	1.000	1.000	1.000	1.000	.723
Covered @ 2'	.622	1.000	1.000	.726	.614
Covered @ 3'	.488	.810	.914	.642	.437
Covered @ 4'	.509	.758	.802	.446	.250

* Note surface absorption values greater than 1 are listed as 1.

Utilizing the two pieces of information above, it becomes obvious that there are off-setting considerations which dictate the best panel spacing for a poultry processing plant. As the spacing between panels is decreased, higher absorption per square foot of ceiling area covered is achieved. However, the unit absorption of each panel utilized declines. The net result is a less efficient utilization of the absorbing properties of each panel. A plant may want to set target reductions in reverberant noise levels and select the corresponding panel orientation absorption values needed to achieve this reduction.

One additional study we performed involved placing panels flat against the ceiling between hanging panels spaced 3' apart (see Figure 16). Reviewing this orientation by absorption per square foot of ceiling area covered, we developed the values in Table 12. While the absorption values exceed those for the 3'

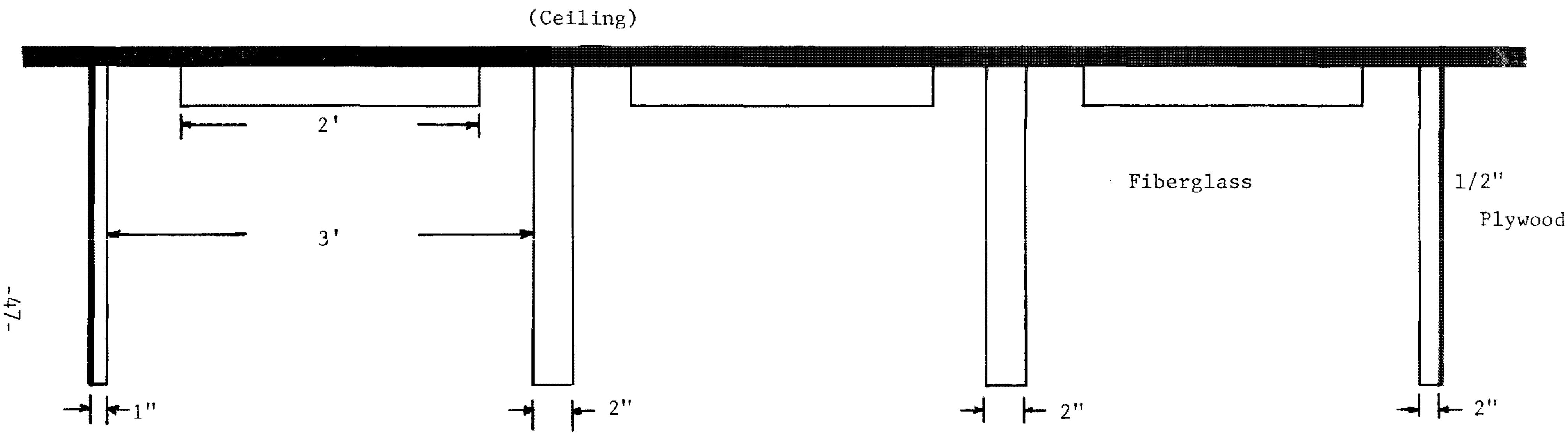


Figure 16 - Schematic of Panel Spacing Test with
Ceiling Panels Added

spacing with no inserts, the additional panels result in the following unit panel absorption rates: 250 Hz (5.9 sabins/panel), 500 Hz (7.1 sabins/panel), 1000 Hz (6.2 sabins/panel), 2000 Hz (5.2 sabins/panel), 4000 Hz (3.5 sabins/panel). If the eight panels had all been hung in a 12' x 4' area, the resulting equidistant spacing between panels would have been 1½'. Using Figure 13, the absorption characteristics of such an arrangement per panel would have been: 250 Hz (4.6 sabins/panel), 500 Hz (7.6 sabins/panel), 1000 Hz (6.8 sabins/panel), 2000 Hz (5.5 sabins/panel), 4000 Hz (3.3 sabins/panel). When the two orientations are compared, there does appear to be some benefit in the 250 Hz octave to placing a panel flat between the hanging orientation. However, the remaining octaves show no appreciable difference in unit panel absorption.

Table 12

ABSORPTION COEFFICIENT OF A CEILING SURFACE
COVERED WITH HANGING PANELS

Panel Spacing	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Covered 3' with panels placed flat in the space between	.990	1.000*	1.000	.862	.580

* Note surface absorption values greater than 1 are listed as 1.

FOCUS ON SOURCE QUIETING

Introduction

Unlike the previous discussion of sound absorbing panels, which has an almost universal application, techniques which quiet noise sources are only effective if they attack the noise generating mechanism and if that source is significant in terms of the overall noise field. As we discovered in our studies, the sources of noise can be many. For instance, a gearbox not properly greased or a drive shaft slightly warped can both produce excessive noise.

It became obvious in our study that proper maintenance was a key factor to holding individual machinery noise levels to a minimum. Yet it also became obvious that machine design often compounded the difficulty of minimizing machine noise levels. In particular, we found fault in the practice of bolting drive motors and pumps directly to large, expansive metal surfaces. This arrangement subsequently amplifies the total sound power emitted as a result of a failing or worn part. Also, there is an industrywide practice of leaving drive motors and pumps exposed rather than covered.

It is in the area of machine design modification and source isolation that we saw the possibility for a general approach to reducing noise levels. The discussions which follow focus on the major sources of noise identified in our previous report[11] along with a few additional sources which offered a potential for significantly contributing to the plant noise problem. This section summarizes our findings regarding source quieting in poultry processing plants.

Current Technology

The current state of the technology for source quieting in poultry processing plants consists primarily of source isolation. For instance, picking machine noise has long been a problem in terms of employee noise exposure in many plants. But it has effectively been dealt with by isolating the machines in a room of their own. Since the machines typically require little attention, only periodic employee exposure takes place.

Improvements made to machinery to enhance productivity, also have lowered noise emissions. For example, hand-held lung guns are being replaced with automatic drawing machines in broiler plants. These drawing machines use no vacuum, which is the source of noise in a lung gun operation.

These events point to the need for an overall awareness of the noise-generating mechanisms common to poultry processing machinery. Rather than to redesign a machine which may soon become obsolete, we chose in this study to evaluate what could be done to a machine, while still in place, to reduce its noise levels. By identifying the noise generating mechanism, we also hoped to make machine designers aware of points to consider in future designs to minimize noise output.

The Chiller

In order to understand the noise-generating mechanism of a chiller, we studied two units.

The first unit was a paddle-type chiller common throughout the industry (see Figure 17). We began the study by filling the chiller with water and

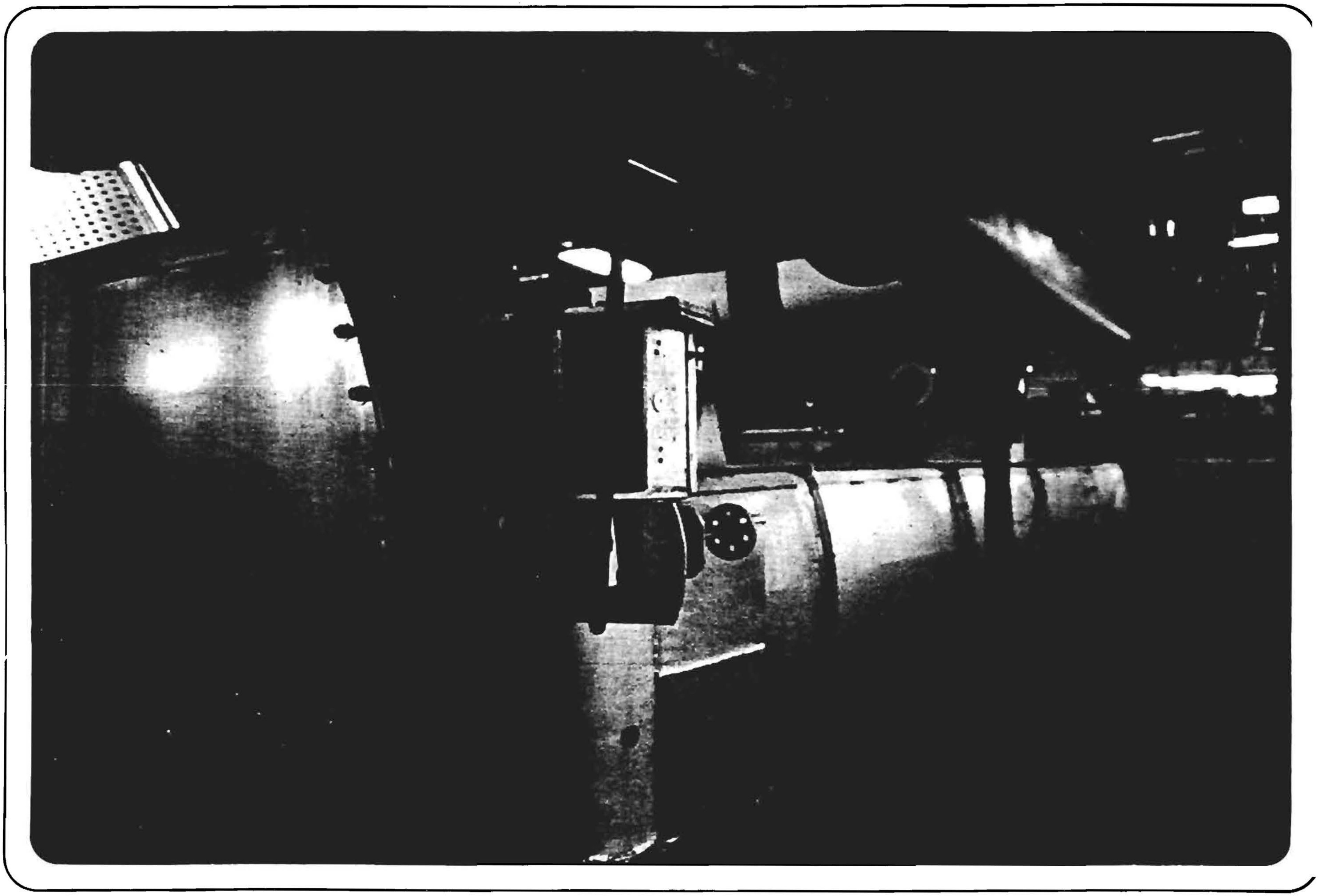


Figure 17 - Paddle Chiller

turning individual components on and off one at a time. The result was a characterization of the noise signature of each component (see Figure 18). When it became obvious that the pumps were the primary noise-generating mechanism for this chiller, we immediately began a detailed study of it.

We first took a series of noise readings on the chiller while varying general conditions. Our test scenario involved operating a single circulating pump on the chiller, first without any water in the chiller and then with water in the chiller. This helped to identify the role water plays in quieting chiller noise. Next, a hood filled with 3" fiberglass padding was placed over the top of the pump housing and drive motor (see Figure 19). This later feature allowed us to reduce noise emanating directly from the pump top and drive motor. Figure 20 shows the impact on the chiller's noise spectra when water was added. The overall level dropped 3.7 dB. Figure 21 shows the impact on the spectra when the hood was then added. The overall level dropped an additional 3.3 dB.

The information in Figures 20 and 21 point out that a tremendous amount of sound energy emanating from the chiller is low frequency (250-2000 Hz). When the drive motor and pump housing top are covered, a full 3.3 dBA reduction in sound pressure level is observed, yet the primary reduction brought about by the hood is centered in the 2000-6000 Hz range. A possible explanation for this occurrence is that the pump and drive motor are unquestionably the noise-generating mechanisms (nothing else mechanical is operating), yet the pump housing and main chiller body are amplifiers of the noise generated. Due to their size, they are more efficient in acoustically transmitting the low-frequency portion of the generated spectrum. Another possible explanation

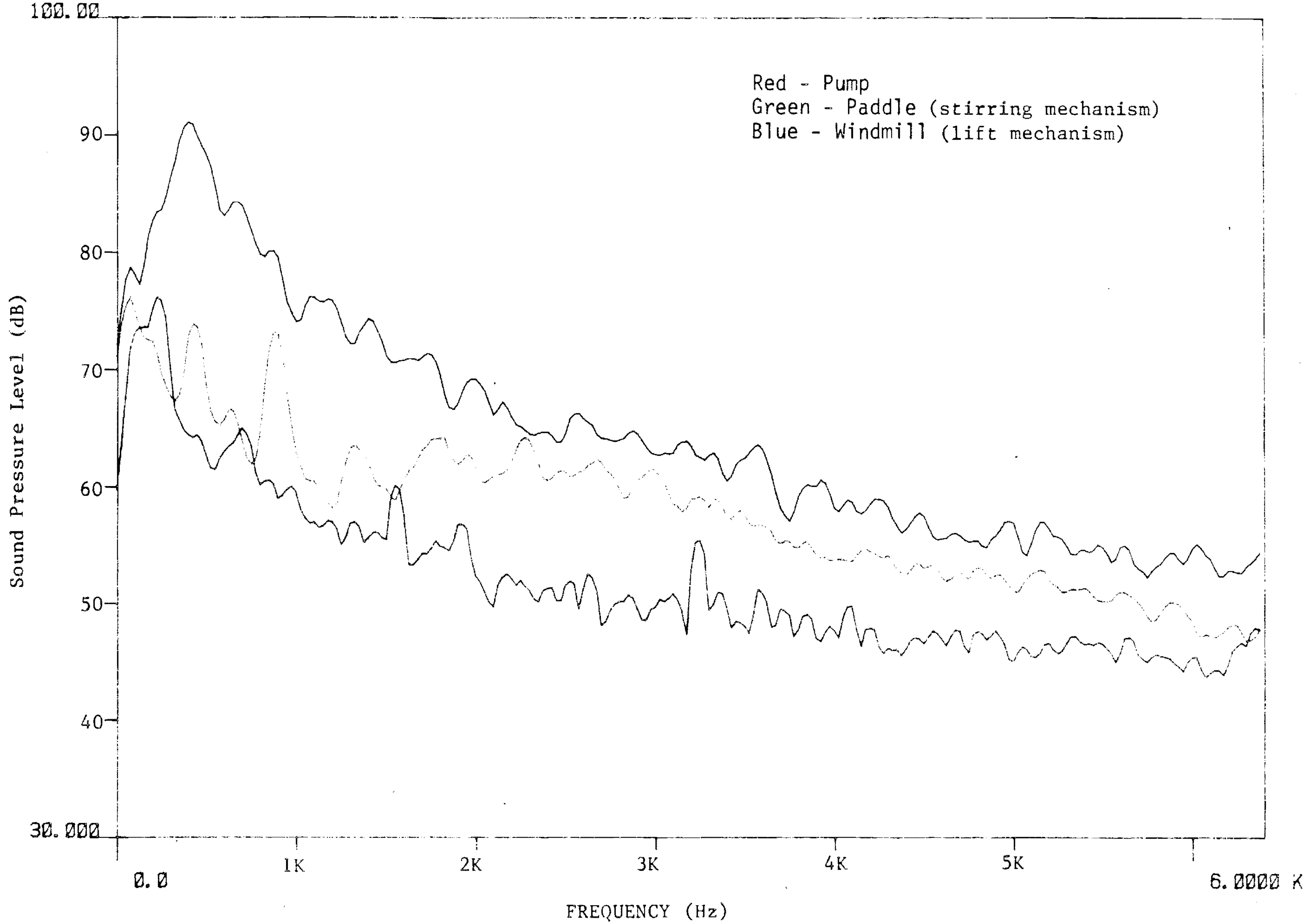


Figure 18 - Spectrum for Individual Components on a Paddle Chiller

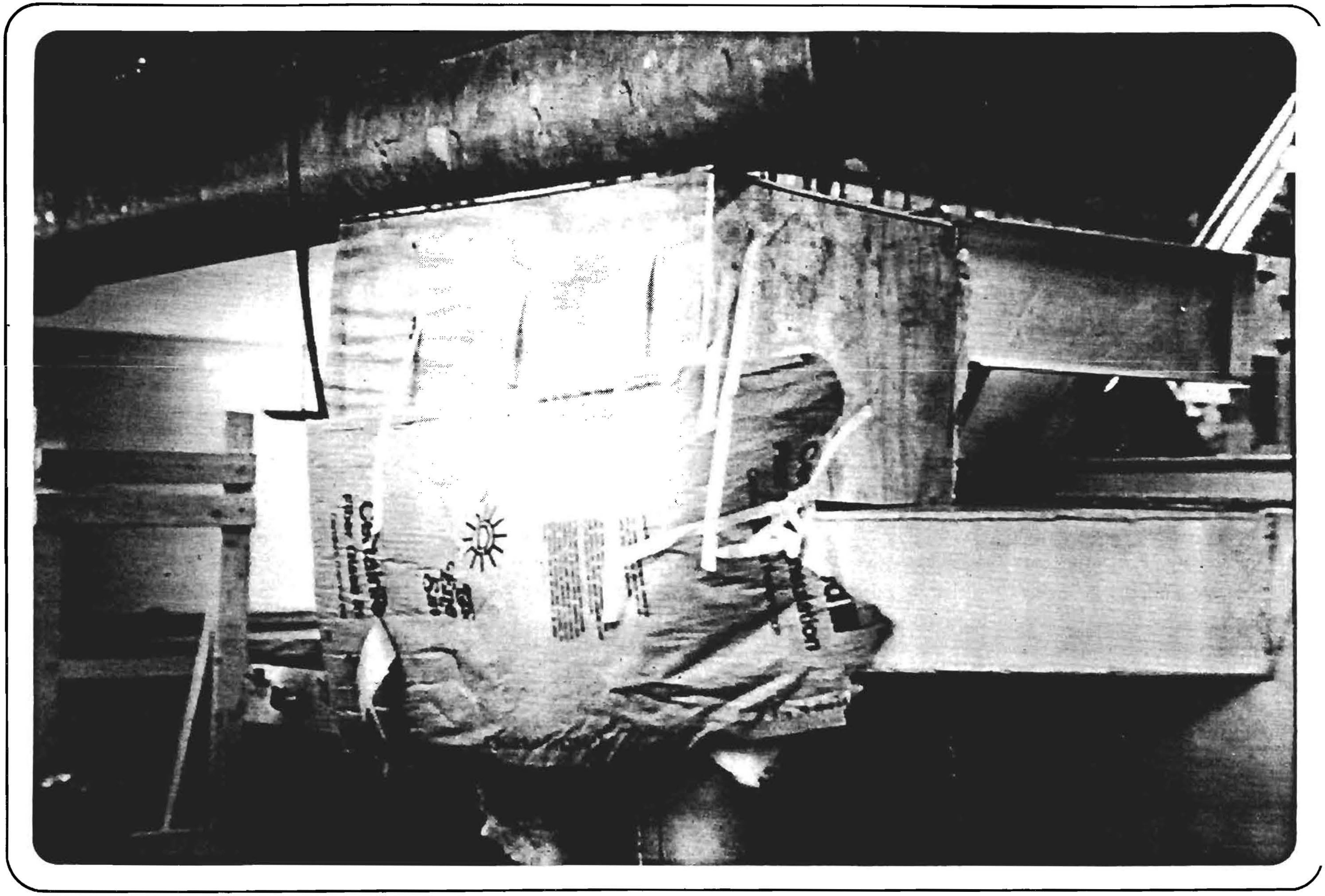


Figure 19 - Hood Mounted over Pump Housing and Drive Motor

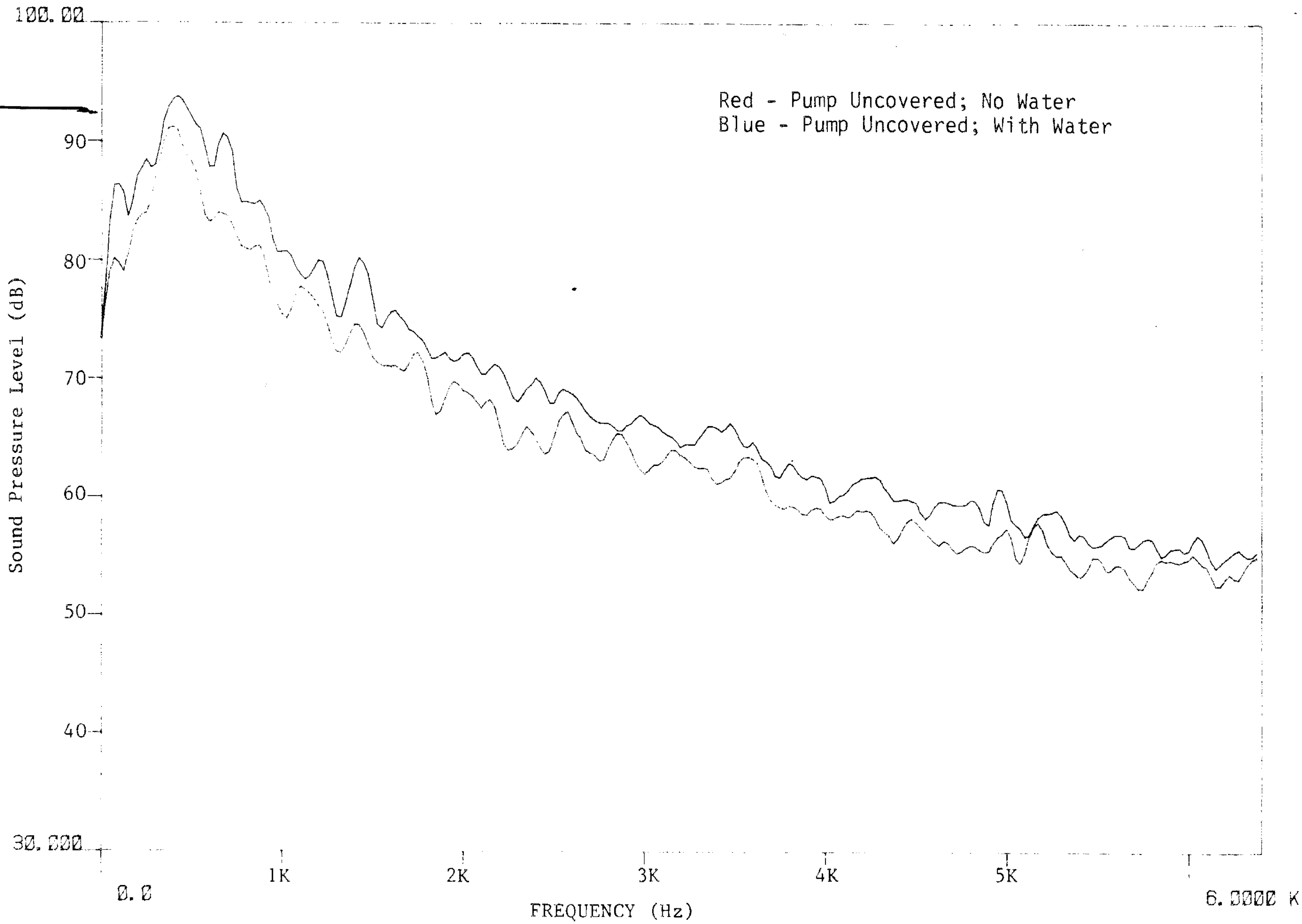


Figure 20 - Sound Energy Spectrum Changes in the Paddle Chiller

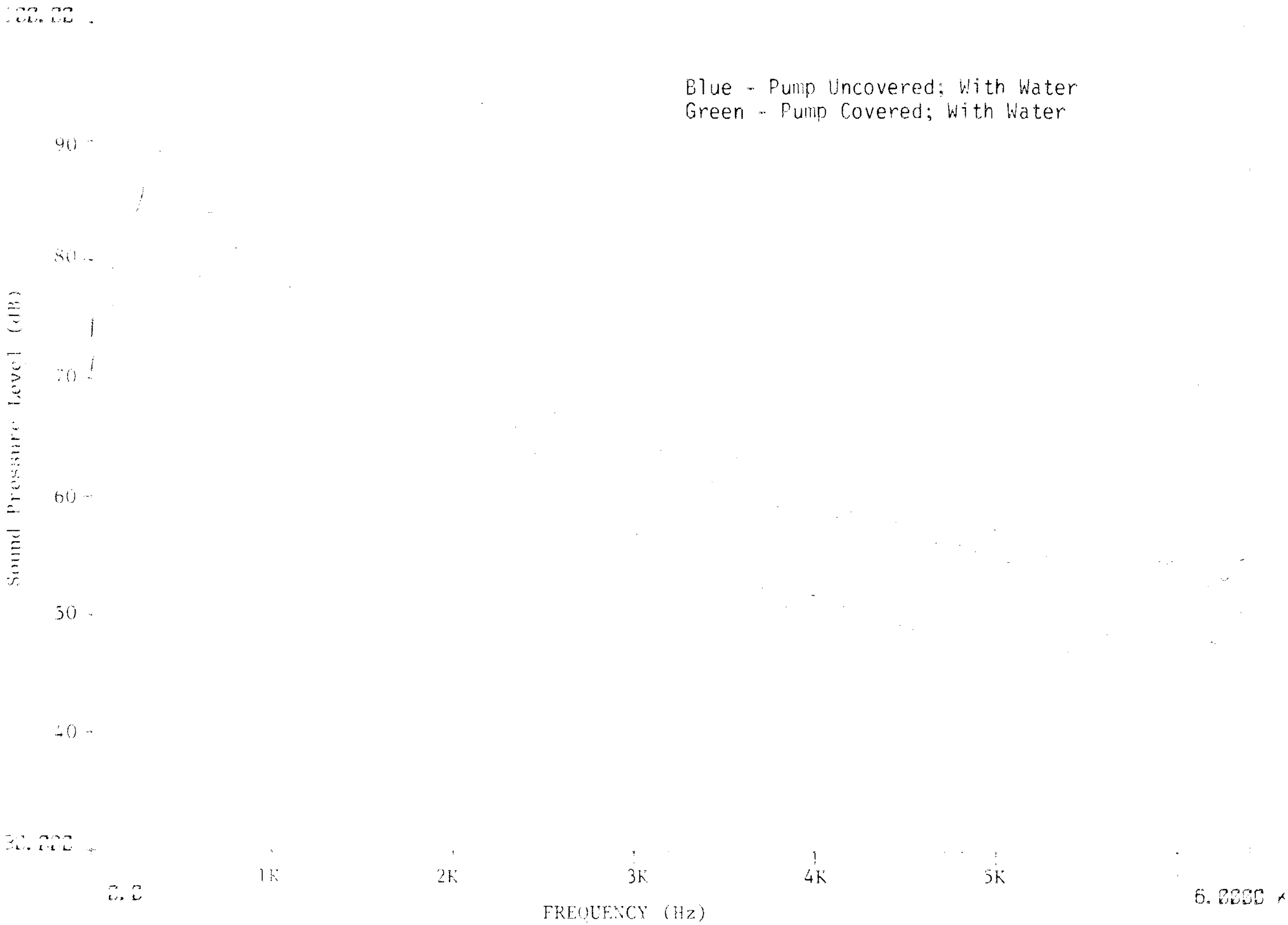


Figure 21 - Sound Energy Spectrum Changes in the Paddle Chiller

is that the acoustic hood was not effective in reducing low-frequency noise and that the pump was the only major noise-generating mechanism.

Using accelerometers and the equipment orientation shown in Figure 22, we attempted to perform coherence and cross correlation analysis to determine the noise signature of various parts of the chiller. Unfortunately, the accelerometer signals contained strong periodicity which prevented our using these techniques properly.

Our conclusion on chiller quieting, therefore, was inconclusive. We do believe that bolting pumps and drive motors directly to a chiller body allows the chiller body to amplify vibrations resulting from their operation, and the sound power transmitted from a chiller potentially can be held in check or possibly reduced by isolating the drives and pumps from the body. This alone, however, may not eliminate the problem, since the drive motors and pumps, by themselves, could be the major noise-generating mechanism. Consequently, where possible, drive motors and pumps also should be enclosed in a sound-absorbing hood.

As mentioned earlier, we studied two chillers. The second chiller was a giblet chiller (see Figure 23). During the course of our study, we discovered a problem in the gear box, and the gear box and drive motor were subsequently replaced. Figure 24 shows the change in sound energy transmitted by the chiller when the replacement was made. Overall, a 16 dB drop in sound pressure level was observed. Clearly this dramatizes the need to isolate and enclose the drive mechanisms. It also points out the need for good maintenance programs.

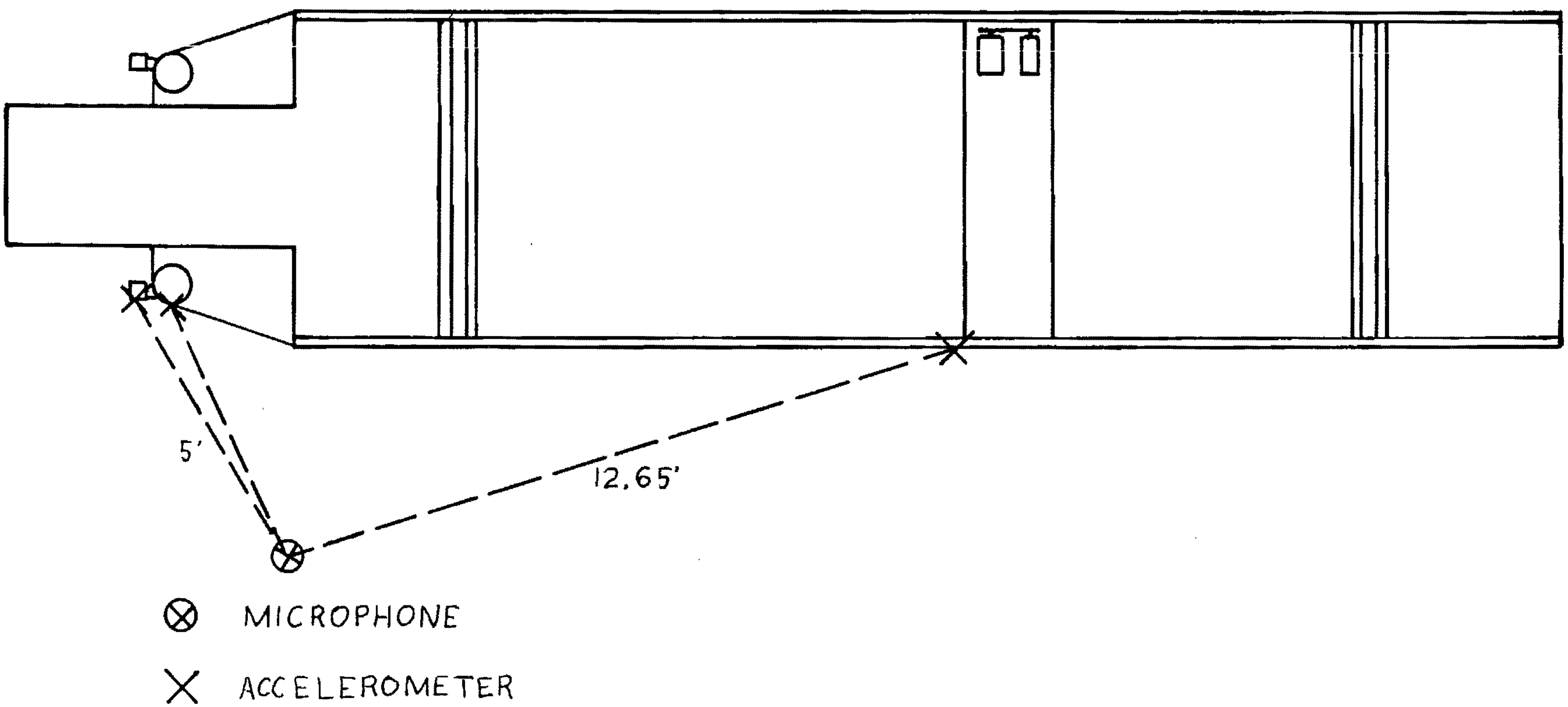


Figure 22 - Accelerometer and Microphone Locations for Chiller Analysis

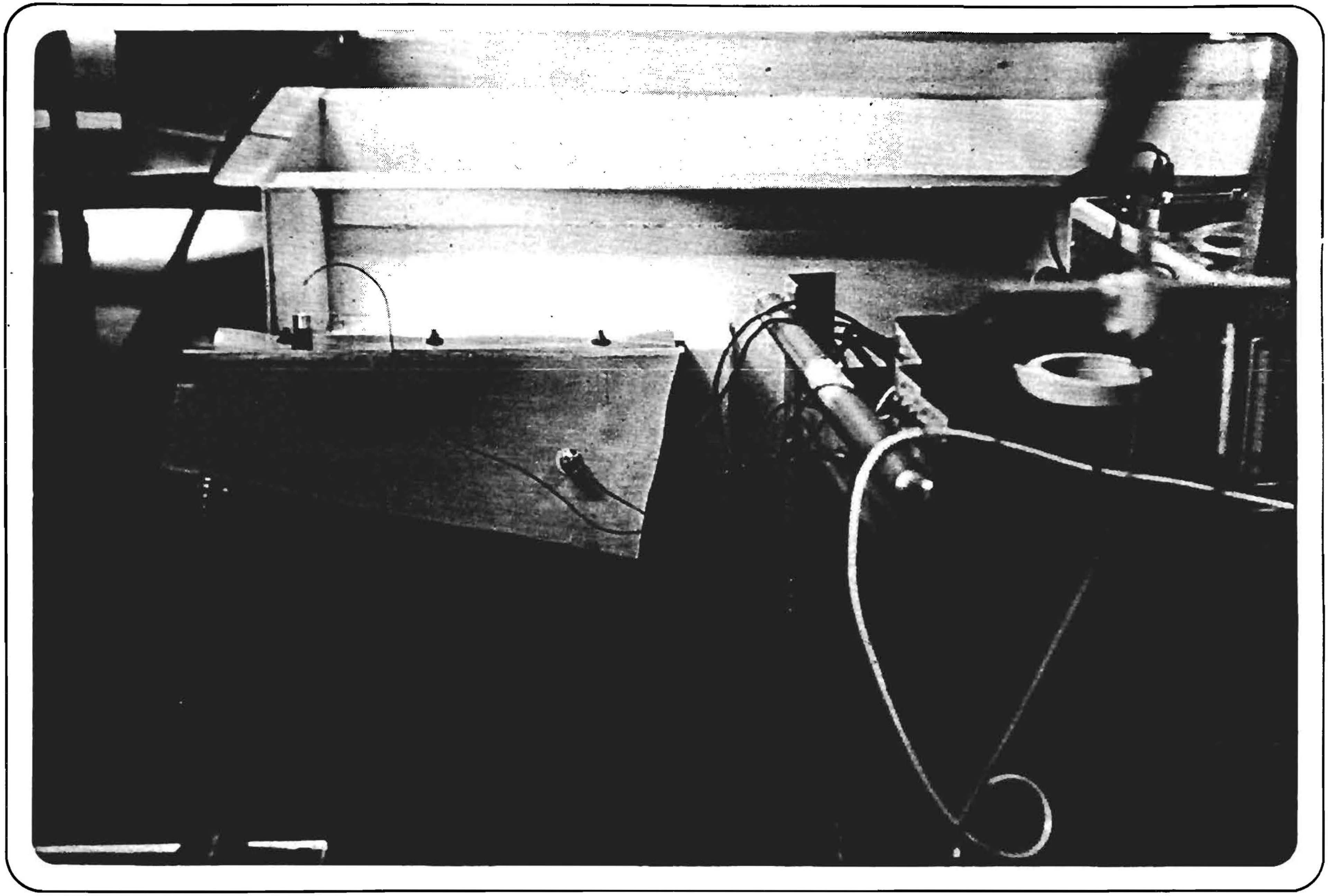


Figure 23 - Giblet Chiller Undergoing Tests

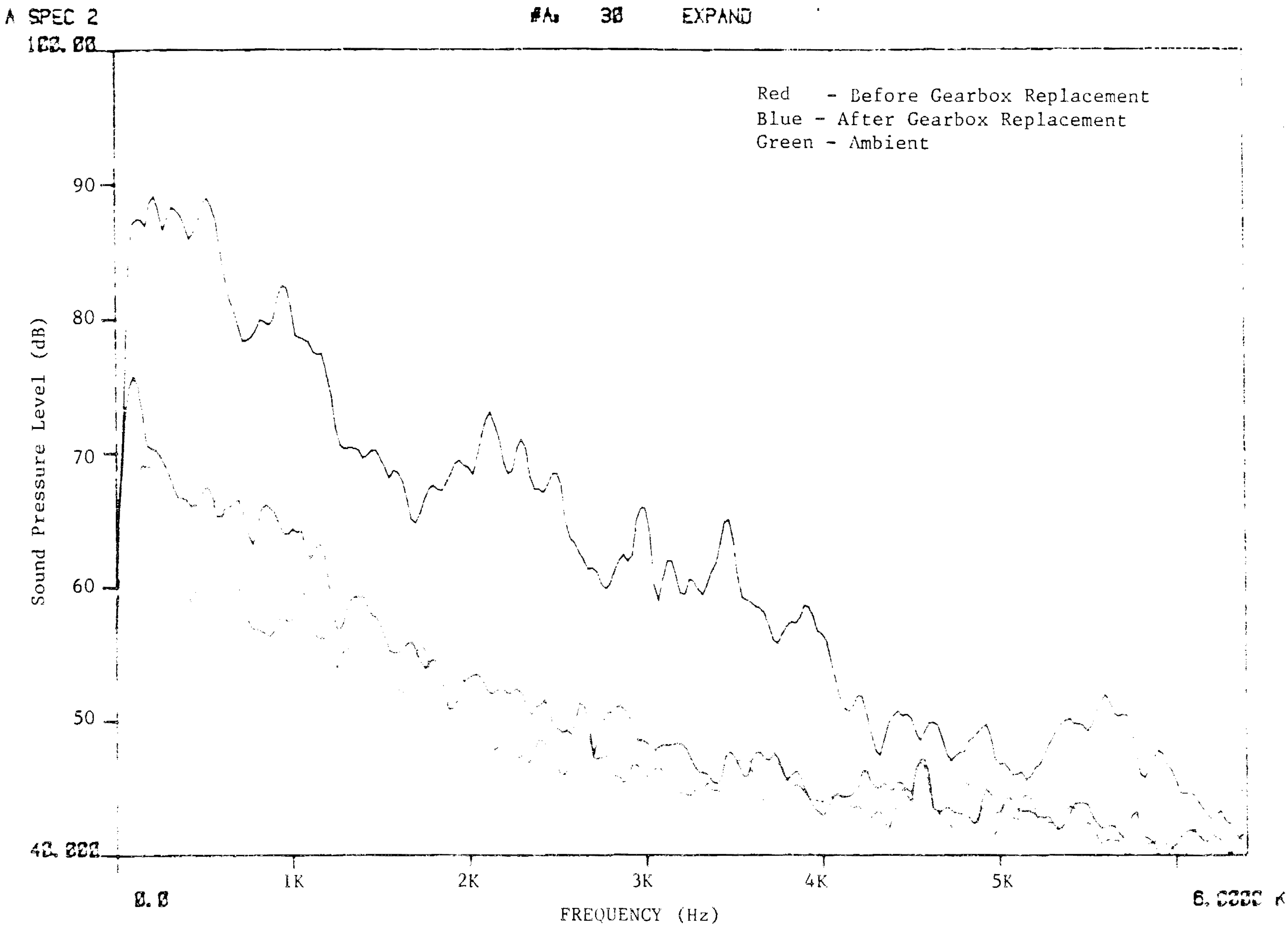


Figure 24 - Sound Pressure Spectrum Changes In The Giblet Chiller

The Lung Gun

Lung gun noise is currently being alleviated by many firms who are replacing them with drawing machines that also pull out lungs. Unfortunately, not all plants can use drawing machines, either because of financial constraints or because they process birds of varying size which cannot be processed with existing drawing devices. In a plant where lung guns were replaced by drawing machines, a dramatic reduction in noise level was observed (see Figures 25 and 26).

Lung gun noise is caused by suction pressure between the surface being cleaned and the gun nozzle. Because this operation takes place in a cavity, resonances are set up which amplify the noise level.[12] Typically, several lung guns are operated within close proximity to one another, compounding the problem (see Figure 27).

In dealing directly with the source, we are aware of at least one research effort[13] which culminated in a hood (see Figure 28) on the lung gun to block the opening to the cavity. While the design lowered noise levels 12 dB, it proved impractical in actual operation because operators complained of obstructed visibility in performing the lung removal.

One method that has worked is to reduce the vacuum on the gun to a level just necessary to perform the pulling function properly. Discussions with one plant indicated that by reducing excessive suction, they lowered sound pressure levels nearly 10 dB at the lung gun stations.

Another method that can work in dealing with lung gun noise is to place plastic barriers between each lung gun station. Since each lung gun is a

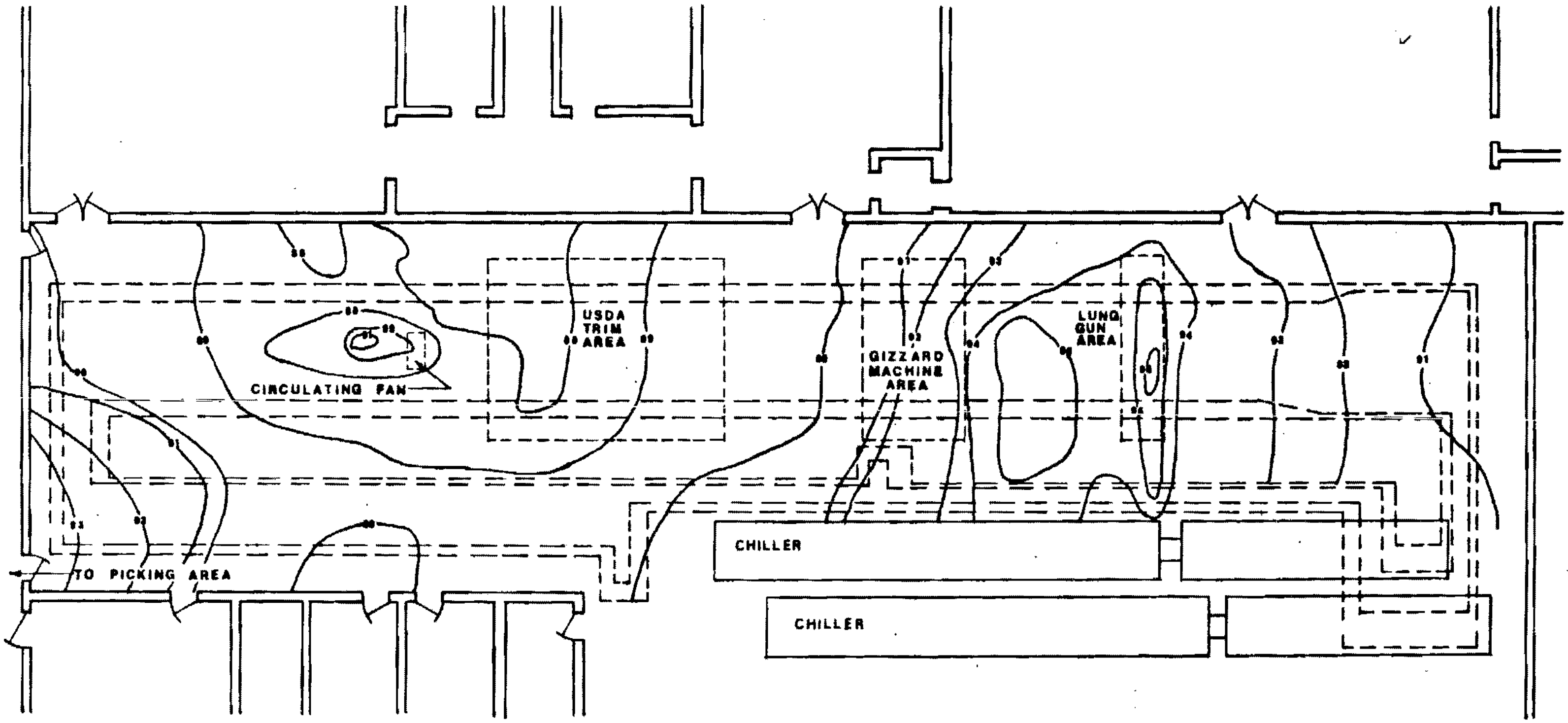


Figure 25 - Noise Contour in Plant with Lung Guns

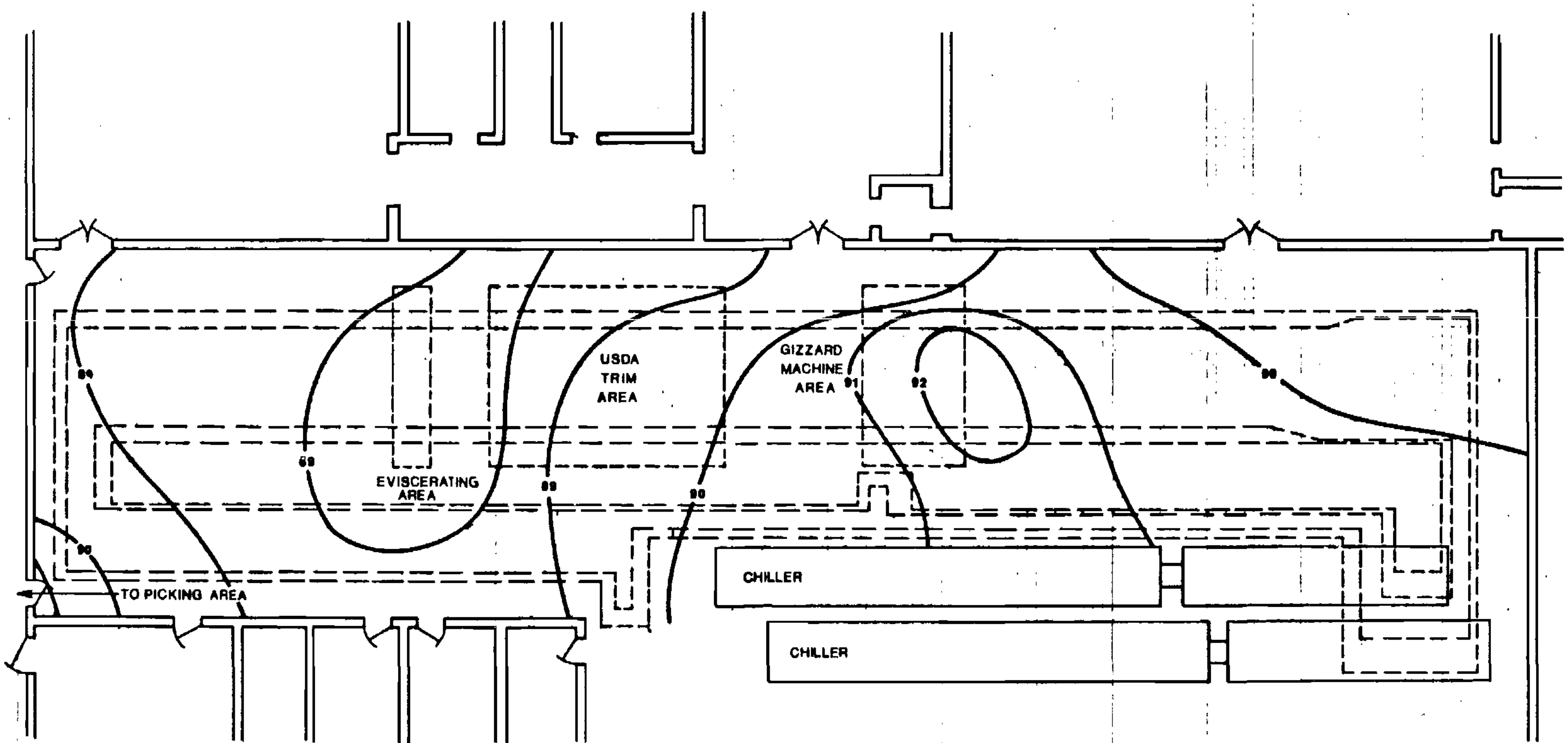


Figure 26 - Noise Contour in Plant with Lung Guns Replaced with Drawing Machines

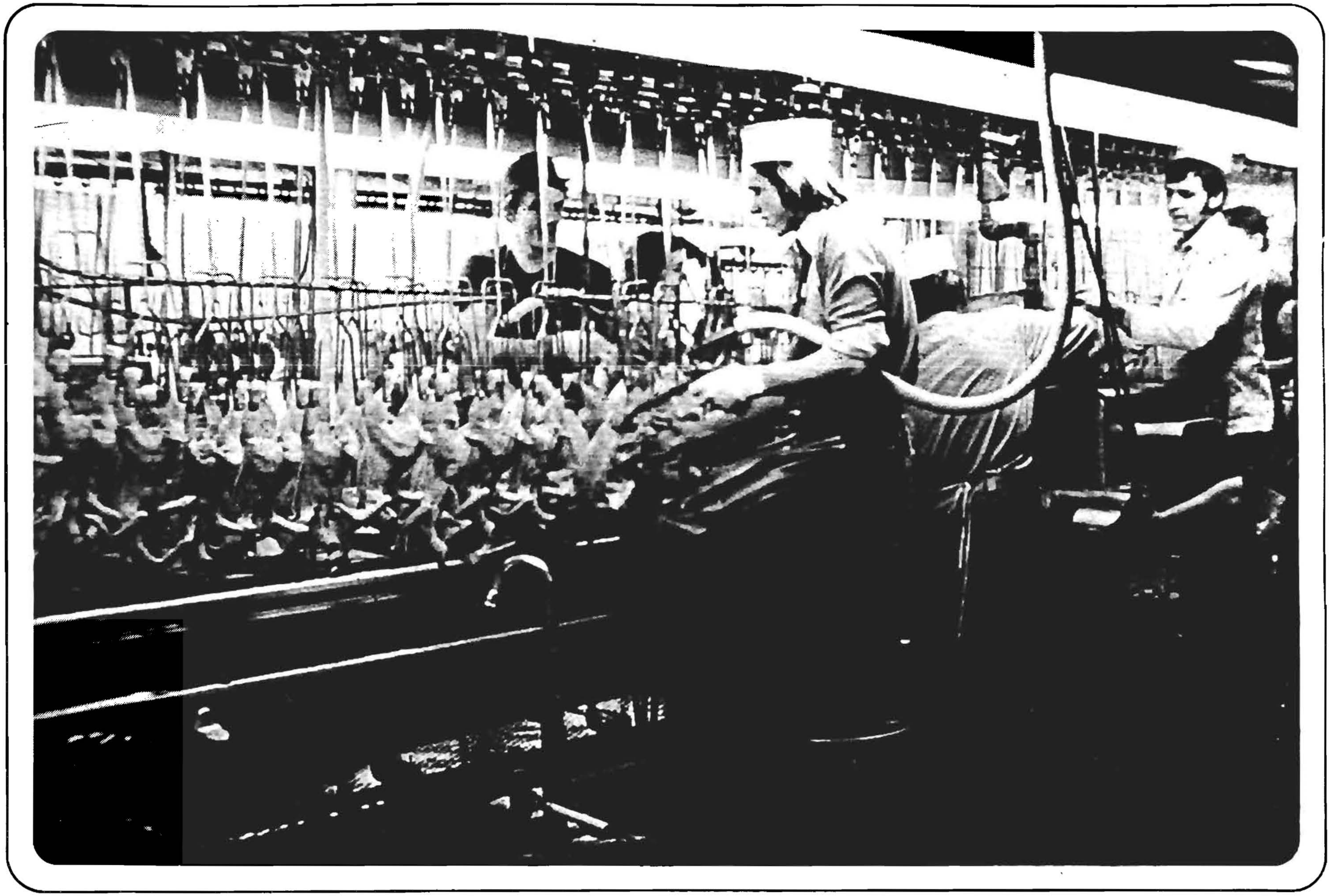


Figure 27 - Typical Lung Removal Area with Hand Held Lung Guns

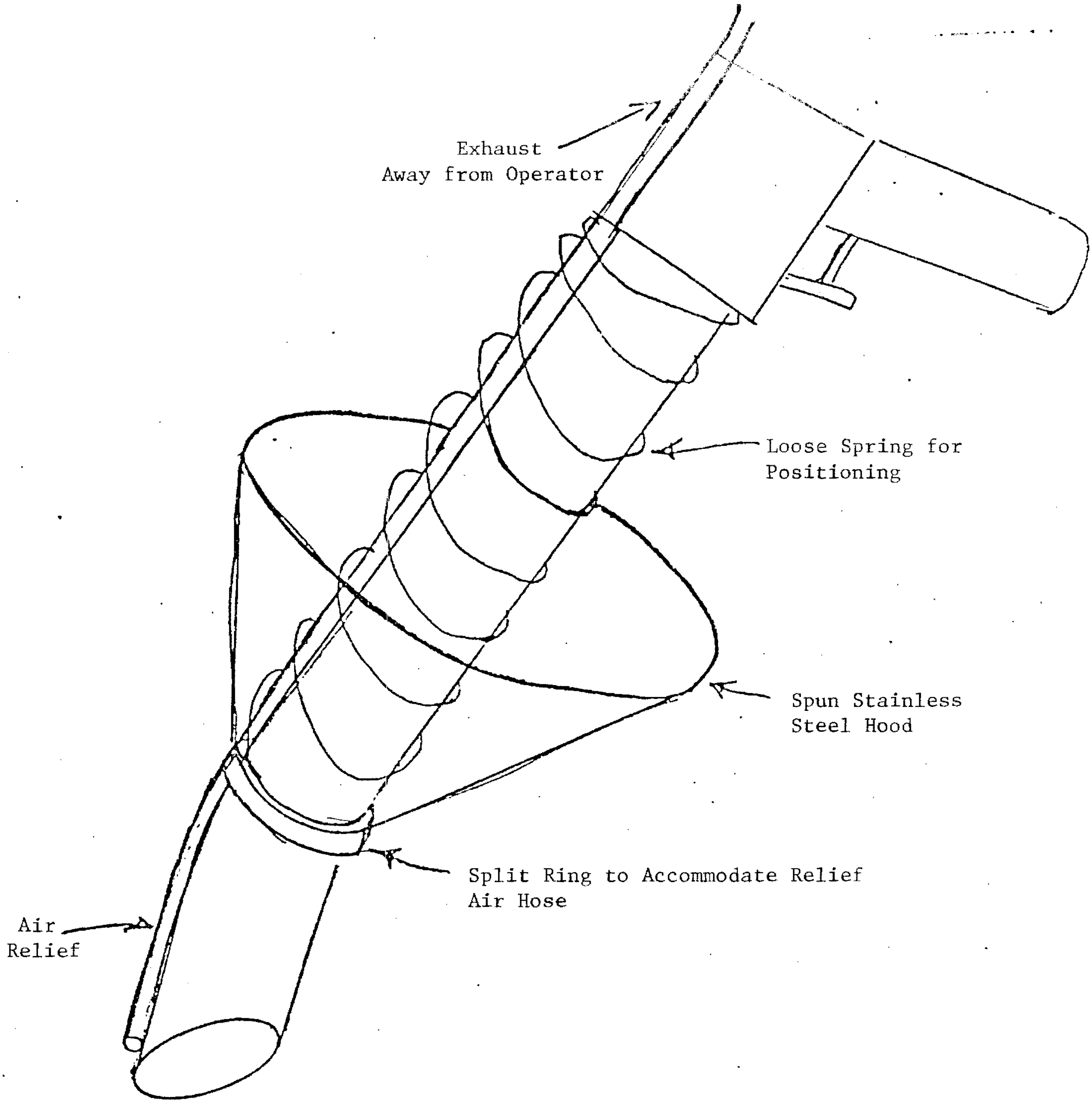


Figure 28 - Lung Gun Hood

single source, isolating the sources can have a tremendous impact on local sound pressure level readings both at and near the lung removal stations. One experiment showed as much as a 14 dB reduction in sound level at the station adjacent to the lung removal operation when a barrier was used.[14] There are, however, problems with putting up barriers, problems which are primarily related to employee morale. In one plant which experimented with vinyl curtains, lung operators systematically cut the curtains down, apparently because they did not like being isolated from fellow employees. This problem perhaps can be overcome by using partial barriers which block the path of direct sound but still allow face to face contact between employees. We further recommend that an absorptive hood be placed directly above the work station, when barriers are used, to prevent sound pressure level buildup at the station.

The Hock Cutter

Hock cutter noise reduction has been accomplished largely by isolating the machine from personnel. Figures 29 and 30 display the observed noise reduction brought about by relocating the hock cutter to another area of the plant. However, isolation techniques are not always successful, either because a large opening is used to convey the birds back into the evisceration room or because many plants still require personnel to work near the machine after isolating it.

A review of the basic design of the hock cutter yields only a few possible explanations for the appreciable noise levels generated by this device. The typical hock cutter has a drive mechanism clustered to one side of the machine which is completely exposed except for a sheet metal safety

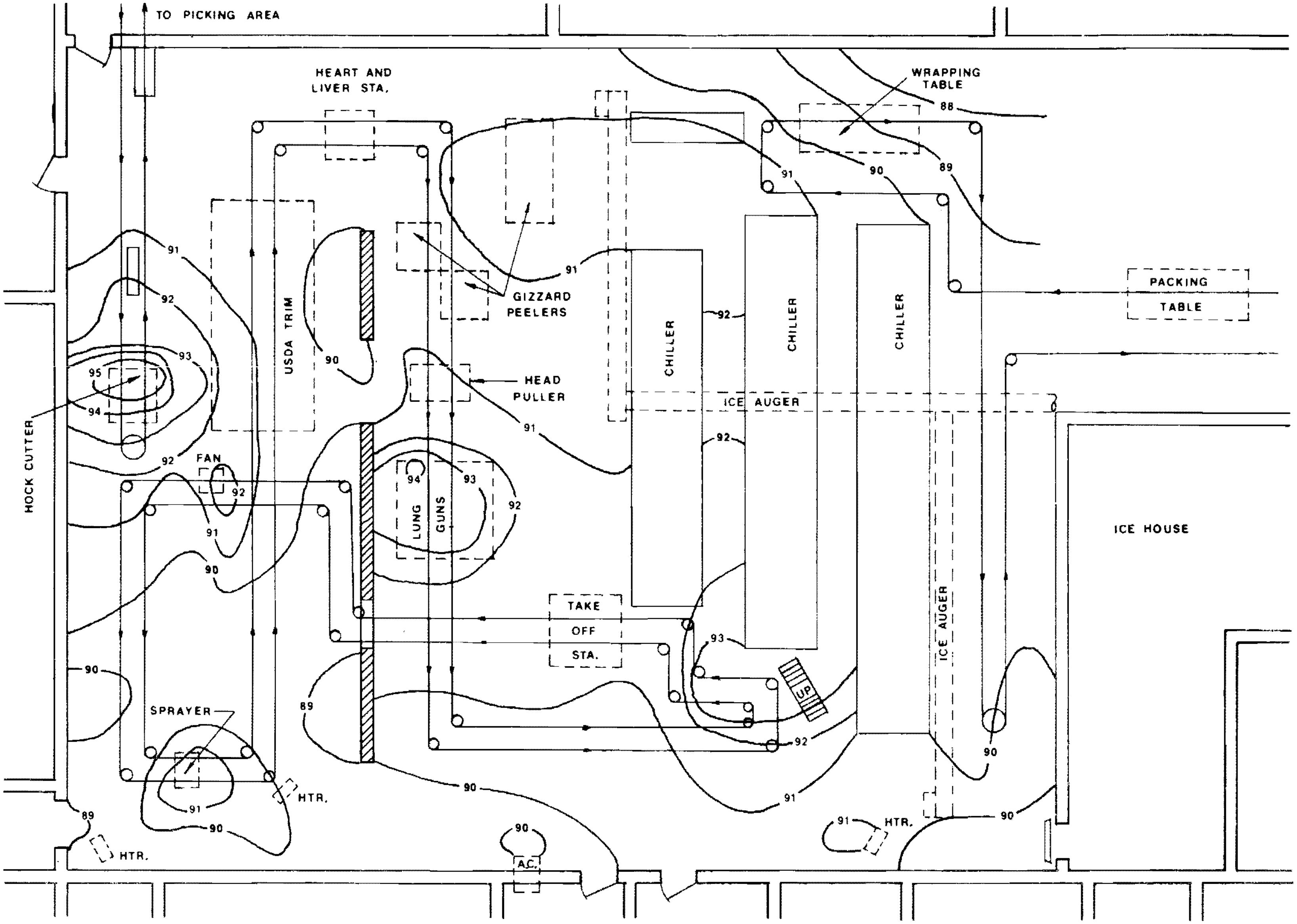


Figure 29 - Noise Contour in Plant with Hock Cutter

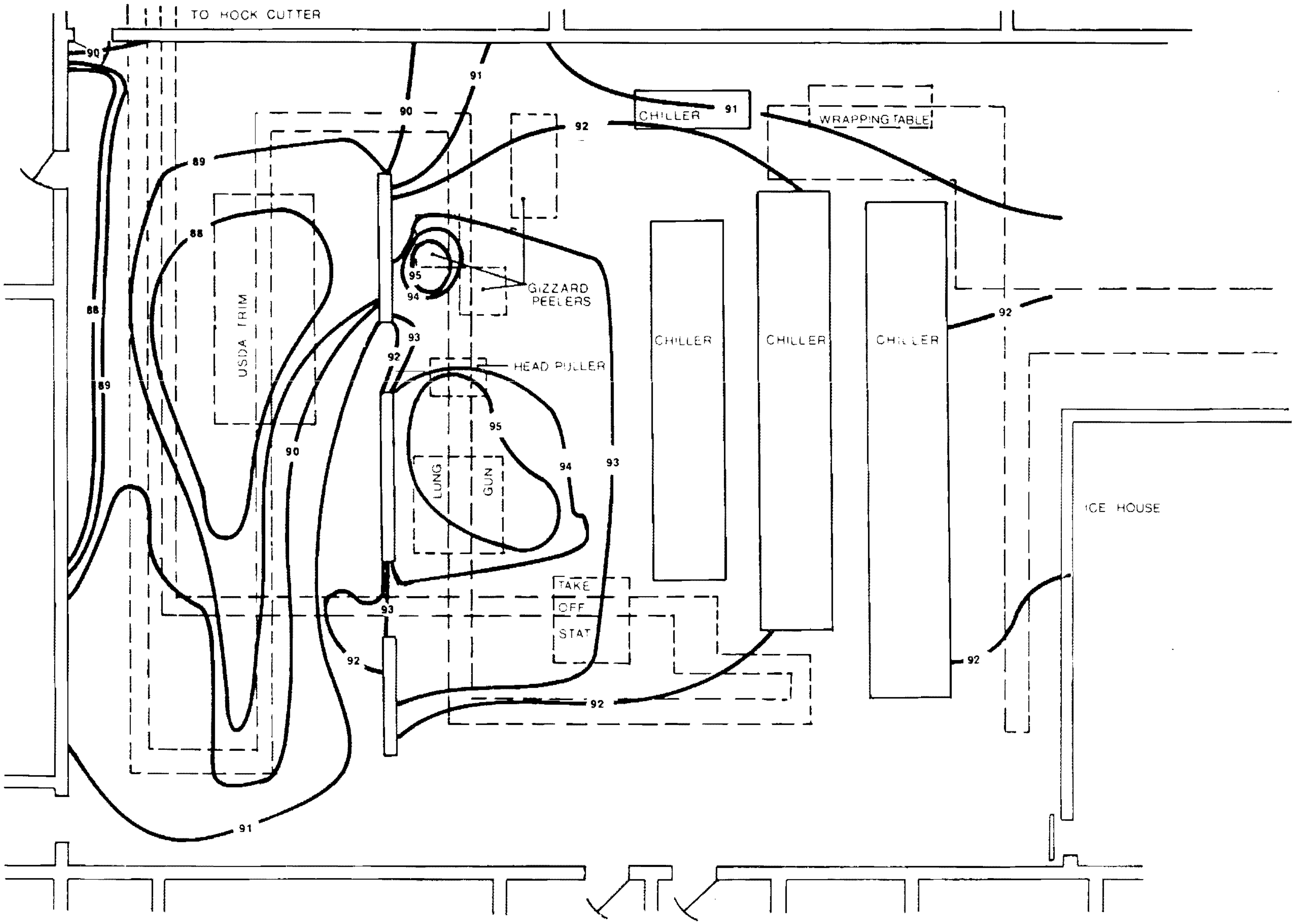


Figure 30 - Noise Contour in Plant with Hock
Cutter Removed

cover plate. These drive motors and drive belts all offer the potential for producing high noise levels.

As a means of attempting to quantify the contribution of the drive motors in generating noise, we mounted accelerometers first on one of the drive motors and then on the frame of a hock cutter and observed the relationship between these transducers and a microphone positioned five feet away (see Figure 31). Again we were unable to utilize coherence or correlation analysis techniques because the accelerometer signals were very periodic.

Therefore, we attempted to quiet the noise source with an enclosure packed with sound-absorbing material. Using a partial housing constructed of plywood and fiberglass (see Figure 32), we enclosed the drive area of the hock cutter. Nearly a 4 dB drop in sound pressure level was observed at the microphone position (see Figure 31). Figure 33 displays the change in the sound pressure spectrum observed during this series of tests.

The Vent Cutter

Vent cutter noise can contribute significantly to local noise levels in a plant. A vent cutter in many cases is merely a pneumatic drill used to open the bird for subsequent evisceration. While newer machine designs exist whereby the drilling is performed automatically by mechanical drive mechanisms, for the pneumatic tools that continue to be used, we evaluated the potential effectiveness of exhaust mufflers. The muffler we selected was a polyethylene design (see Figure 34) which was washable and rugged.

Figure 35 shows the change in sound spectra, measured one foot away from the exhaust part, both before and after muffler attachment. A noise reduction

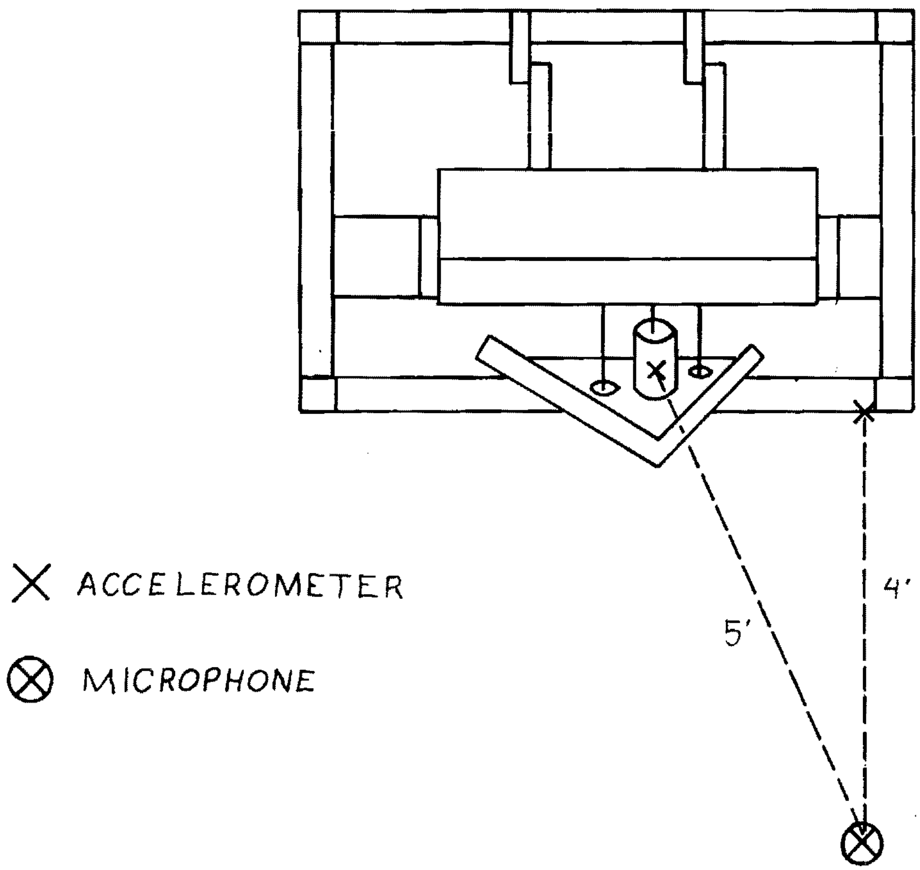


Figure 31 - Accelerometer and Microphone Locations
For Hock Cutter Analysis

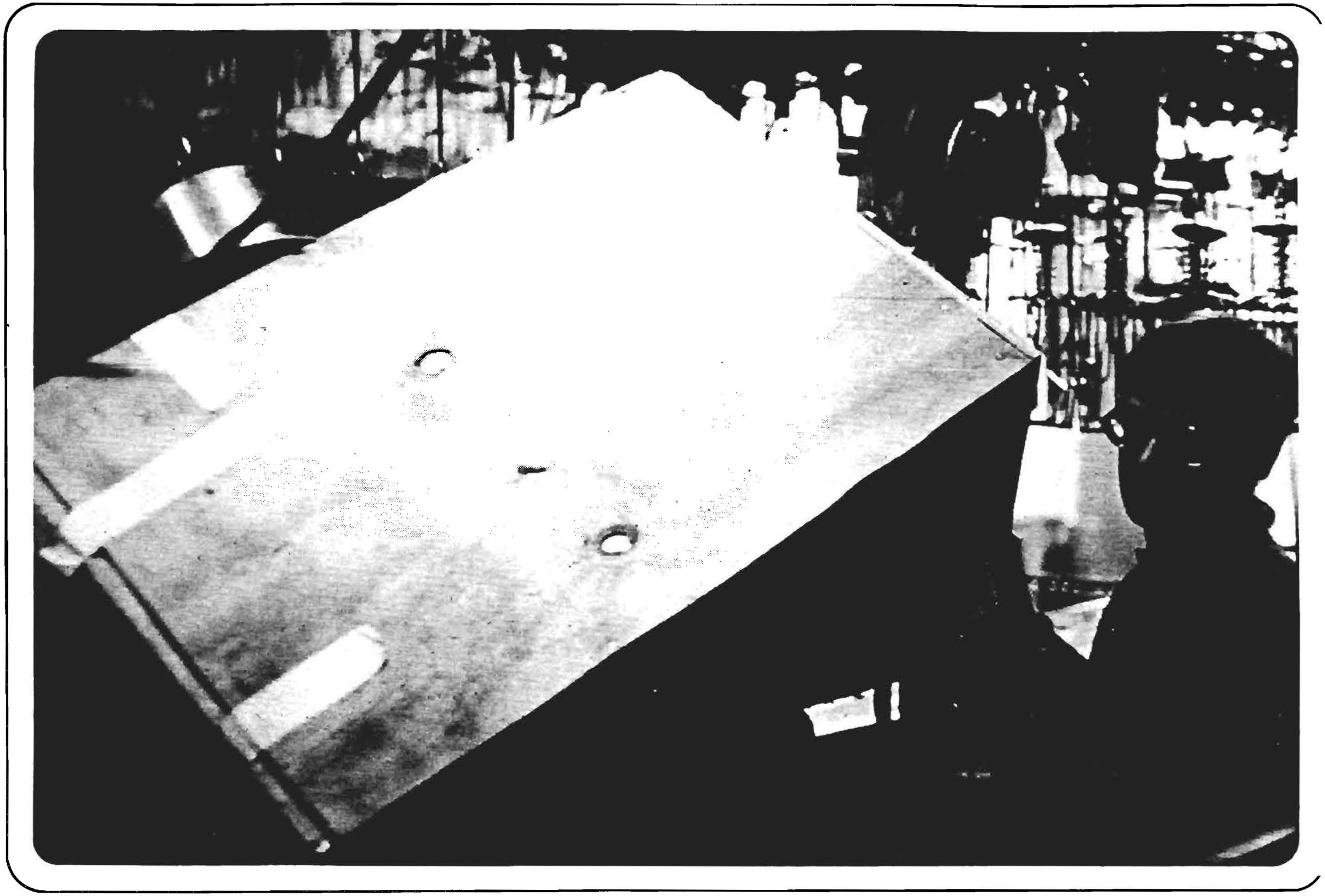


Figure 32 - Partial Housing Placed Over Hock Cutter Drive Motors

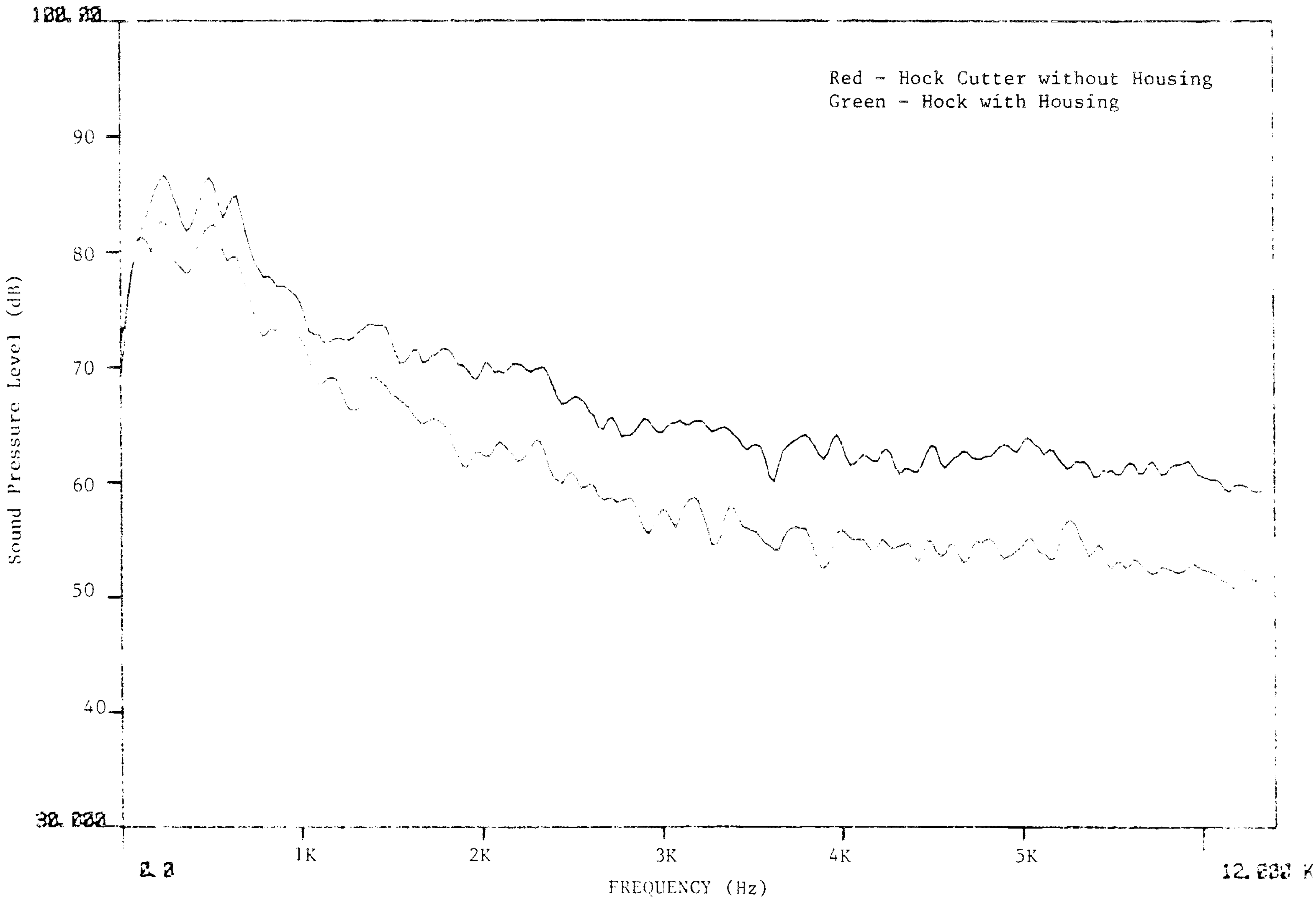


Figure 33 - Sound Energy Spectrum Changes in the Hock Cutter

Figure 34 - Pneumatic Muffler Tested

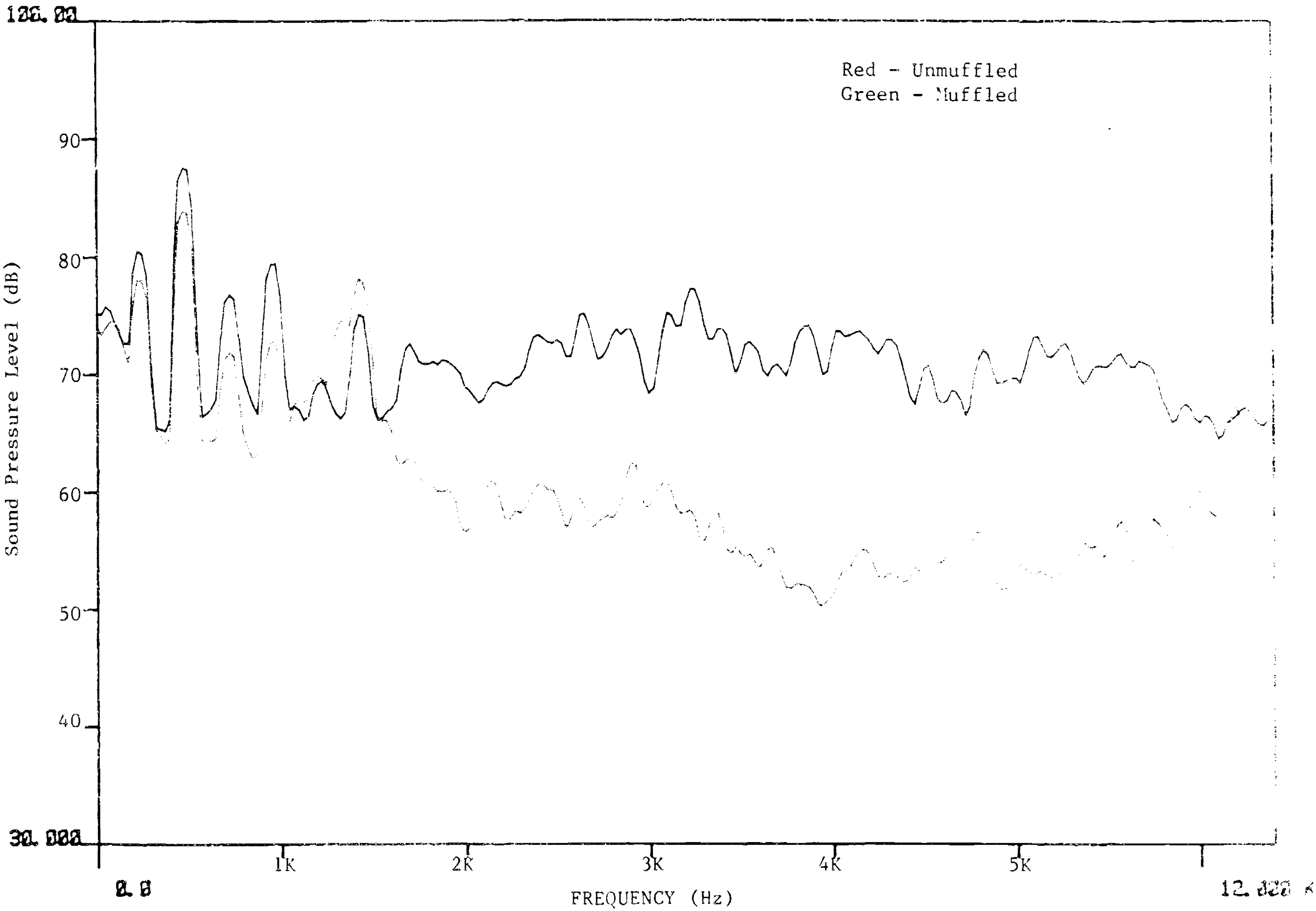


Figure 35 - Sound Energy Spectrum Changes in the Vent Cutter

of 5 dB was observed.

Perhaps the only potential problem with utilizing these muffling devices is their potential for plugging if the air supply is not properly filtered. However, if the air supply filter is working properly, then the muffler we tested should offer little obstruction to normal tool operation. An arrangement is possible, using exhaust hoses on each tool connected to a central overhead exhaust header, to minimize the potential for plugging on air systems with marginal filtration efficiency.

Ice System

Ice troughs and dump stations are another potential noise problem area. While they do not always emit high levels of broadband noise, discrete frequency discharges can produce appreciable noise levels observable above the general din.

Fortunately, energy conservation efforts can help to justify putting jackets on ice troughs. These jackets, if properly designed and maintained can also reduce noise levels associated with ice transport.

Ice drop stations also provide noise-generating mechanisms because of metal-to-metal and ice-to-metal contact. As a means of dealing with this problem, metal-to-metal contact can be minimized through gasketing of contact points. Ice-to-metal contact noise also can be minimized either by utilizing exterior vibration dampening material on the metal surfaces or by replacing the metal with plastic parts. Many modern plastics exhibit excellent strength qualities as well as vibration suppression qualities, making them ideal. As an example, an auto assembly plant has utilized a new plastic to replace a metal component in its assembly line pull chain[15]. The plastic exhibited excellent strength characteristics while also greatly reducing chain noise.

Vibration Monitoring

As pointed out earlier, maintenance is an important feature in reducing overall noise emissions in poultry processing plants. One method of identifying machines in need of repair is to take periodic vibration readings on critical components itmes (such as motor drives, etc.).

We acquired a portable vibration meter, for approximately $1,000, which was quite useful in taking quick and reasonably accurate vibration readings.* Our meter provided both velocity and displacement data. It was useful more than once in pinpointing excessive vibration levels.

*The meter purchased was a model 306 vibration meter manufactured by IRD Mechanalysis. This mention of the meter does not constitute its endorsement by the Georgia Tech Engineering Experiment Station or any of the project sponsors. This mention is for informational purposes only.

CONCLUSION

Workable solutions to the poultry processing plant noise problem do exist. Our study indicated, however, that care has to be given to durability and practicality.

In the area of absorption, a major weakness in current panel designs is the use of PVF film covers. On a typical panel, the cover amounts to approximately 10% of the total cost. If a stronger covering material is chosen, the cover could rise to nearly 40% of total cost, but it is the cover that is the critical design element of the panel. When it fails, the entire panel, not just the cover, must be replaced. Hence, we must conclude that cover design is a major factor in panel design and should impact panel selection.

We also concluded that using a vertical hanging arrangement is an efficient way to utilize noise panels. Our research showed that 3-foot spacings approximated the absorption characteristics, per square foot of ceiling covered, obtained by laying panels flat against the surface, but represented a one-third savings in the amount of material used. We also must note that through the use of a hanging arrangment, tighter spacing can be utilized to actually increase total absorption and to improve low-frequency absorption. However, these increases come at a progressively higher cost because of the greater volume of panels required to cover a given area.

In the area of source quieting, a major weakness was found in common plant maintenance procedures. Improperly maintained machines can very easily become major noise problems. A schedule of periodic vibration checks, using a portable vibration meter, is a good way to spot machinery in need of immediate attention.

Modifications to machine design, on the other hand, can reduce the potential impact of maintenance lapses. Chiller designs, for instance, should have drive motors and pumps decoupled from the main chiller body. Drive motors and pumps should be enclosed in hoods lined with absorbing medium. These measures can prevent a failing part from leading to a major noise problem.

Hock cutter noise seems to be attributable to the drive motor area. With the inclusion of an absorptive hood over the drive mechanisms, sound pressure levels near this device have the potential for significant reductions.

Lung gun noise, admittedly, is difficult to abate. Because of sanitary restrictions, we foresee no immediate quieting measure to deal directly with the noise-generating mechanisms. Automatic drawing machines do provide suitable substitutes, in many cases, to lung guns and have substantially lower noise levels because of the absence of a vacuum. Where a lung gun must be used, we believe partial barriers between the stations constitute a plausible solution for sound containment. However, to be fully effective, we further suggest that an absorbing hood be placed immediately over the station to minimize sound buildup.

Pneumatic tools should have exhaust mufflers placed on them to reduce noise levels in the immediate vicinity of their operation. Likewise, energy conservation measures can lead to lower noise from the ice transport system since insulation can be specified to reduce thermal loss and sound generation and transmission.

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2. Personal conversation with Robert Huschle and G. T. Scheldahl.

3. Personal conversation with Cliff Miller, E. I. Du Pont de Nemours & Co., Inc.

4. American National Standards Institute, *Standard Methods of Test for Sound Absorption of Acoustical Materials in Reverberant Rooms* (ASTM C423-77), (New York, 1977), p. 3.

5. M. J. Crocker and A. J. Price, *Noise and Noise Control* (Vol. 1), (Cleveland: CRC Press, 1975), p. 229.

6. *Ibid*.

7. American National Standards Institute, *op. cit*., p. 2.

8. Philip M. Morse, *Vibration and Sound*, (New York: McGraw-Hill, 1948), p. 33.

9. J. B. Moreland, "Controlling Industrial Noise by Means of Room Boundary Absorption," *Noise Control Engineering* (November-December 1976), p. 148.

10. Crocker and Price, *op. cit.*, p. 226.

11. J. C. Wyvill *et al.*, *loc. cit.*

12. William M. Ihde, "Final Report of Noise Investigations for the Poultry Industry," Report to the Poultry Industry Advisory Committee on Safety and Health, March 22, 1973, p. 3.

13. *Ibid*.

14. *Ibid*., p. 4.

15. "UHMW Plastic Outdoes Hardened Steel," *Design News*, March 10, 1980, p. 39.

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APPENDIX A

BURN TEST ON TEMPERKOTE®

I. INTRODUCTION

This report is a presentation of results of the tunnel test on a material submitted for testing by Georgia Institute of Technology.

The test was conducted in accordance with the provisions of the American Society for Testing and Materials Standard Method of Test E 84-80, "Surface Burning Characteristics of Building Materials," also known as the Steiner Tunnel Test. This method is similar to ANSI 2.5, NFPA No. 255, UBC No. 42-1, and UL No. 723.

This standard should be used to measure and describe the properties of materials in response to heat and flame under controlled laboratory conditions. It should not be used for description, appraisal, or regulation of the fire hazards of materials under actual fire conditions. There are no considerations made for results that may be obtained if the material being evaluated were tested in combination with other building materials.

The fire performance of any material in the light of present knowledge cannot be evaluated on the basis of any one test. The test result presented here applies only to the specimen tested and is not neces sarily indicative of apparent identical or similar materials. All test data are on file and are available for review by authorized persons.

II. PURPOSE

The tunnel test method is intended to compare the surface flame-spread and smoke developed measurements in relation to asbestos-cement board and select grade red oak flooring surfaces. A material is exposed to a flaming fire exposure adjusted to spread the flame along the entire length of a red oak specimen in 5½ minutes during a 10-minute test duration, while flamespread over its surface and density of the resulting smoke are measured and recorded. Test results are computed relative to the red oak specimen, which has a rating of 100, and the asbestos-cement board, which has a 0 rating, and are expressed as Flame Spread Index and Smoke Developed Index.

III. DESCRIPTION OF MATERIAL TESTED

CTC Test Number	-	1080-2538
Identification	-	FR/Film
Composition	-	Bonded Sailcloth/Polyester
Weight	-	2.5 ounces/yd^2

IV. PREPARATION AND CONDITIONING OF TEST SPECIMEN

The material being evaluated was adhered to 1/4 inch asbestos-cement flexboard with VPI #100 Epoxy Adhesive. The adhesive was applied to the board using a short-nap paint roller. The film was then placed into the adhesive and rubbed to remove entrapped air bubbles. The prepared specimen was then conditioned to equilibrium in an atmosphere maintained at 70°F and 50% relative humidity.

V. TEST PROCEDURE

The zero reference and other data critical to furnace operation were verified by conducting a 10-minute test using 1/4 inch asbestos-cement board on the day of the test. Periodic tests using NOFMA certified select grade red oak flooring provided data for the 100 reference. The material was then tested within parameters outlined in the standard test method procedure on January 27, 1981.

VI. TEST RESULTS

The test results, computed on the basis of observed flame front advance and the integrated area under the recorded curve of the smoke density apparatus, are presented in the following table. In recognition of possible variations in results due to limitations of the test method, the results are computed to the nearest number divisible by five.

Test Specimen	Flame Spread Index	Smoke Developed Index
asbestos-cement board	0	0
red oak flooring	100	100
(1080-2538) FR Film	10	25

Although not a requirement of ASTM E 84-80, Fuel Contributed may be reported for reference purposes. The Fuel Contributed is 0 for the material tested when computed in accordance with ASTM E 84-75.

The data for flamespread and smoke developed are shown as solid lines on the graph at the end of the report.

VII. OBSERVATIONS DURING TESTING

Ignition over the burners was noted at 0.86 minutes. The flame front advanced to 2.7 feet at 5.16 minutes with a maximum temperature recorded during the test of 576°F.

Slight dripping of the molten specimen occurred during the test. Blistering of the surface was noted after 0.5 minutes and continued throughout the test. There was no afterflame after the igniting burners were extinguished.

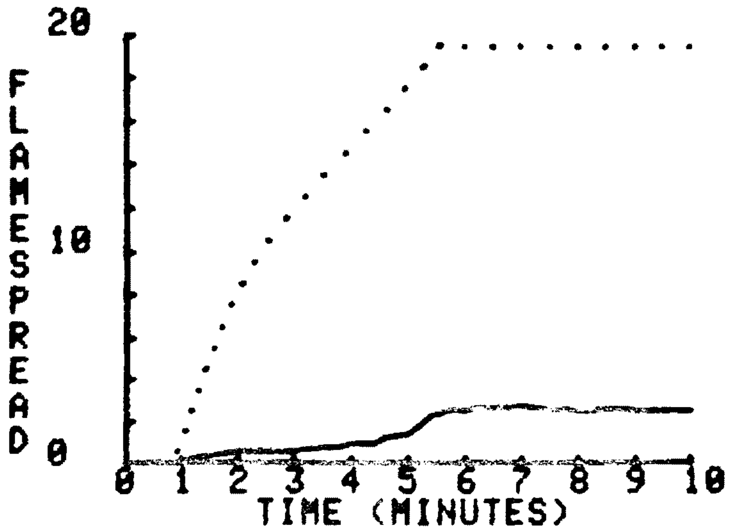
E 84-80 TUNNEL TEST DATA SHEET

CLIENT: GEORGIA INSTITUTE OF TECHNOLOGY
TEST NUMBER: 1080-2538
MATERIAL IDENTIFICATION: FR FILM
DATE: 27 JAN 81

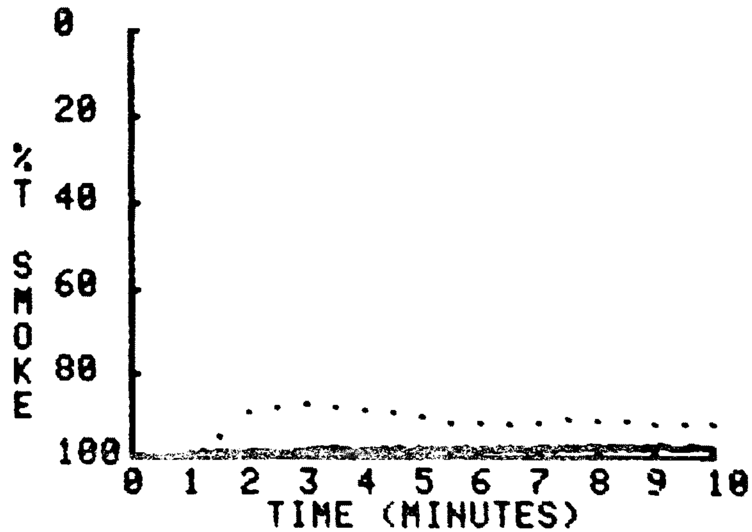
TEST RESULTS:

TIME TO IGNITION = 0.86 MINUTES
DISTANCE MAXIMUM SPREAD = 2.7 FEET
TIME TO MAXIMUM SPREAD = 5.16 MINUTES

FLAMESPREAD CLASSIFICATION	=	10
FUEL CONTRIBUTED	=	0
SMOKE DEVELOPED	=	25



dotted line = red oak



APPENDIX B

QUALIFYING A
REVERBERANT TEST ROOM

Qualifying a Reverberant Room

The reverberation chamber used for testing during this study was constructed of painted brick walls, a painted concrete floor, and a painted plywood ceiling. Per the suggestion of ANSI/ASTM C 423-77 and as more fully explained in Noise and Noise Control - Volume I, by M.J. Crocker and A. J. Price, nine stationary sound-reflective panels were hung at random orientations near the corner areas of the room. The reflective panels were constructed of 1/4" x 4' x2' masonite sheets which were slightly curved to further break up any room resonance modes. These reflective panels were used to increase the diffusion of the sound field in the chamber and reduce the spatial variance of the sound decay measurements. Practically speaking, this meant that fewer microphone positions were required to achieve a given measurement precision.

Qualification tests on the reverberation chamber were performed to insure its suitability for obtaining meaningful acoustic measurements. The ANSI/ASTM C 423-77 standard requires that the average absorption coefficient of the room surfaces at each frequency be less than .06 after a correction for air absorption has been made. Using the equipment arrangement shown in Figure 1-A and the microphone positions shown in Figure 1-B, nine decay curves were observed for each microphone position.

The room absorption, in sabins, was then calculated from the ANSI/ASTM C 423-77 formula:

$$A_1 = \frac{.9210XVX60}{C} X \frac{1}{T}$$

where

V = volume of room (ft^3)
T = average time required for the sound field to decay 60dB (seconds)
C = speed of sound (ft/sec)

The average absorption coefficient of the room surfaces (αR) was determined from the following formula:

$$\alpha R = \frac{A1}{S_R} - \frac{4mV}{S_R}$$

where

S_R = total surface are of room surfaces (ft^2)
m = air absorption factor (@ 2000HZ = .000625 ft^{-1} and @ 4000HZ = .001575 ft^{-1}.)

The results of the tests are shown in Table 1-A. From this table it can be seen that the average absorption coefficient of the room surfaces at each frequency is less than .06 as required by the standard.

Table 1-A

	500Hz	1000Hz	2000Hz	4000Hz
Spatially averaged T, in seconds	2.270	2.124	1.959	1.793
Spatial variance T, in % of T	1.40	2.50	2.19	2.52
Absorption, in sabine (A_1)	77.63	82.97	89.96	98.29
Average α_R for room surfaces	.0485	.0519	.0562	.0614
Average α_R corrected for air absorption	.0485	.0519	.0505	.0471

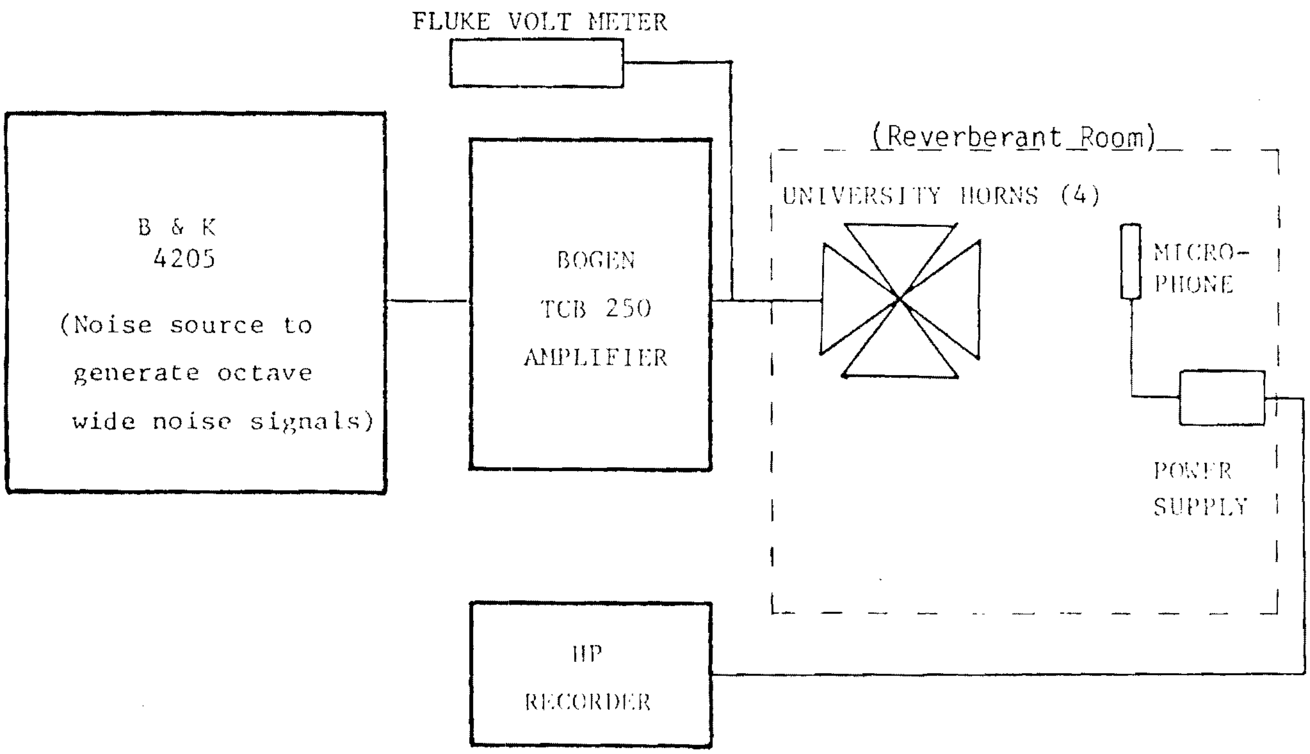
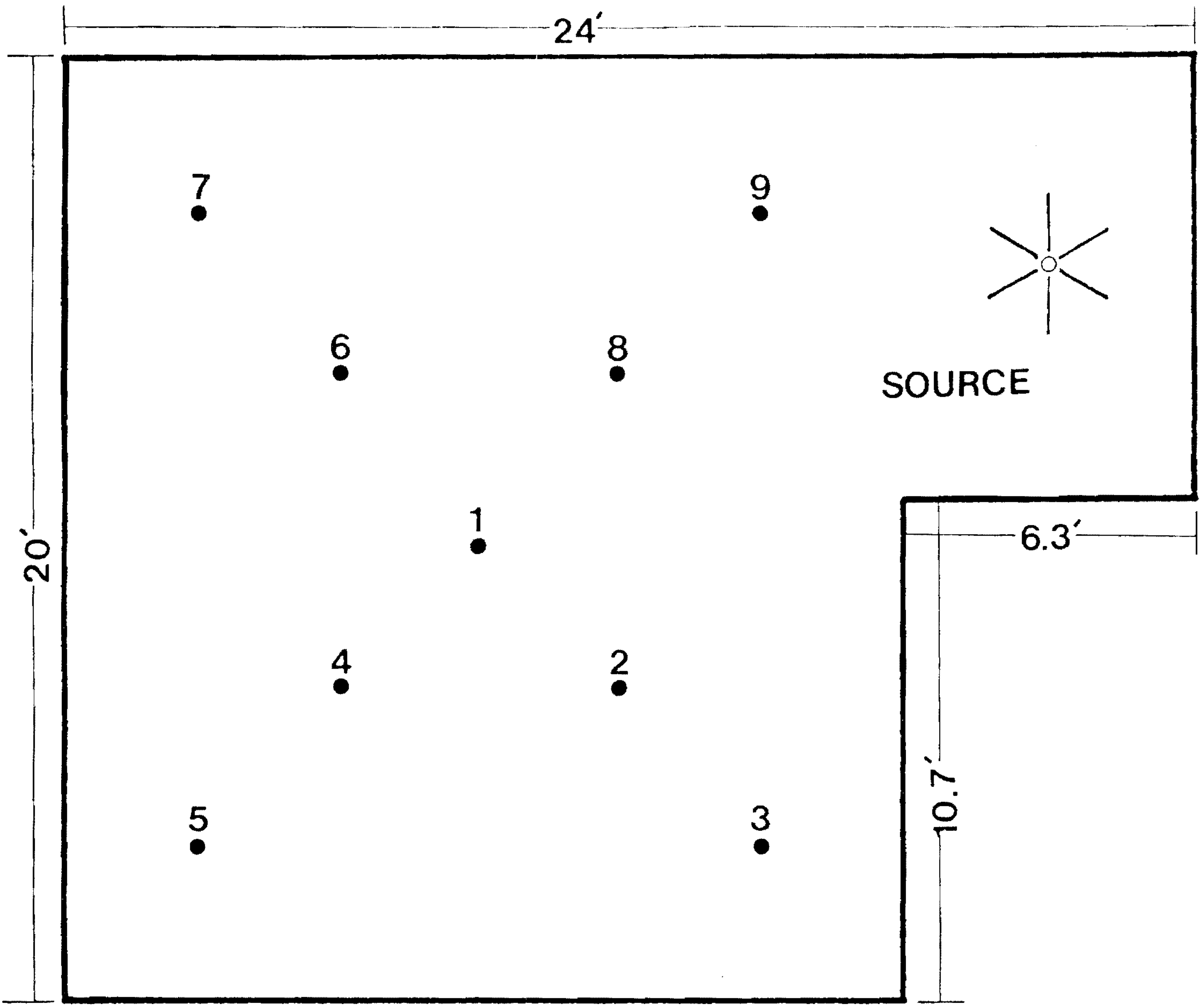


Figure 1-A - Equipment Arrangement



MICROPHONE HEIGHT - 3.4 ft

ROOM HEIGHT - 8.8 ft

Figure 1-B - Measurement Points in Reverberant Room

5-27-80

ERRATA SHEET

For

A STUDY OF POULTRY PROCESSING PLANT NOISE CHARACTERISTICS AND POTENTIAL NOISE CONTROL TECHNIQUES

1. On page 17 the equation should read:

$$\alpha\ SAB = \frac{4}{S}\left[\frac{1}{\text{antilog}\frac{Lp-Lw}{10} - \frac{Q\theta}{4\pi r}}\right]$$

2. On page 20, the equation should read:

$$\alpha\ SAB = \frac{.161\ V}{TS}$$

3. On page 20, in Table 7, the value of α SAB for the Central Soya plant should be .068

4. On page 20, the note at the bottom of the page should also contain the following comment:

   "The constant (.161) was obtained from the following calculation:

$$\left[\frac{4 \times 60}{4.34\ (343.5)}\right]$$

   The factor of 4 in this calculation represents the initial energy absorption of room surfaces in a diffuse reverberant field. Since non-diffuse conditions existed at Central Soya the factor of 4 was reduced to 2 (see page 17) and this constant similarly was halved on calculations for that plant."

5. On page 28, the note at the bottom of the page should read:

   "Reference 2, page 228. Note that due to the non-diffuse conditions at the Central Soya plant, a factor of 2 rather than 4 was used for it (see page 17)."

6. On page 9, the equation should read:

$$\overline{Lp} = 10 \log \left[\frac{\Sigma\ Si\ (\text{antilog}\ \frac{Lpi}{10})}{2\ \pi\ r^2}\right]$$

A STUDY OF POULTRY PROCESSING PLANT NOISE CHARACTERISTICS AND POTENTIAL NOISE CONTROL TECHNIQUES

<u>Final Report</u>

for

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ACKNOWLEDGMENT

While this effort reflects the concerns of an industry, its completion depended on the dedication of a few. We would especially like to thank Mr. Jim Burruss, Jr., of Tip Top Poultry and Mr. John Norris of Central Soya of Athens, Inc., for their coordination and assistance in gathering data; Mr. Chet Austin of Tip Top Poultry and Terry Walden of Central Soya Poultry for the cooperation and use of their respective plants in studying this problem; and Mr. Abit Massey of the Georgia Poultry Federation, without whom this study mght never have occurred.

THE GENERAL ENVIRONMENT

Introduction

In order to characterize the environment in a typical poultry processing plant, noise contours were developed for two representative plants: Central Soya of Athens, Inc., Athens, Georgia, and Tip Top Poultry, Inc., Marietta, Georgia. Contour information was restricted to the evisceration area of both plants because nearly 60 percent of all process employees are stationed in this area during a normal work shift.

Both plant evisceration areas were composed of tile walls, sheet metal ceilings, and concrete floors. Processing was performed in an assembly-line fashion in which the birds travel through the area on overhead shackles while personnel remain at fixed stations. Processing machinery was present throughout the area. Plant personnel worked in 8-hour shifts with 1/2 hour for lunch.

Data Acquistion

The measurement procedure used to gather contour data on the general environment consisted of taking readings in a grid pattern laid out for the evisceration area. Unfortunately the congestion of machinery and personnel sometimes prevented readings from being taken in certain areas of the plant. Figures 1 and 2 show the placement of microphones for measurements in the two plants.

To speed record taking, three microphones were attached, three feet apart, to an aluminum bar mounted on a tripod (see Figure 3). This allowed three measurements to be taken at one time. All measurements were tape recorded to allow level and frequency analysis in the laboratory. Additional readings, using a hand-held sound level meter, were taken in inaccessible areas. A complete list of the equipment used and the general arrangement of equipment for data gathering and analysis are presented in Appendix A.

Noise Contour Development

All observed noise levels were recorded, by grid position, on a plot of each plant. These levels were A-weighted and time-averaged over a two-minute interval. On each plot, lines of constant noise level were then drawn. The resulting contours are presented in Figures 4 and 5. Appendix B provides time histories and frequency analysis of selected data points observed throughout each plant.

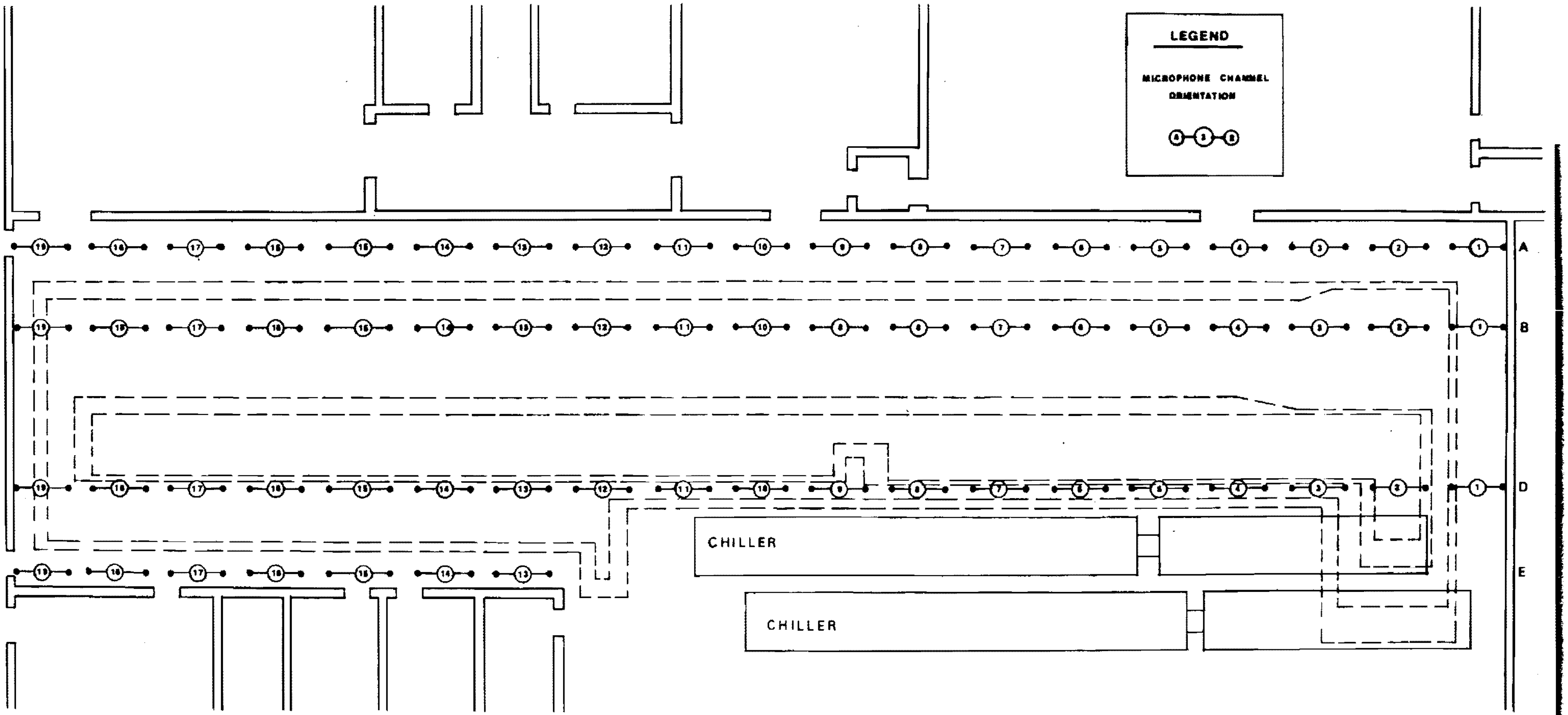


FIG. 1 LOCATION OF MEASUREMENT POINTS IN CENTRAL SOYA PLANT

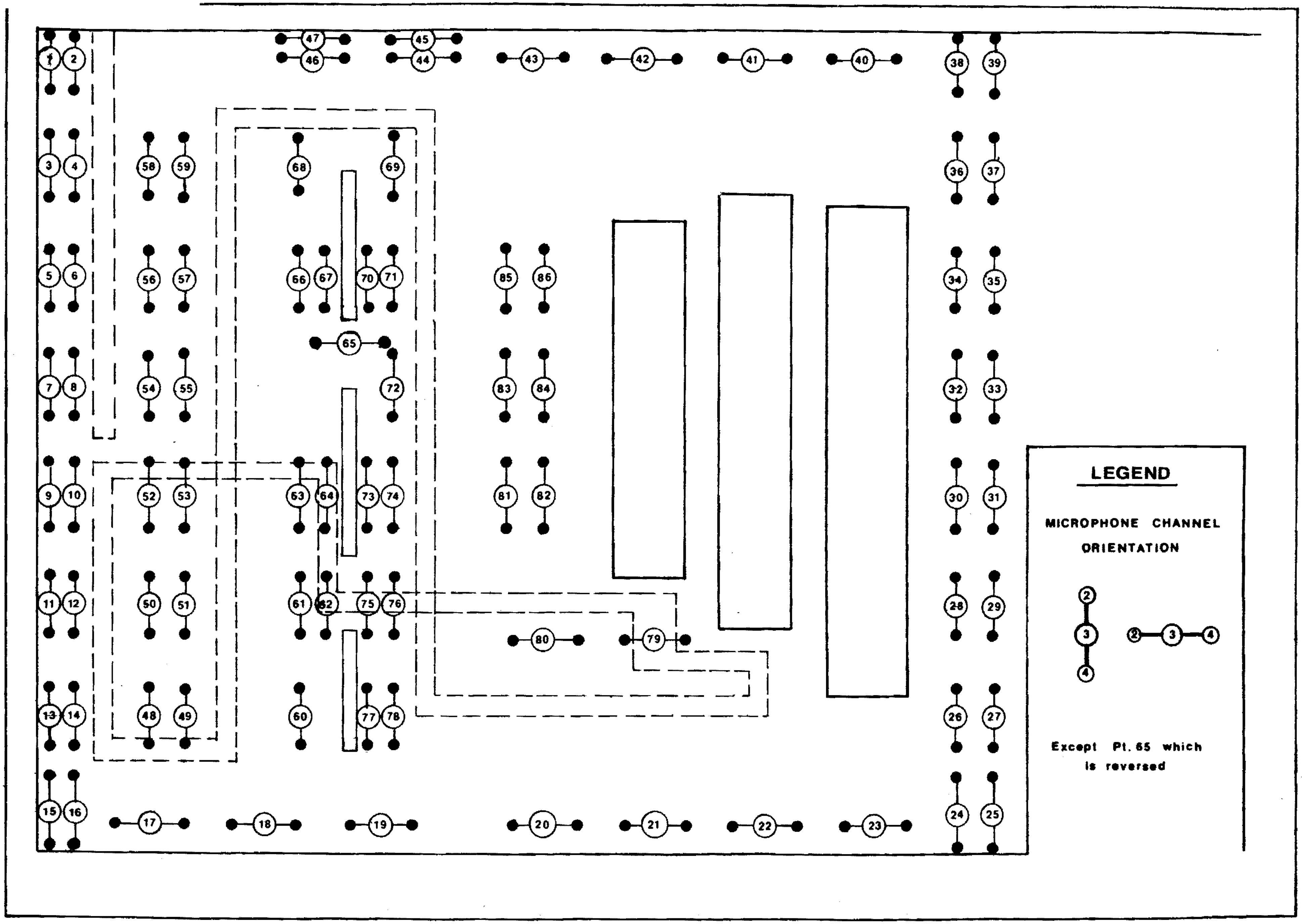


FIG. 2 LOCATION OF MEASUREMENT POINTS IN TIP TOP PLANT

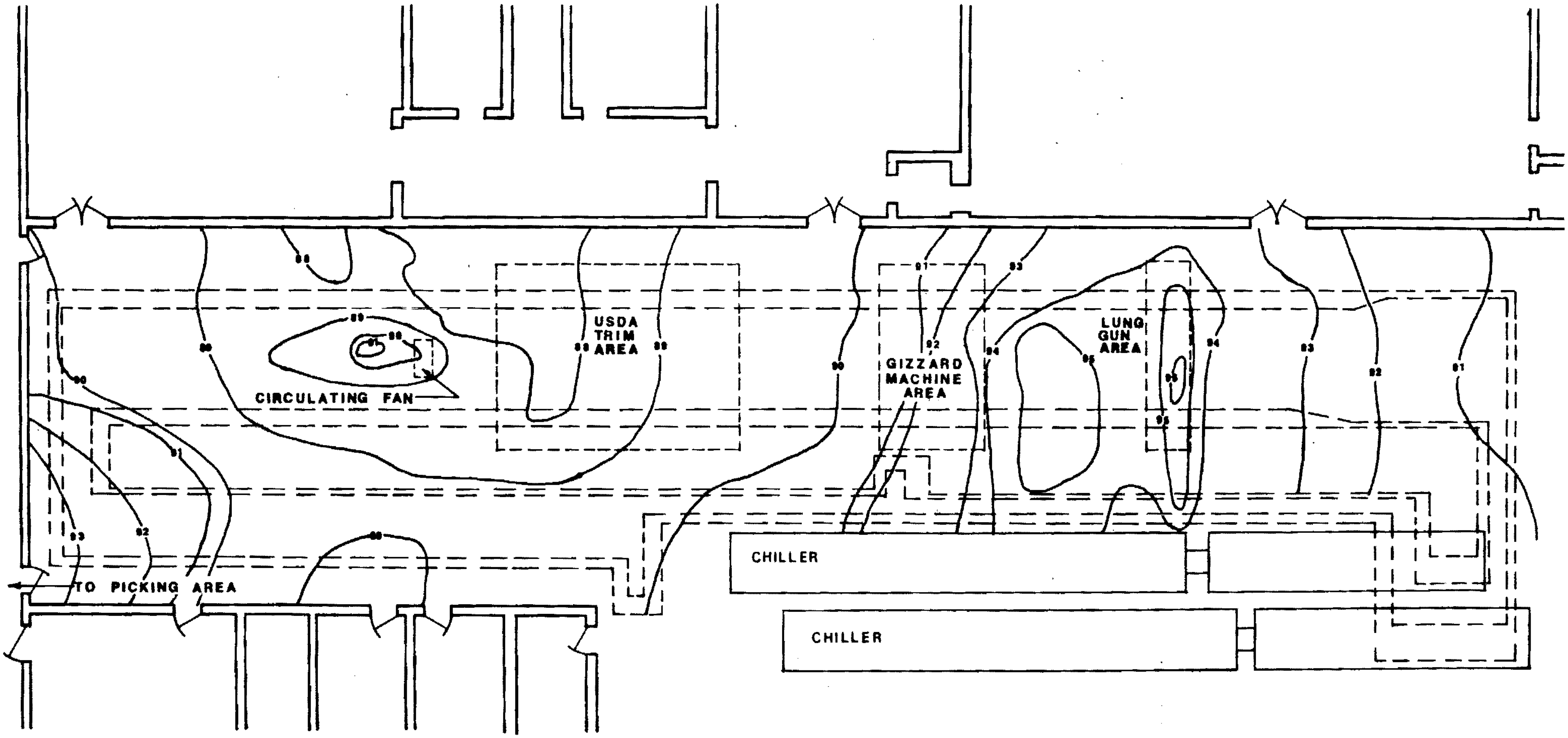


FIG. 4 NOISE CONTOURS FOR CENTRAL SOYA PLANT

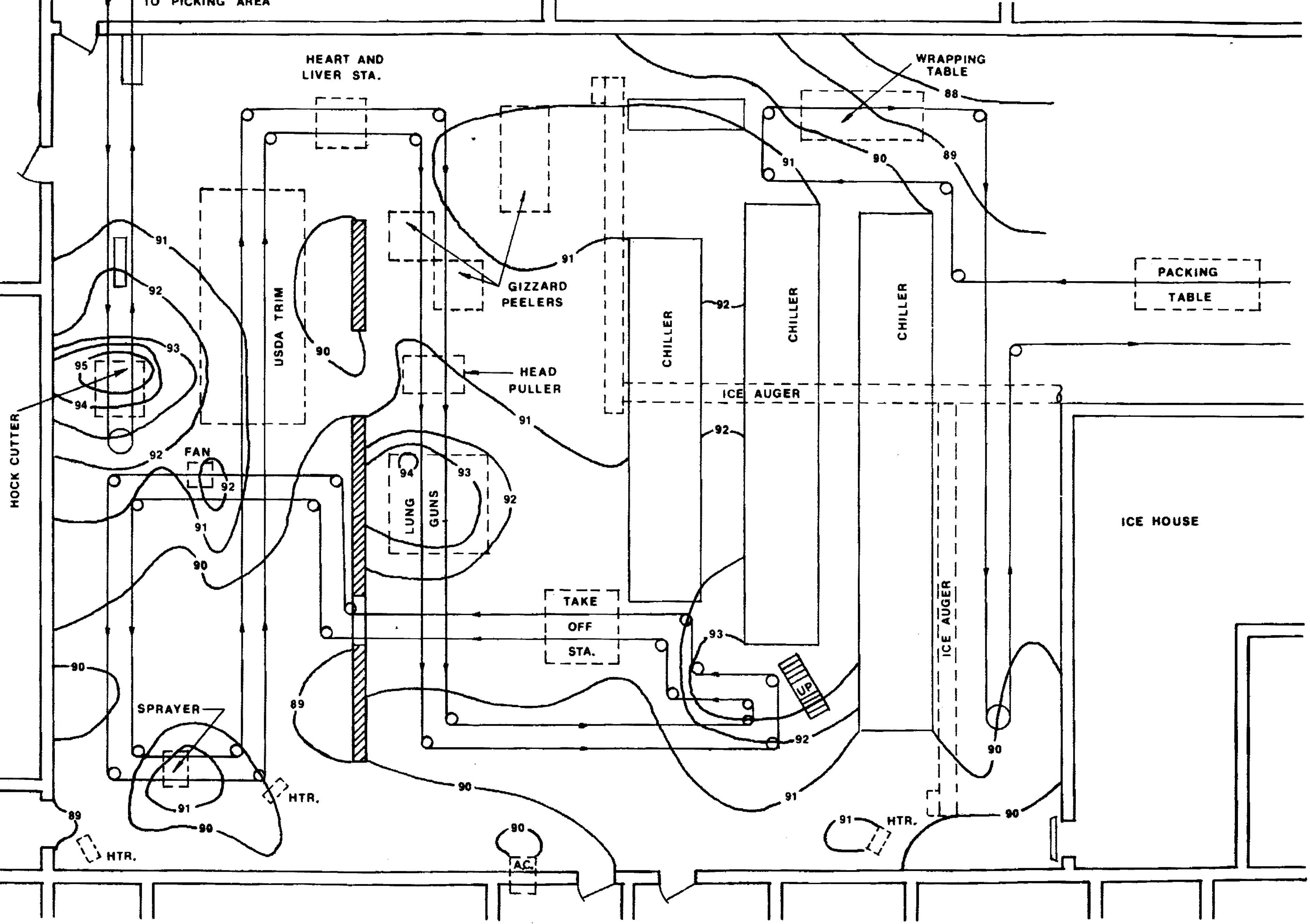


FIG. 5 NOISE CONTOURS FOR TIP TOP PLANT

The Noise Environment

The noise contours display specific information about the noise environment. For instance, in the plot for the Central Soya plant (Figure 4) there are only three areas of the plant where the noise contours converge. Within these areas the apparent sources of noise are the lung guns, a component of the chiller, a circulating fan, and a source from the picking area. Because there are two hock cutters immediately on the other side of the wall in the picking area which are exposed to the evisceration area through conveyor portals in the wall, it is probable that they are contributing substantially to the noise coming from the picking area.

The contours also provide information on the type of noise fields throughout the plant. Since the surfaces of the plant were composed of hard materials, it is probable that a reverberant noise field exists throughout much of the plant. Since this plant is irregular in shape, having one dimension many times that of another, it is expected that the reverberant noise field will not be uniform in level, but rather will decay in level with increasing distance from those sources contributing to it.* The noise field observed in Figure 4 does exhibit a continual but gradual decrease in levels well below the free field rate of 6 dB/doubling distance with increasing distance from the primary noise source areas. Consequently, it is probable that much of the noise field being observed is predominantly reverberant.

The noise contours for the Tip Top plant (Figure 5) indicate six areas of the plant where the noise contours converge. Within these areas, the apparent sources of noise are the lung guns, a component of the chiller, an air jet on the spray wash station, the hock cutter, the gizzard peelers, and an exhaust fan. Furthermore, this plant also appears to have much of its noise field dominated by reverberant noise, as evidenced by those areas of uniform level throughout much of the plant. Because this plant is more symmetrical than the other, a uniform reverberant field should be expected.

In comparing the two plants, only three identified sources are similar: the lung guns, chiller component, and hock cutter. Both plants also appear to have much of their noise field dominated by reverberant noise. Because the frequency spectra observed throughout both plants were extremely similar (see Appendix B), it would appear that both plants have a similar reverberant noise environment as defined by major contributing sources and absorption characteristics.

*Reference 1, page 4-13.

REVERBERATION EVALUATION

Introduction

Since the noise environment observed in both plants was thought to be substantially influenced by reverberation, a series of tests were performed to quantify the reverberant environment within each plant.

Direct/Reverberant Field Test

The first method used to evaluate reverberation entailed introducing a source of known response characteristics into each plant and observing the resultant noise field. Since any observed noise field is a combination of direct and reflected noise levels and the direct noise field of the source was known, the reverberant noise field was subsequently determined.

Qualifying the Source. The source selected for use in the direct/reverberant field tests was a 12-inch paper speaker, which received a white noise input signal boosted at maximum gain though a 30-watt amplifier. To observe the speaker's output response, it was placed in an anechoic chamber on the Georgia Tech campus. Figure 6 displays the measured directivity characteristics observed for one plane of the speaker's response. The frequency response data were obtained through octave band filtering of the measured broadband response characteristics.

To complete the characterization of the speaker's response, the sound power output related to each response curve was calculated. Since each response pattern was symmetrical about the perpendicular axis to the speaker, it was assumed that the three dimensional response pattern would essentially bear the same characteristics observed in Figure 6 in any plane rotated about this perpendicular axis. Using coordinates for the midpoint of 10 equilateral triangles forming the surface of a hemisphere around the front of the source, the average sound pressure level over the hemisphere was calculated, using the following formula:

$$\bar{Lp} = 10 \log \left[\frac{Si \left(\text{Antilog } \frac{Lpi}{10}\right)}{2 \pi r^2} \right]^*$$

Where

$\bar{Lp}$ = space averaged sound pressure level

Si = surface area of i^{th} segment

*Reference 2, page 152.

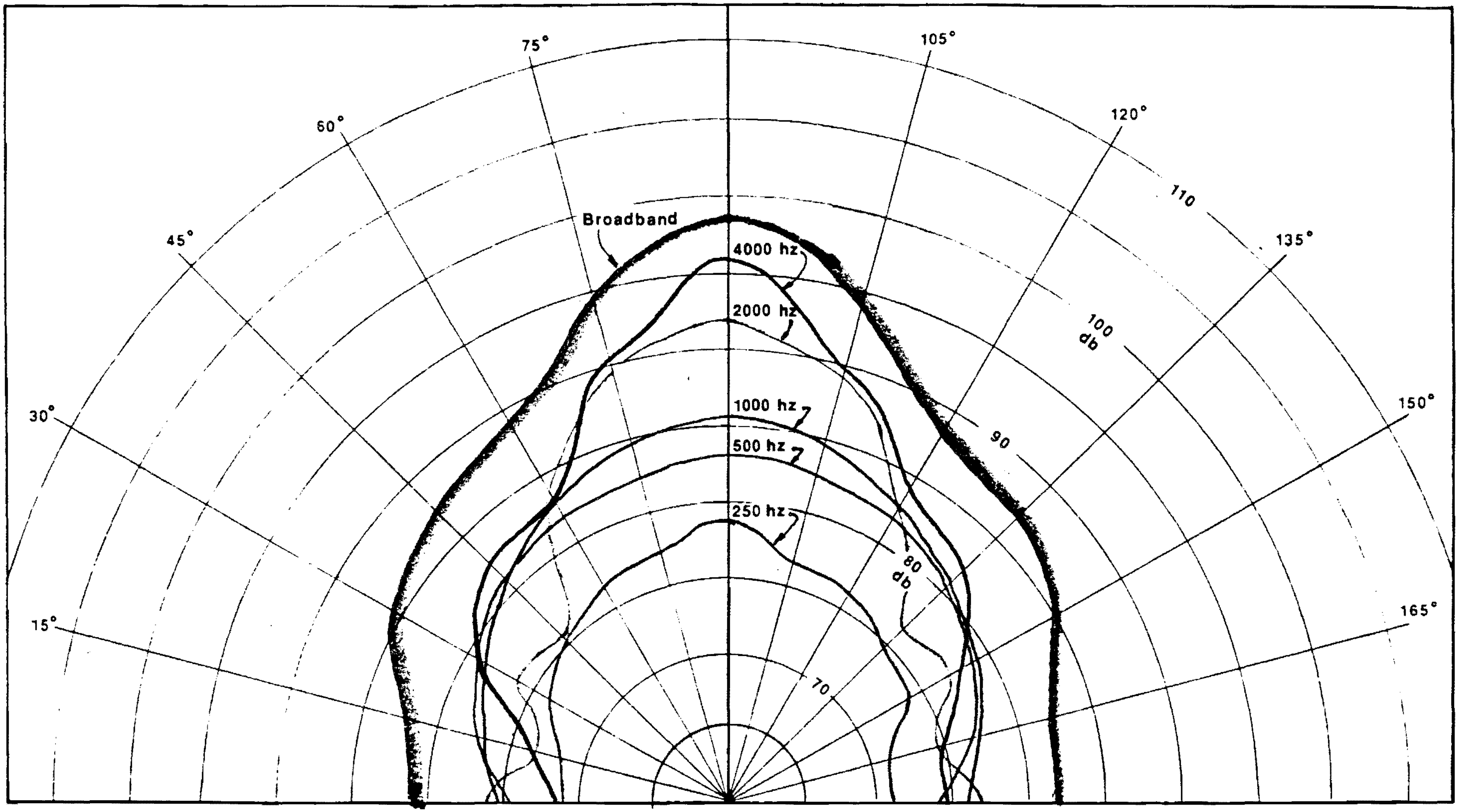


Fig. 6 Speaker Directivity Pattern *

* Observed at a radius of 1.83 meters from center of speaker

Lpi = sound pressure level at the midpoint of i^{th} segment

r = radius of hemisphere (meters)

Table I presents the ten values of Lpi used in the calculation. Si was 2.1 square meters for each triangle and r = 1.83 meters.

Table I

SOUND PRESSURE LEVELS USED TO CALCULATE SPEAKER SOUND POWER OUTPUT

Octave Band Lpi (dB)

Point (i)	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Broad-band
1	71.8	77.8	78.0	74.8	76.0	83.0
2	75.0	81.0	81.2	76.5	82.3	87.6
3	71.8	77.8	78.0	74.8	76.0	83.0
4	71.8	77.8	78.0	74.8	76.0	83.0
5	75.0	81.0	81.2	76.5	82.3	87.6
6	72.2	77.8	77.4	73.4	75.0	82.8
7	72.2	77.8	77.4	73.4	75.0	82.8
8	75.5	81.0	81.5	76.2	82.6	87.6
9	72.2	77.8	77.4	73.4	75.0	82.8
10	78.6	83.0	85.7	91.7	96.0	98.2

The sound power level was then determined using the following formula:

$$Lw = \bar{L}p + 20 \log r + 10 \log 2\pi \ ^*$$

where

Lw = sound power level

$\bar{L}p$ = space averaged sound pressure level

r = radius of hemisphere (meters)

Table 2 contains the calculated values of sound power output for the speaker.

*Reference 2, page 155.

Table 2

SOUND POWER LEVELS CALCULATED FOR TEST SPEAKER

Octave Band	Lw (dB)
250	87.50
500	92.99
1000	93.78
2000	95.72
4000	99.99
broadband	103.03

The Field Test. The source was then taken to each plant and positioned as shown in Figures 7 and 8. The speaker was placed on the floor of the plant facing the ceiling for both tests. With the source powered, measurements were taken at one-foot intervals on either side of the speaker in a single plane. Additional spot readings outside that plane were taken at several locations near the source to establish the level variation for the entire area surrounding the speaker (see Figures 7 and 8 for the location of all measurement points).

Figures 9 and 10 display the broadband levels observed in the measurement plane for each plant. It should be noted that an accident occurred during the Tip Top plant testing in which the speaker was sprayed with water prior to the measurements on the right hand side of the speaker. This appears to have reduced the response output of the speaker to some extent. Appendix C contains the results of octave band filtering of each of the measured values.

Since the measurement points intersected the directivity pattern of the speaker, the direct field levels were determined based on the following calculations:

$$Lp = Lp\theta - 20 \log \frac{r}{ro}$$

Where

Lp = direct field sound pressure level for the measurement point

Lpθ = sound pressure level obtained from the speaker directivity pattern for the angle corresponding to the measurement point.

r = distance from speaker to measurement point (meters)

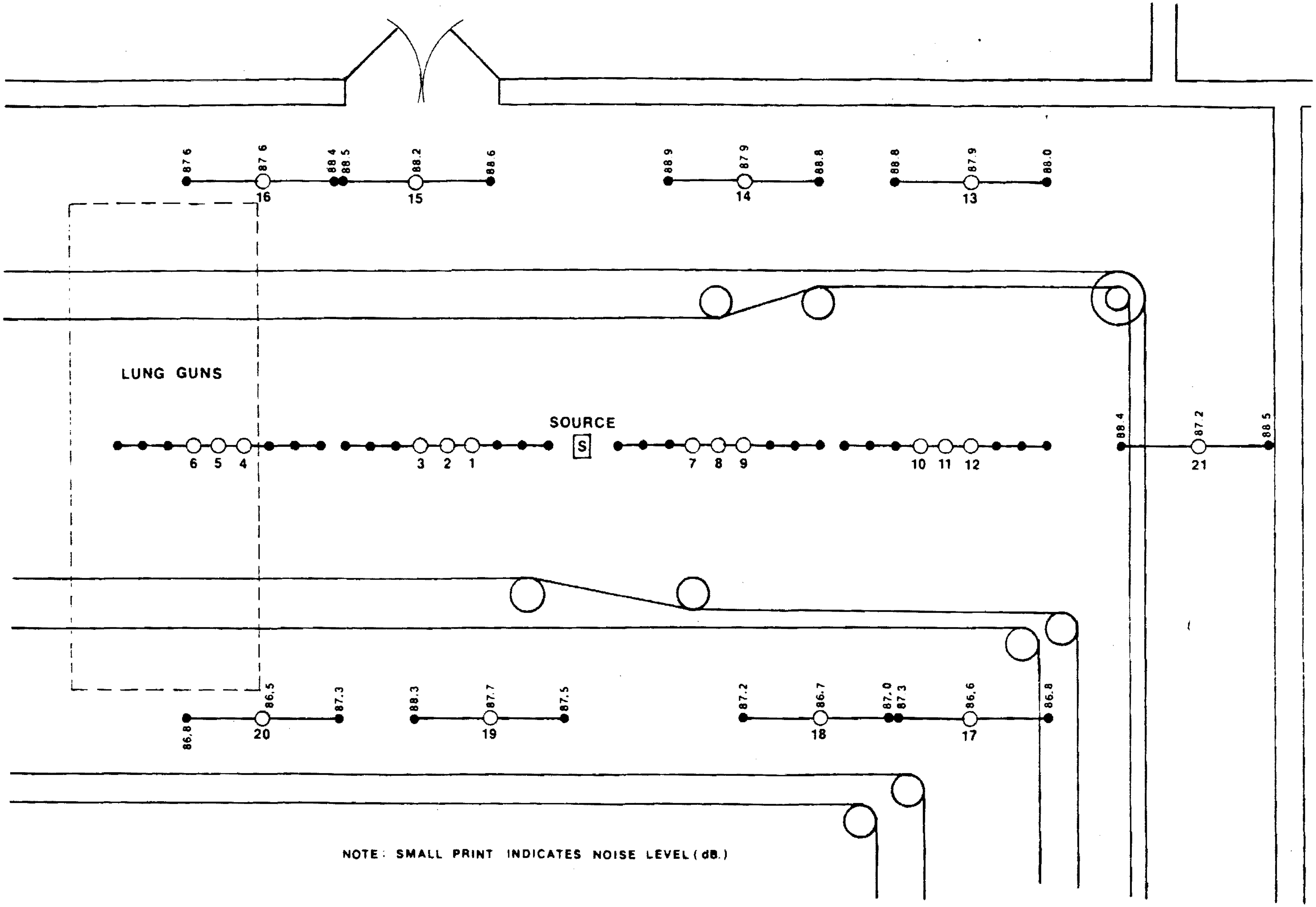


FIG. 7 LOCATION OF MEASUREMENT POINTS FOR
DIRECT-REVERBERANT TEST
CENTRAL SOYA PLANT

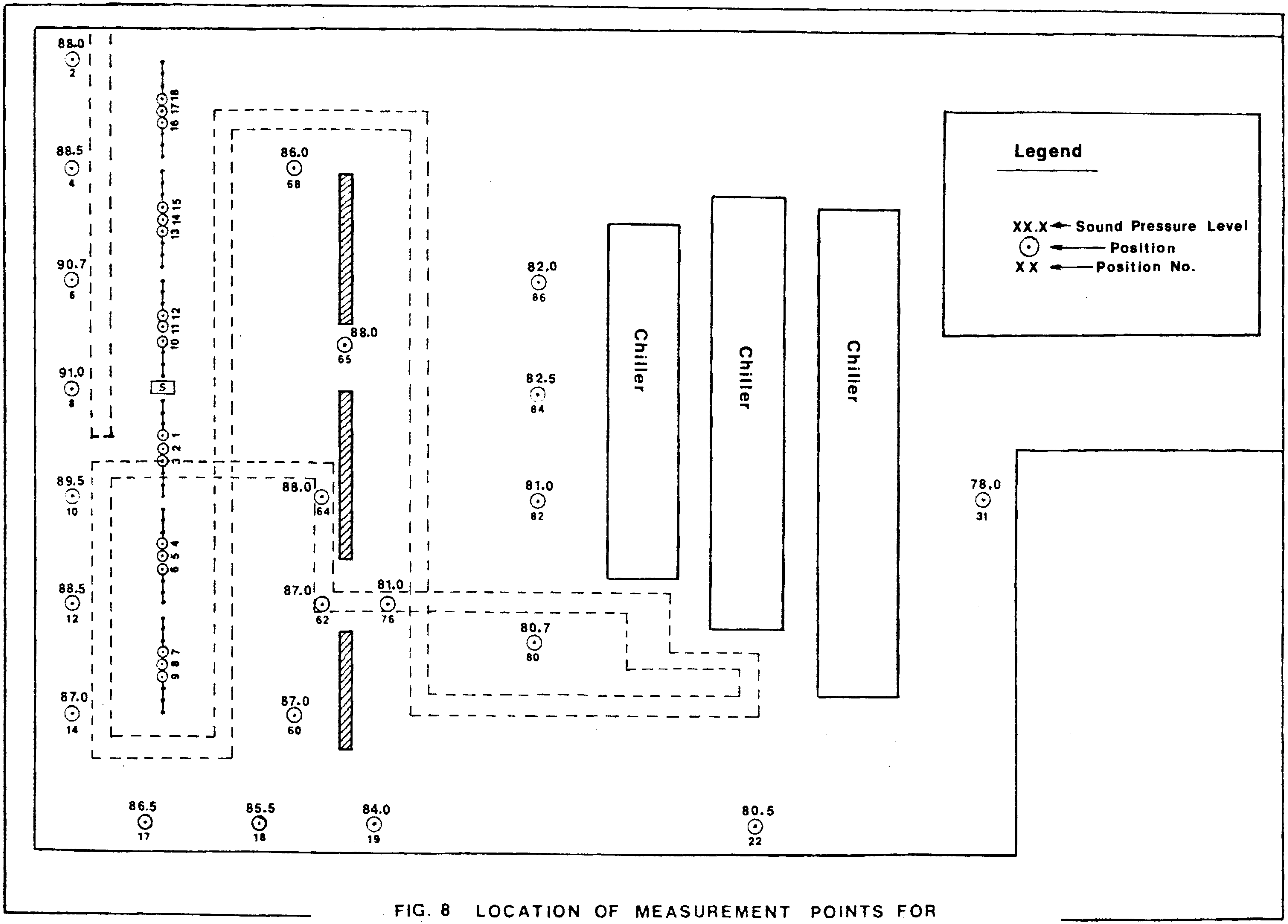


FIG. 8 LOCATION OF MEASUREMENT POINTS FOR DIRECT-REVERBERANT TEST TIP TOP PLANT

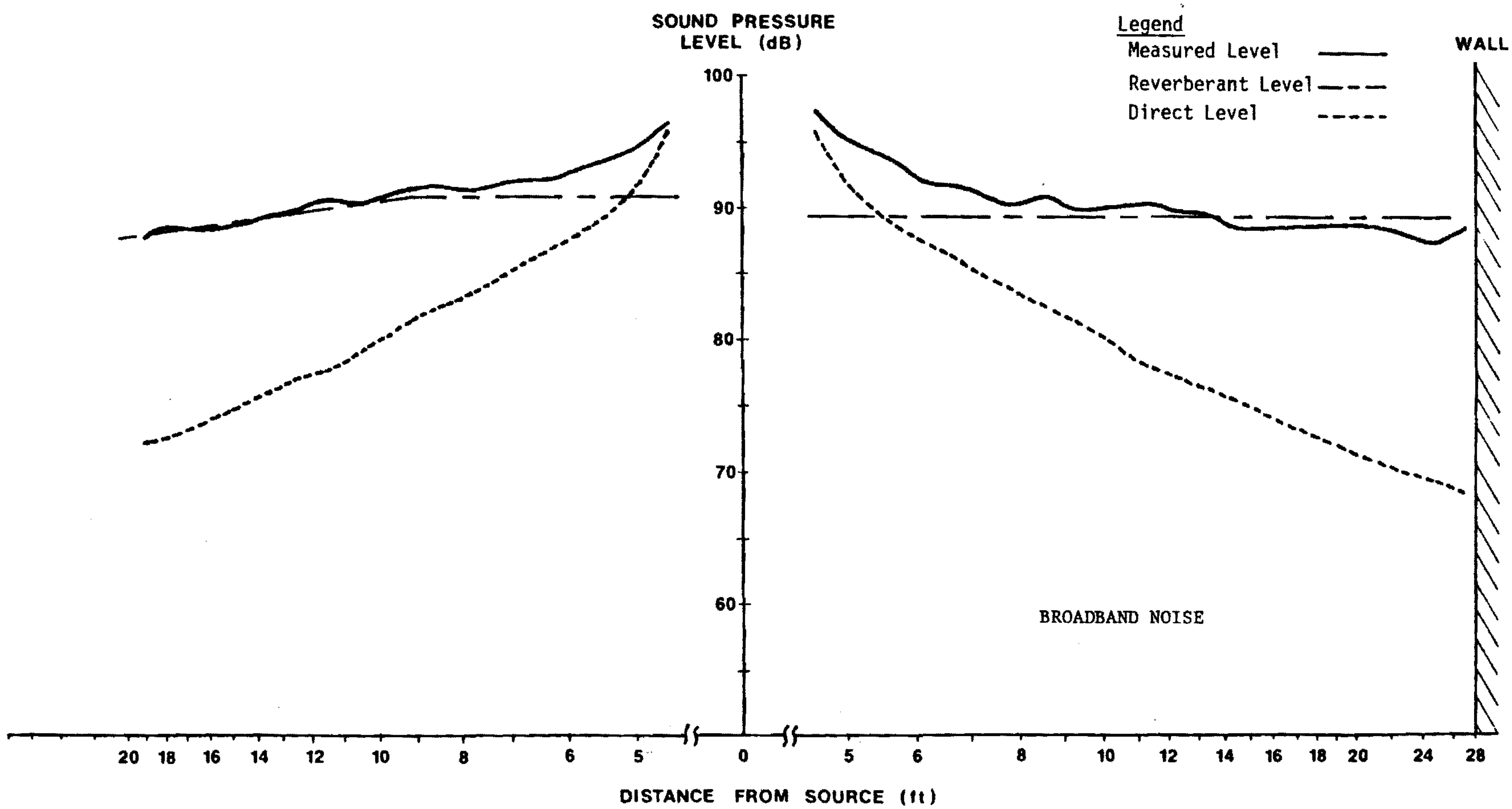


Fig. 9 Direct / Reverberant Noise Fields for Test Speaker

CENTRAL SOYA PLANT

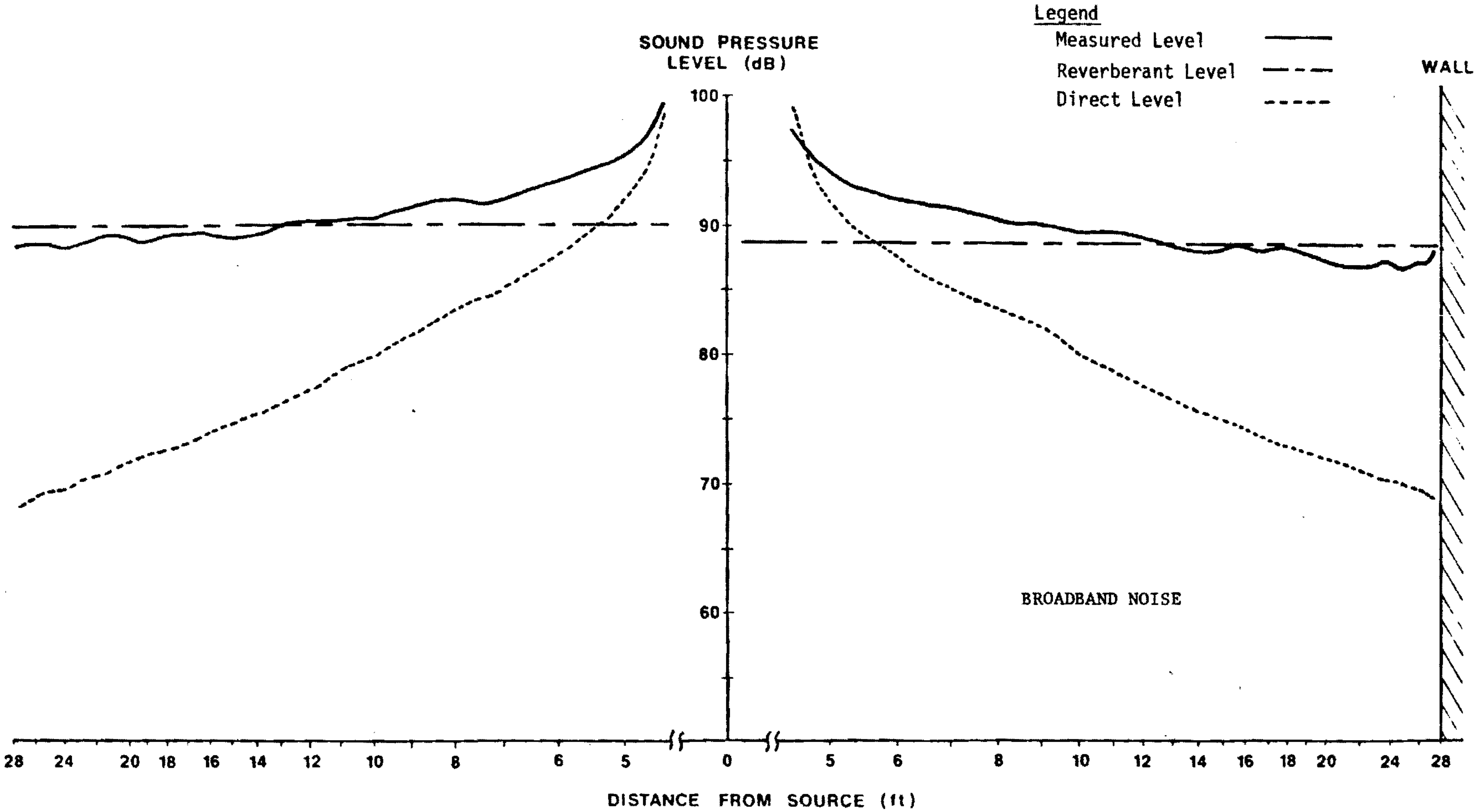


Fig. 10 Direct/Reverberant Noise Fields for Test Speaker
TIP TOP PLANT

ro = distance from speaker at which $L_{p\theta}$ was measured (meters)

From these figures, it is apparent that the overall level observed at distances beyond a few feet from the source are substantially influenced by the reverberant noise field. However, the reverberant field in the Central Soya plant does not appear to be uniform in level to the left of the speaker, but rather decays at a rate of approximately 3dB/doubling of distance from the source. This phenomenon has been observed by others for rooms in which one dimension is more than five times that of another.* For the Central Soya plant, the room length of 51.2 meters is nearly ten times the ceiling height of 5.5 meters. This is not true of the Tip Top plant where the largest dimension is roughly four times that of the smallest.

Defining the Reverberant Environment

The information obtained from the direct/reverberant field test was used to calculate the average surface absorption coefficient for each plant, using the following equation:

$$\alpha SAB = \frac{4}{S} \left[\text{antilog} \frac{Lp-Lw}{10} - \frac{Q\theta}{4 \pi r}\right]^{**}$$

Where

- αSAB = average sabine surface absorption coefficient
- S = surface area of the room (meters^2)
- Lp = measured sound pressure level (dB)
- Lw = calculated source sound power level (dB)
- $Q\theta$ = directivity factor of the source
- r = distance of measurement point from the source (meters)

In order to make this calculation, the sound pressure level measured at a distance of nine feet was used for the Tip Top plant. For the Central Soya plant, since the reverberant noise field was not uniform in level, the nine foot reading was attenuated at a rate of 3 dB/doubling of distance from the source to the picking room wall, and the resulting reverberant field levels were space averaged. The corresponding direct field contribution for the equation at this equivalent distance from the source was estimated to be small and was therefore neglected in the calculation for this plant. Table 3 presents the Lp values used in the calculation of surface absorption coefficient for each plant.

---

*Reference 1, page 4-13.

**Reference 2, page 228. Note that the factor of 4 was derived for diffuse conditions. Since non-diffuse conditions were observed in the Central Soya Plant, a factor of 2 was used for it.

Table 3

MEASURED SOUND PRESSURE LEVELS (dB)

Central Soya Plant[a/] Octave Band		Tip Top Plant[b/] Octave Band	
250 Hz	72.9	250 Hz	80.7
500 Hz	73.9	500 Hz	82.0
1000 Hz	76.9	1000 Hz	84.8
2000 Hz	77.9	2000 Hz	85.0
4000 Hz	76.9	4000 Hz	85.8
Broadband	83.9	Broadband	91.1

a/ Space averaged level for reverberant field.

b/ Measured at nine feet from the source.

Since the equation called for a measure of the directivity of the speaker to determine the direct field contribution, the following procedure was used to calculate this value. The sound pressure level at the measurement point which would be provided by a nondirectional source was calculated using the total sound power output of the source. This sound pressure level was then compared to the sound pressure level actually provided by the direct sound field at the measurement point. The ratio of the actual direct level to that level which would have been provided by a nondirectional source defined the directivity factor (Q_θ).* Table 4 presents calculated values for the Tip Top plant measurement point where the direct field entered into the calculation.

Table 4

SOURCE DIRECTIVITY FACTORS FOR TIP TOP MEASUREMENT POINT USED TO CALCULATE SURFACE ABSORPTION COEFFICIENTS

Octave Band	Q_θ
250 Hz	.879
500 Hz	.767
1000 Hz	.611
2000 Hz	.225
4000 Hz	.383
Broadband	.315

*Reference 2, page 159.

The final input to the calculation was the total surface area of the test room. For the Central Soya plant the test area was defined as the total evisceration area. However, for the Tip Top plant, the wall in the middle of the evisceration area provided an effective barrier for containing sound and, therefore, was used to define one wall of the test area. The total surface area of the Central Soya plant test area was calculated to be 1834 square meters and that for the Tip Top plant test area was calculated to be 627 square meters.

Using these inputs, the average surface absorption coefficient for each plant was calculated and is presented in Table 5.

Table 5

ESTIMATED SURFACE ABSORPTION COEFFICIENTS[a/]

Central Soya Plant		Tip Top Plant	
Octave Band	SAB	Octave Band	SAB
250 Hz	.031	250 Hz	.032
500 Hz	.088	500 Hz	.089
1000 Hz	.053	1000 Hz	.053
2000 Hz	.066	2000 Hz	.077
4000 Hz	.222	4000 Hz	.187
Broadband	.089	Broadband	.104

a/ Values include any contribution from atmospheric absorption as well.

It should be noted that there were some energy losses during testing attributable to openings in some of the surface boundaries defining the test areas. Furthermore, no allowance was made in the calculations for nonsurface absorption such as by air, a factor which had approximately a 15% impact on the surface absorption coefficient calculated for the 4000 Hz octave band. However, it is believed that the coefficients in Table 5 reasonably approximate the absorptive qualities of the test rooms.

Reverberant Field Decay Test

The second test used to confirm the values obtained from the direct/reverberant test consisted of exciting each plant with noise, then terminating the source of the noise and measuring the time needed for the noise level in the room to decay 60 decibels.

This decay time provided yet another measure of the average absorption coefficient for surfaces in the test area, through the following equation:

$$\alpha SAB = \frac{161V}{TS} *$$

Where

αSAB = Average sabine absorption coefficient

S = Total room surface area (meters^2)

V = Total room volume (meters^3)

T = Reverberation decay time (seconds)

Each plant was excited with noise from a 22 caliber, blank pistol for the test. This source provided sufficient sound power to thoroughly excite the test area but unfortunately provided only broadband comparative values. It was positioned at the location of the speaker in Figures 7 and 8 and was pointed toward the ceiling. Measurements were taken nine feet from the source. Figures 11 and 12 show the time history of the measured decay rate of the sound field in each plant following the pistol shot. The full 60 dB reverberant decay time was determined from these figures, using straight line extrapolation. These values were then inserted into the above equation, using the room statistics for each test area given in Table 6.

Table 6

ROOM STATISTICS FOR REVERBERANT FIELD DECAY TEST

Central Soya Plant	Tip Top Plant
V = 3110 m^3	V = 847 m^3
S = 1834 m^2	S = 627 m^2

With these inputs, the average broadband suface absorption coefficient for each plant was calculated and is presented in Table 7.

Table 7

ESTIMATED BROADBAND SURFACE ABSORPTION COEFFICIENT USING PISTOL SHOT

Central Soya Plant	α SAB = .136
Tip Top Plant	α SAB = .093

*This calculation also produced values which include any contribution from atmosphere absorption. Source: Reference 2, page 238.

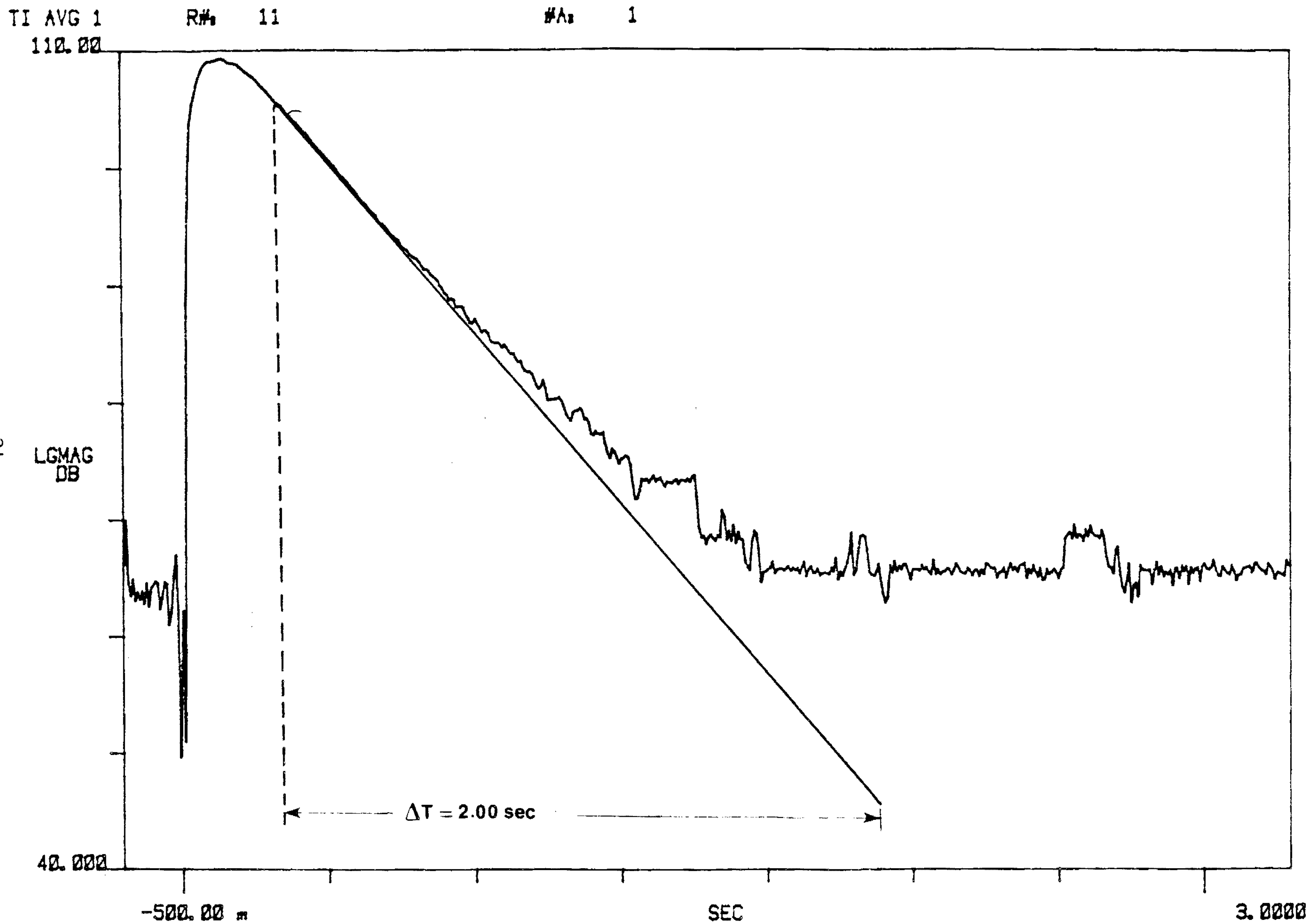


Fig. 11 Time History of Reverberant Noise Field Decay

CENTRAL SOYA PLANT

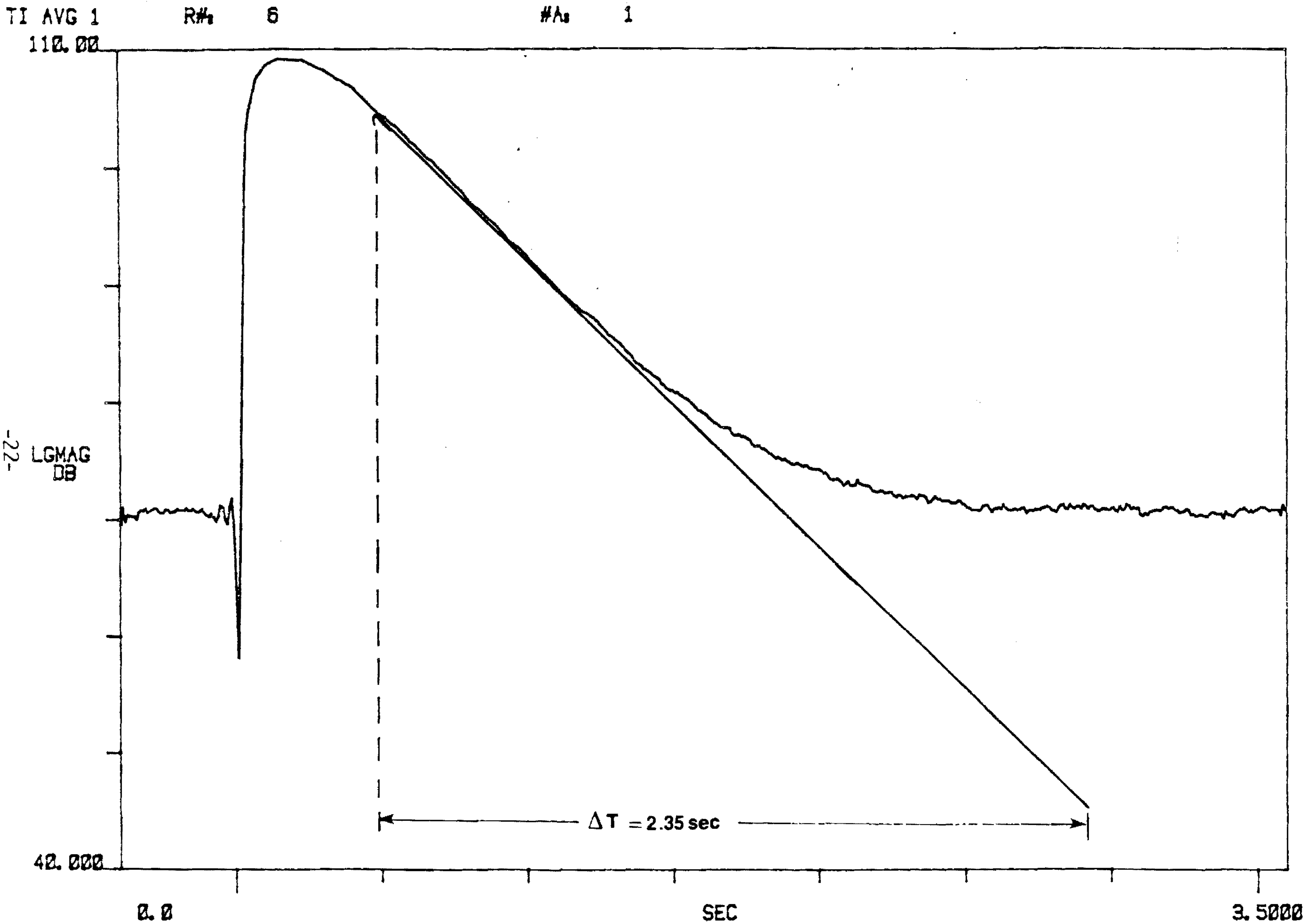


Fig. 12 Time History of Reverberant Noise Field Decay
TIP TOP PLANT

The values in Table 7 are reasonably close to the broadband values shown in Table 5, thereby confirming these values. Due to the non-diffuse conditions existing in the Central Soya Plant, the decay curve for it seems to exhibit some non-linearity which was not accounted for in the straight-line extrapolation. This may explain part of the difference between the abosorption coefficient determined for it by this method and that determined by the direct/reverberant field method.

SOURCE EVALUATION

Introduction

Observations made earlier of the general environment indicated only a few major sources were distinguishable above the general din. In order to complete an assessment of the poultry noise problem, a study of these noise sources was performed.

Sound Power Estimates

Using the information contained in the contours of Figures 4 and 5, an estimate was made of the A-weighted sound power output of all distinguishable noise sources. The technique used involved observing that contour line which was within 2 to 6 feet of the apparent acoustical center of the source, calculating the area encircled by the contour line, determining the radius of a circle with an equivalent area to that enclosed by the contour, and assuming a symmetrical hemispherical contour in the vertical plane. These inputs were then applied to the following equation:

$$Lw = \bar{L}pH + 20 \log r + 10 \log 2\pi^{*}$$

Where

Lw = estimated A weighted sound power output

$\bar{L}pH$ = A-weighted sound pressure level of the observed contour line

r = radius of circle with equivalent area to that encircled by the contour line.

The selection of 2 to 6 feet was made because contour lines closer than 2 feet typically will be in the near field of the source, while those farther than 6 feet typically will reflect significant reverberant noise field contributions. Unfortunately, certain contour lines within these distance limits were still unduly influenced by contributions from either the reverberant environment or another nearby source. Consequently, any source whose contour pattern appeared to be significantly influenced by activities other than from the direct noise field of that source was listed as having a sound power output which was not determinable from the contour data.

Applying the information contained in the contour plots, the values in Table 8 were developed.

---

*Reference 2, page 155.

Table 8

ESTIMATED SOUND POWER OUTPUTS OF MAJOR SOURCES

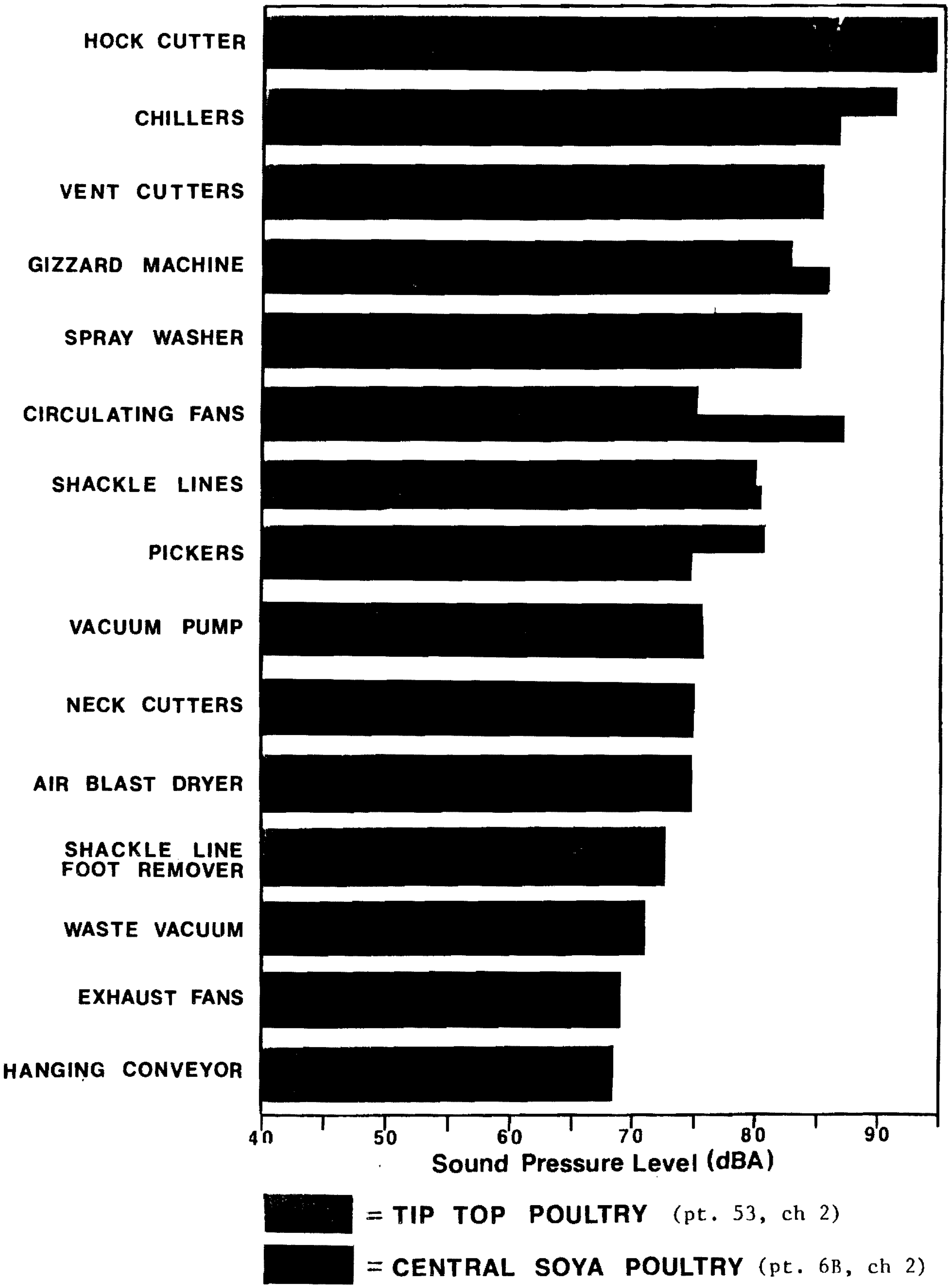
Central Soya Plant		Tip Top Plant	
Lung Guns	108.2dBA	Lung Guns	102.7dBA
Chillers	not determinable	Chillers	102.7dBA
Fan	94.7dBA	Exhaust Fan	not determinable
Hock Cutters	103.9dBA	Hock Cutters	100.2dBA
		Drying Air	94.7dBA
		Gizzard Peeler	not determinable
Total	109.7dBA	Total	107.05dBA

From these estimates, it appears that the top three noise sources in both plants are the lung guns, a chiller component, and hock cutters. The data in the Central Soya plant, however, need qualifying. The chiller component was positioned so that the lung guns masked much of its observable contribution. However, it is apparent in Figure 4 that a large contribution is coming from the chiller area as noted by the presence of a local increase in sound pressure level in the area immediately between the lung guns and the gizzard peelers. Since the gizzard peelers are apparently not producing that intense a signal, only an item on the chillers appears capable of being the second source. Also the hock cutters in the Central Soya plant were positioned in the picking room such that the combination of their outputs and the reverberant field associated with the pickers could have resulted in observed sound pressure levels more intense than those associated with the direct field of just the hock cutters. These two points are made so that the reader can apply caution when liberally interpreting the benefits of source sound power reduction in the Central Soya plant.

Source Contribution Assessment

As a means of evaluating the contribution of all sources to a locally observed sound pressure level in the noise contour of Figures 4 and 5, a microphone was located at point 6B, channel 2, in the Central Soya plant (see Figure 1) and point 53, channel 2, in the Tip Top plant (see Figure 2). With all sources turned off in each plant, individual sources were turned on and off one at a time. Figure 13 presents the A-weighted sound pressure levels observed for each source tested in each plant. Appendix D provides frequency contribution information about each source in addition to a comparison of the combined frequency spectra of all sources tested to that observed at

Fig. 13 Source Contribution A-Weighted Sound Pressure Level at a Single Point in Each Plant



that point in each plant under normal operating conditions. It should be noted at this time that a few major sources were not operated in each plant because of difficulties encountered at the time of testing.

These findings provide information which must be interpreted cautiously. For instance, the measurement point was close to some sources and far away from others implying care be taken in comparing source levels. Also, many of the sources were operated under conditions not typical to normal usage, such as the chillers, which were operated without ice or water, and the neck cutter, which lacked animal fat from the chickens to prevent an uncharacteristic whine.

This analysis, however, does provide some insight into the hurdles which can arise from keying reduction efforts on only one source, by displaying how the contributions of other sources can become significant even though they are currently masked during normal conditions.

THE PROBLEM IDENTIFIED

Using the data from the previous sections, an analysis was performed to determine if essentially all of the noise levels currently observed in each plant were directly and indirectly the result of only the few "major" sources identified. Since the direct effects were observable in the contour plot, only the indirect effects or the contribution of these sources to the reverberant field needed analysis. To perform the analysis, the following equation was used:

$$Lpr = Lw + 10 \log \left(\frac{4}{S \, \alpha SAB} \right)^{*}$$

Where

Lpr = sound pressure level of the reverberant field

Lw = sound power output of major noise sources

S = surface area of evisceration area

αSAB = average broadband surface absorption coefficient

In this calculation, the values of α SAB utilized were those for broadband noise from Table 5. Using the surface area values contained in Table 9, the calculations were performed.

Table 9

SURFACE AREAS ESTIMATED FOR TOTAL EVISCERATION AREA IN EACH PLANT

Central Soya Plant	Tip Top Plant
1834 m^2	1669 m^2

The calculations yielded the following results:

Central Soya Plant

Lpr = 90.6 dBA

Tip Top Plant

Lpr = 90.7dBA

These values were reasonably close to the A-weighted sound pressure levels observed in the reverberant field of each plant per Figures 4 and 5:

*Reference 2, page 228. Note that due to non-diffuse conditions, a factor of 2 rather than 4 was used (see page 17).

Central Soya Plant

Lpr = 90.4dBA (space averaged)

Tip Top Plant

Lpr = Between 90 and 91dBA

Therefore, it appears that the reverberant noise field in these plants is currently powered by only those few "major" noise sources identified in the contour plots.

As a result of these findings, it now becomes evident why there have been many failures in reducing overall plant noise levels. Since most efforts are focused on source quieting, only those efforts which are focused on a major source will be successful in significantly reducing noise levels, and even then the success will depend on the presence or absence of other intense noise sources. Clearly, therefore, a plant must know its major noise sources if source quieting is to be successful. On the other hand, increasing surface absorption in the plant will almost assuredly reduce noise levels in much of the plant through its impact on the reverberant noise field. But, even this solution will be limited in its overall effect by the nature of each plant's reverberant noise field and the distribution and total sound power output of sources throughout the plant.

POTENTIAL SOLUTIONS

In discussing potential solutions to the poultry processing noise problem, it should be stressed that each plant will have differing circumstances which impact their ability to effectively implement certain changes. Nonetheless, these solutions appear practical on the whole for the industry.

Source Solutions

There has been activity in the area of noise reductions at the source. Some actions have deliberately focused on noise reduction, others on productivity improvement. Here is an overview of possible solutions to reducing noise from sources in a poultry processing plant.

Lung gun noise is currently being alleviated in many plants with the use of drawing machines which also pull out lungs. Drawing machines are being widely used in broiler plants which process a relatively uniform bird size. Unfortunately, plants which process hens or a wide range of bird sizes cannot use the existing drawing machines. For these plants, there have also been studies* to baffle or shield noise from the body cavity during the lung gun operation. However, these baffled lung guns have not been used extensively because the baffles are difficult to keep clean and obstruct the view of the operator.

Efforts to quiet hock cutters have been restricted largely to isolating the machine from personnel. There are several designs of hock cutter available, but none are particularly quiet.

Chiller noise can typically be alleviated through vibration dampening. Impact noise from ice drop-off stations is often observable on ice slush chillers. This noise can be reduced through dampening of metallic surfaces in the ice delivery system, as well as by reducing the ice load through energy conservation efforts to jacket the chiller trough. Refrigerated chillers can further eliminate the need for ice altogether.

Lastly, the importance of regular and proper machinery maintenance cannot be overemphasized as a means of controlling source noise. Worn bearings, misaligned drive shafts, and improperly lubricated fittings can all turn a normally quiet machine into an unusually loud machine.

---

*References 4 and 5.

Room Acoustic Solutions

There has also been activity in the area of increasing the absorptive qualities of a plant.

For the most part, panels made of absorbant material, such as fiberglass or foam, have been developed.* These panels have been covered with plastic films to meet USDA requirements for use in food plants. But difficulties have occurred in the plastic film withstanding the harsh elements of most plants. Perhaps the single biggest problem is shearing of the plastic cover which renders the panel unacceptable for continued use by USDA requirements.

If a design could be developed which utilized a screen to protect the plastic film while remaining transparent to noise or if a cover could be designed of a film tough enough to withstand cleaning and other routine operations, then absorbing panels would clearly help in reducing the transmission of sound in the reverberant noise field.

*References 4 and 5.

CONCLUSION

In general, the poultry processing noise problem is the result of loud sources and reflective surfaces. Within the evisceration area, where nearly 60% of all processing personnel are stationed, it can be concluded that only a few major sources (lung guns, a chiller component, and hock cutters) are responsible for essentially all direct and reverberant sound pressure levels currently observed during normal operations. Consequently, any efforts to reduce the noise problem must first address the sound power output of these sources and/or the absorptive qualities of the room.

Reducing the sound power of major sources can be accomplished either by redesign or source isolation. Studies of redesign have been performed on many items.* The lung guns in particular have had several redesigns proposed. The thrust of these designs has been to shield the sound originating in the body cavity from the suction process. However, these baffled lung guns have not been used extensively because the baffles are difficult to keep clean and obstruct the view of the operator.

Isolation of a source has also been performed on such items as pickers and in some instances hock cutters. However, as was shown in the Central Soya plant, not all isolation mediums have been totally effective.

For either source quieting or isolation to work, the technique will need to be simple and inexpensive and not substantially change the manner in which processing is currently done. Yet, for every decibel of total sound power reduction achieved, a corresponding decibel reduction in observed sound pressure level will be noticed, perhaps not uniformly, but on a space average throughout the plant. The key words here, however, are <u>total sound power reduction</u>. It must be remembered that other sources, which are currently unidentifiable, will begin to contribute significantly to total sound power as the levels of the current major sources are reduced. This implies that a compounding problem exists as lower and lower sound pressure levels are sought.

Increasing the absorptive qualities of the plant is also an area where some studies have been performed.** However, difficulties have arisen with both cost and durability. Still, there is optimism that a design exists which will meet all criteria. Treatment of only the ceiling areas of the two plants studied could help reduce overall sound pressure

*Reference 3.

**Reference 4 and 5.

levels approximately 5dB on average. The ceiling of the Central Soya plant contains approximatley 35% of the total surface area and of the Tip Top plant contains approximately 30% of the total surface area.

However, room absorption is also limited in the total sound pressure level reduction achievable. This is because as reverberant levels decline, direct field levels from more obscure sources will begin to control local sound pressure levels. By reducing the intensity of the reverberant field, however, the potential for the current problem of the exposure by processing personnel being controlled by one or two noise sources will be reduced, which will provide both long-lasting and far-reaching benefits.

REFERENCES

1. Handbook of Noise Control, Cyril M. Harris, 2nd Edition, McGraw-Hill, 1979

2. Noise and Vibration Control, Leo Beranek, McGraw-Hill, 1971

3. Summary of Noise Investigations for the Poultry Industry, W. M. Idhe, Report to the Poultry Industry Advisory Committee on Safety and Health, 1973

4. Materials for Noise Reduction in Food Processing Environments, S. A. Waggoner, J. F. Shackelford, F. F. Robbins, Jr., and T. H. Burkhardt, APPLIED ACOUSTICS, Vol. 11, 1972

5. Clean and Quiet Baffles and Panels, Owens Corning Fiberglass, Publication 1-SD-9224, 1979

EQUIPMENT USED FOR DATA ACQUISTION & ANALYSIS

Microphones: B+K Precision condenser-type acoustic transducers were used for all sound pressure level measurements.

Channel	Cartridge Type	Serial No.	Preamp. Type	Serial No.
1	4165	775332	2619	748130
2	4165	750790	2619	748145
3	4165	708529	2619	748110
4	4165	732743	2619	748132

Power Supply to Pre-Amplifier: Two type 2807 B+K twin channel power supplies.
Tape Recorder: Hewlett-Packard type 3964A Instrumentation Tape Recorder.

Power Source for Field Use: All microphones and tape recorders were operated from a TRIPP-LITE 400-watt inverter that was powered from a 12-volt automobile battery. The use of the inverter was necessary to make the data-gathering equipment more portable and to reduce the problems encountered with voltage fluctuations and power line noise that were present in some of the plants where we acquired data.

Sound Source: The source for the reverberation time was a .22 caliber blank pistol.

The source for the direct field/reverberent field comparison was a B+K type 4205 white noise generator connected to a Bogen 30-watt power amplifier. The power amplifier drove a 12-inch paper loudspeaker that was mounted in an 18-inch square wooden box.

Analyzer: All time records and spectra were computed on a Hewlett-Packard type 5420A digital signal analyzer. The results were plotted with a Hewlett-Packard type 8972 four-color graphics plotter.

RMS Averages: All root-mean-square averages were determined with a fluke type 8010 digital multimeter.

A-Weighting: B+K Type 2203 Precision sound level meter was used to A-weight all readings. This meter was also used to take auxiliary readings in the plants.

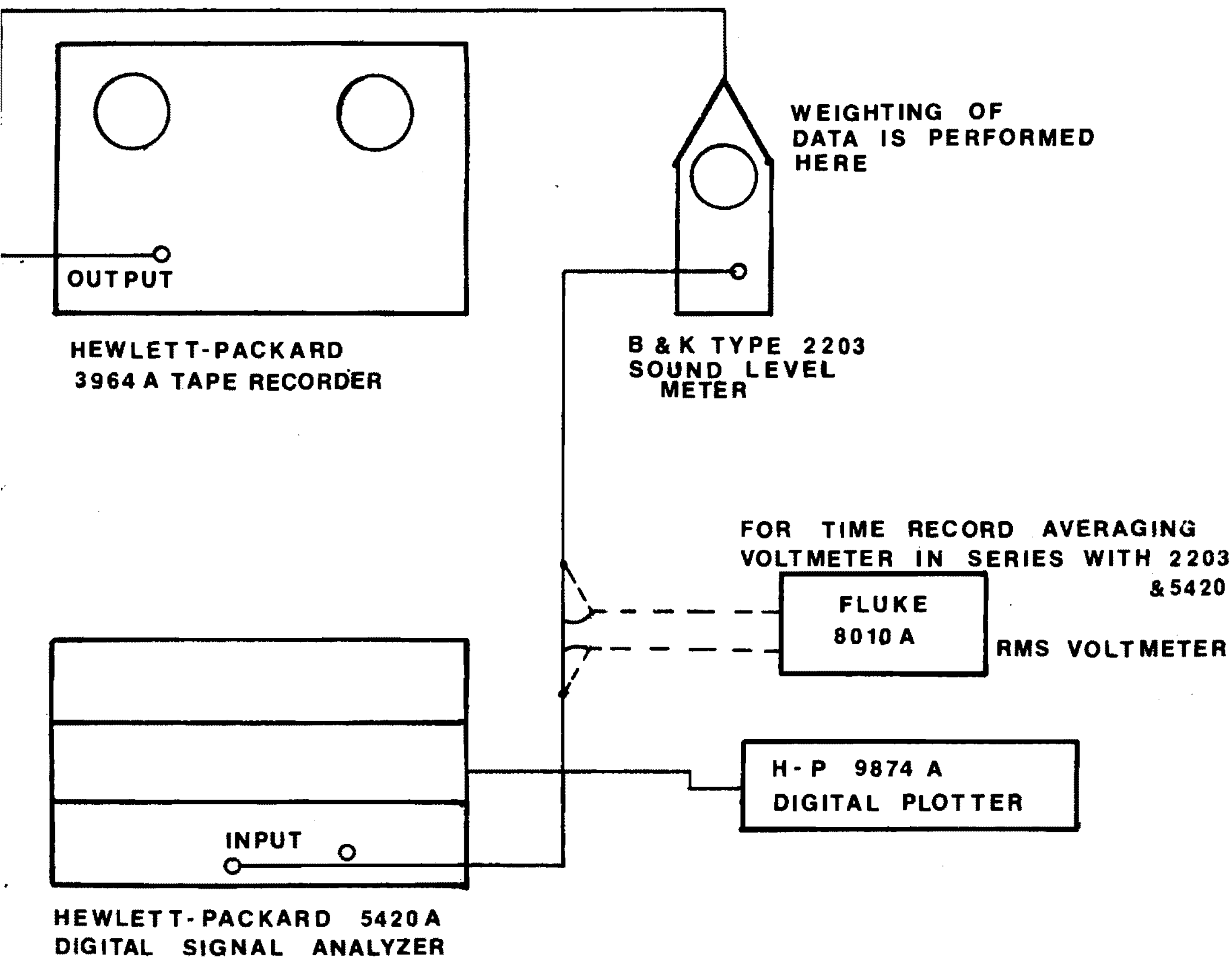


Figure 1A

DATA ANALYSIS CONFIGURATION

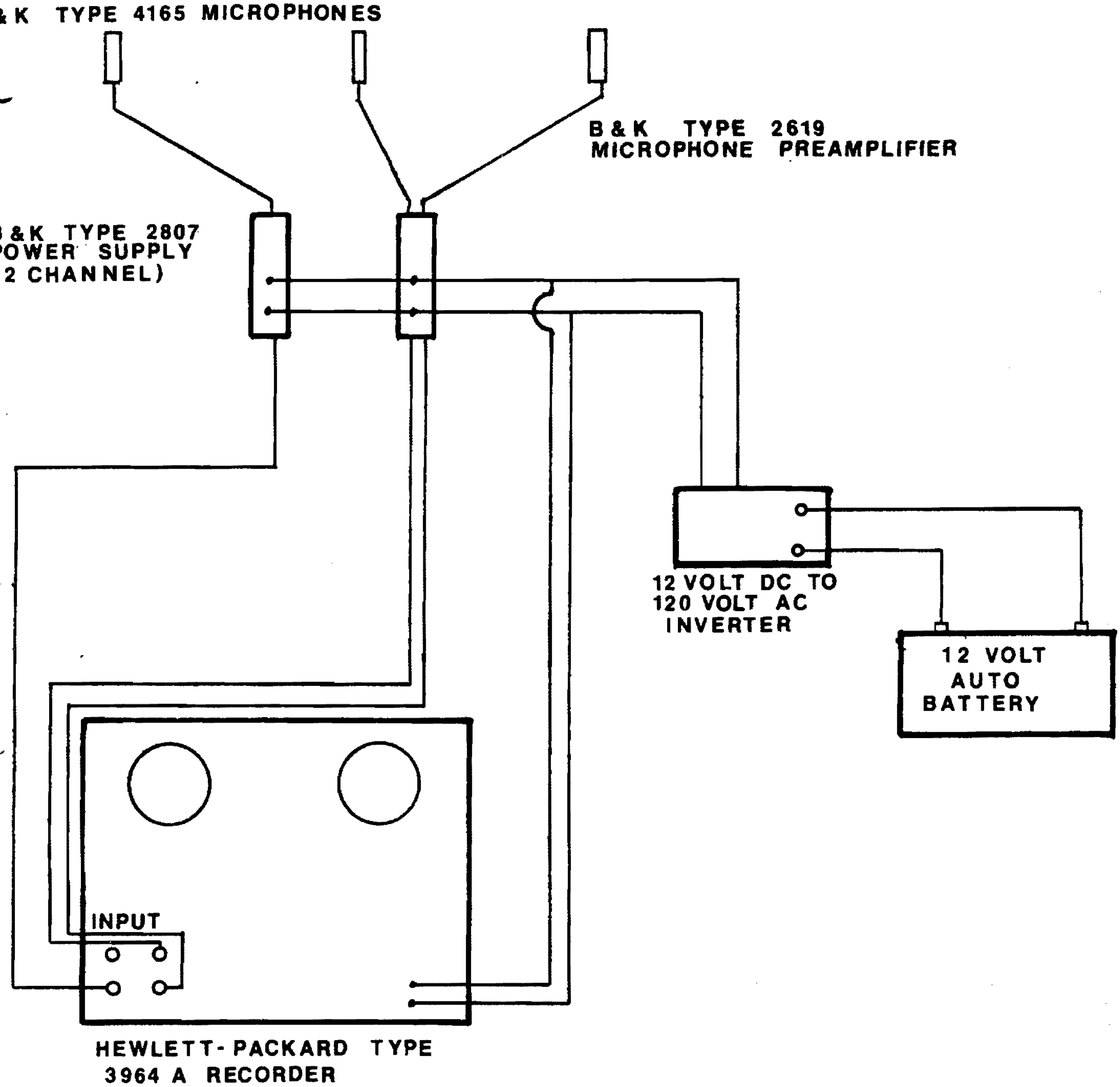


Figure 2A

DATA RECORDING CONFIGURATION

Appendix B

GENERAL PLANT ENVIRONMENT DATA

GENERAL PLANT ENVIRONMENT DATA

The figures in this appendix show frequency spectra and time histories of selected measurement points observed in both plants during normal operations. While not exhaustive, these points provide an example of the frequency characteristics observed throughout the noise field. The plant name and measurement position for each graph are noted in the upper right-hand corner. These values correspond to those coordinates listed in Figures 1-B and 2-B. Both Linear and A-weighted readings are presented for each point selected.

Warning: The frequency data are presented in both a linear and logarithmic fashion. Since the analyzer used was only capable of performing constant bandwidth analysis the logarithmic presentation is merely a distorted presentation of the constant bandwidth analysis. It is presented here only for those readers who are more familiar with viewing constant percentage bandwidth outputs.

Again, it must be stressed that the logarithmic presentations are not the result of constant percentage bandwidth analysis, but merely a distorted presentation of constant bandwidth analysis.

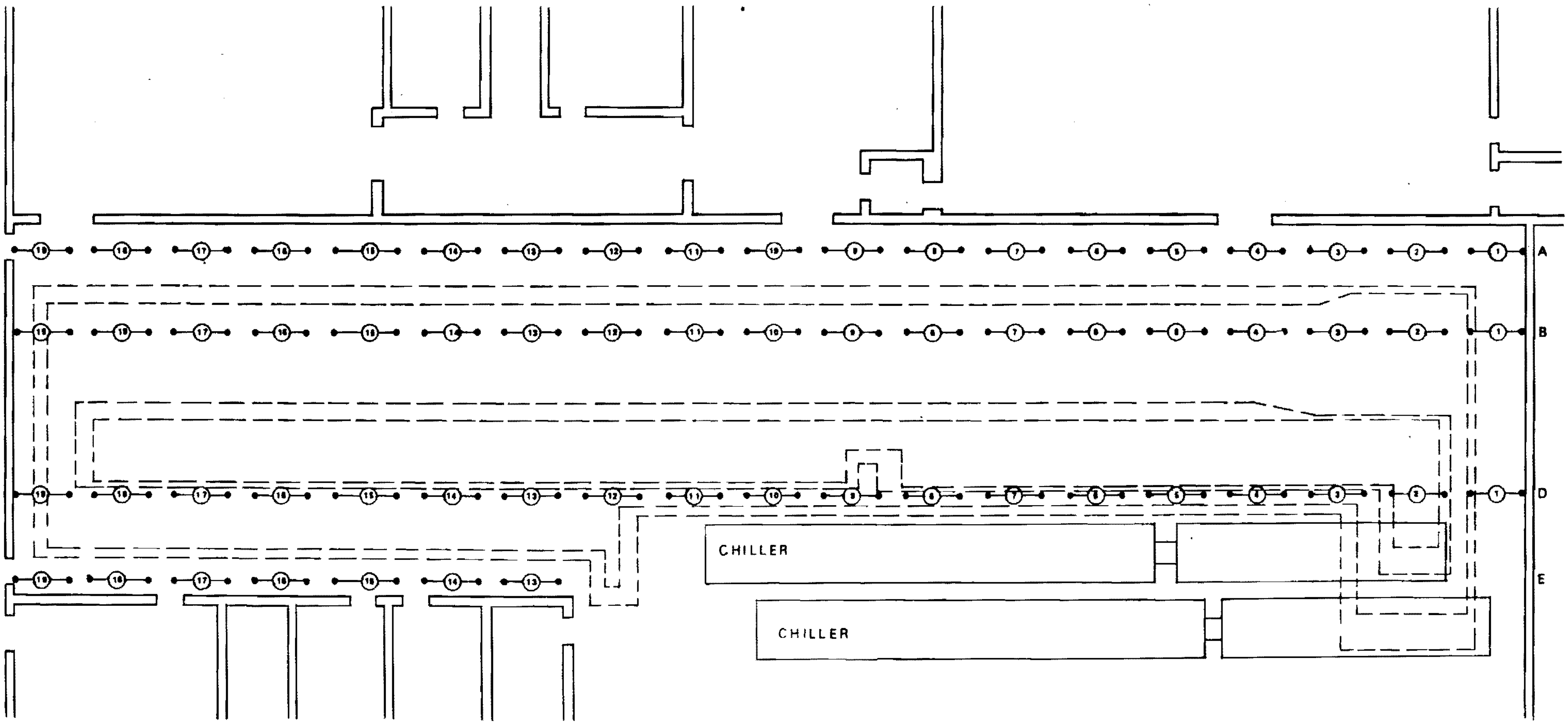


FIG. 1 B LOCATION OF MEASUREMENT POINTS IN CENTRAL SOYA

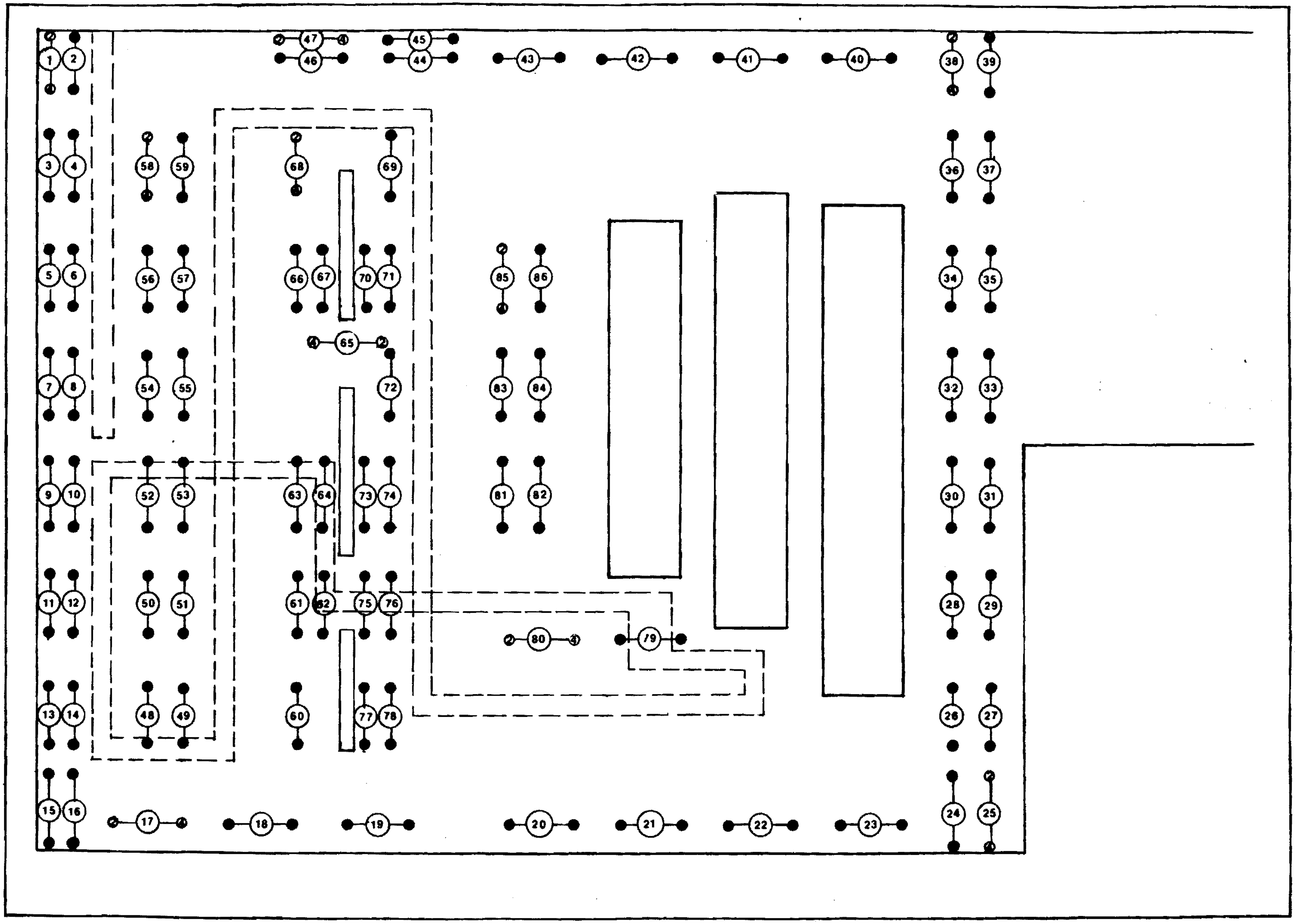
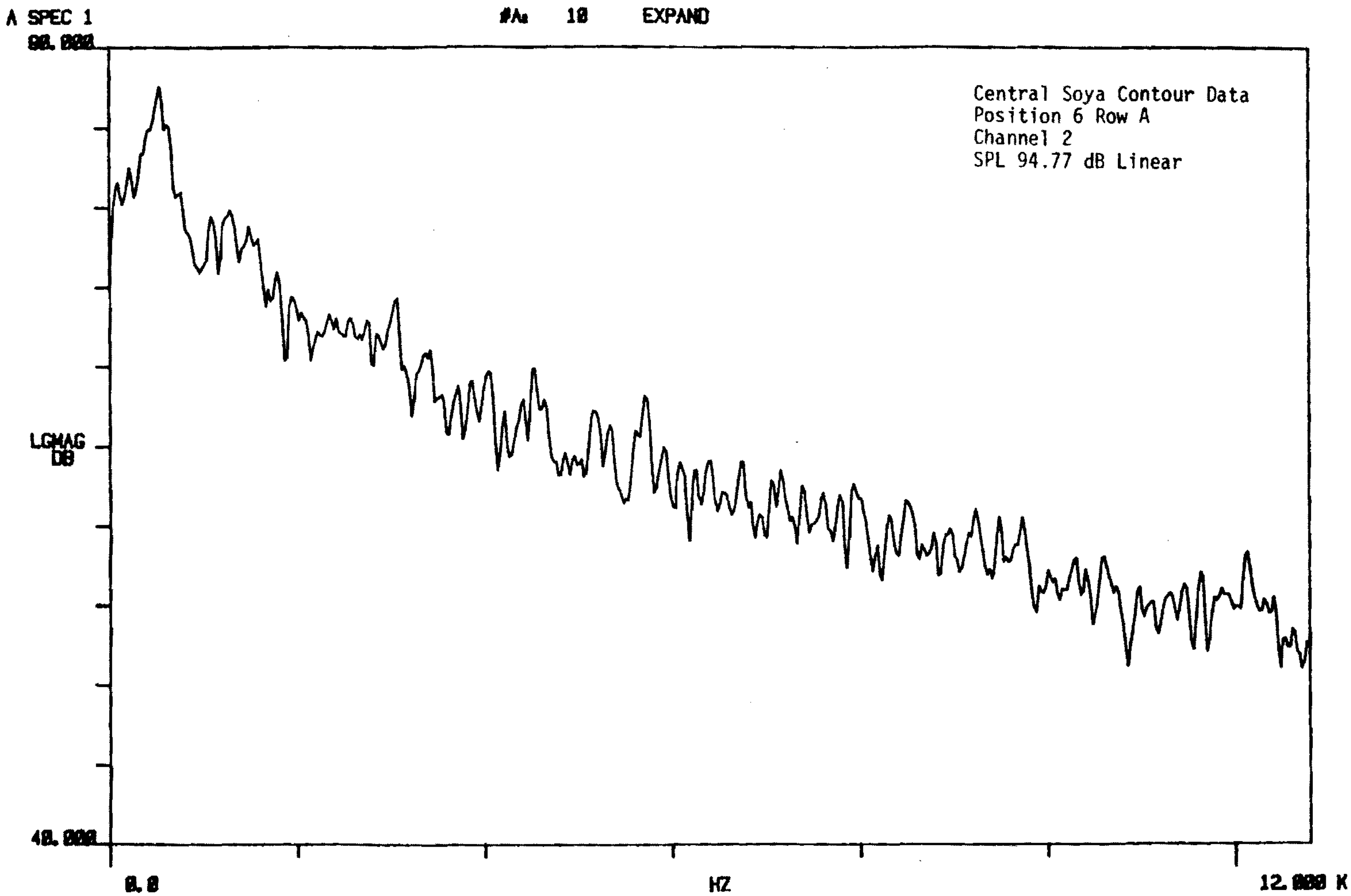


FIG. 2B LOCATION OF MEASUREMENT POINTS IN TIP TOP

Noise Contour

Data

FREQUENCY SPECTRA CENTRAL SOYA PLANT



A SPEC 1
#A: 10
EXPAND
80.000
Central Soya Contour Data
Position 6 Row A
Channel 2
SPL 94.77 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 3B

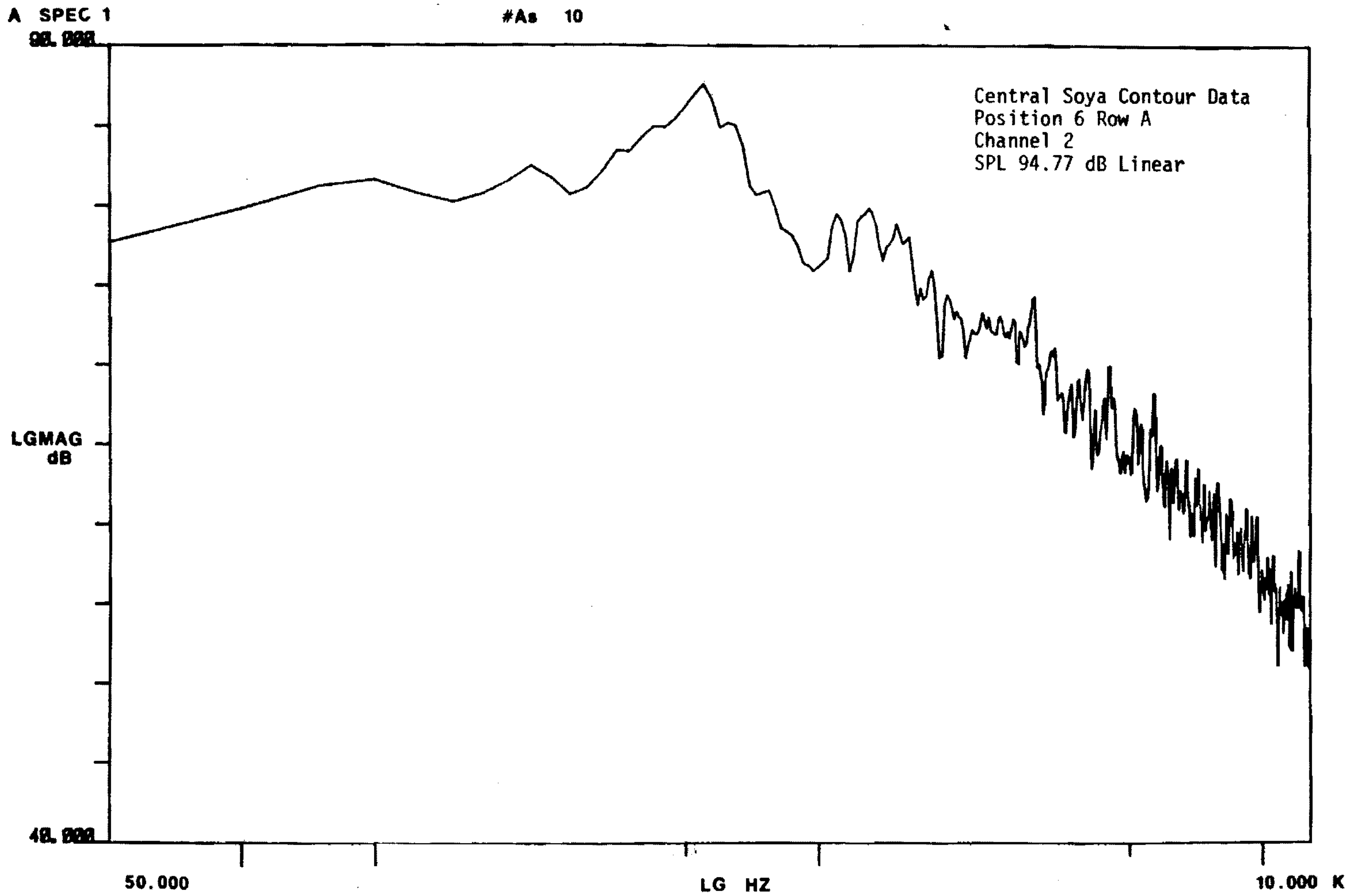

A SPEC 1
#As 10
90.000
Central Soya Contour Data
Position 6 Row A
Channel 2
SPL 94.77 dB Linear
LGMAG
dB
40.000
50.000
LG HZ
10.000 K

Figure 4B

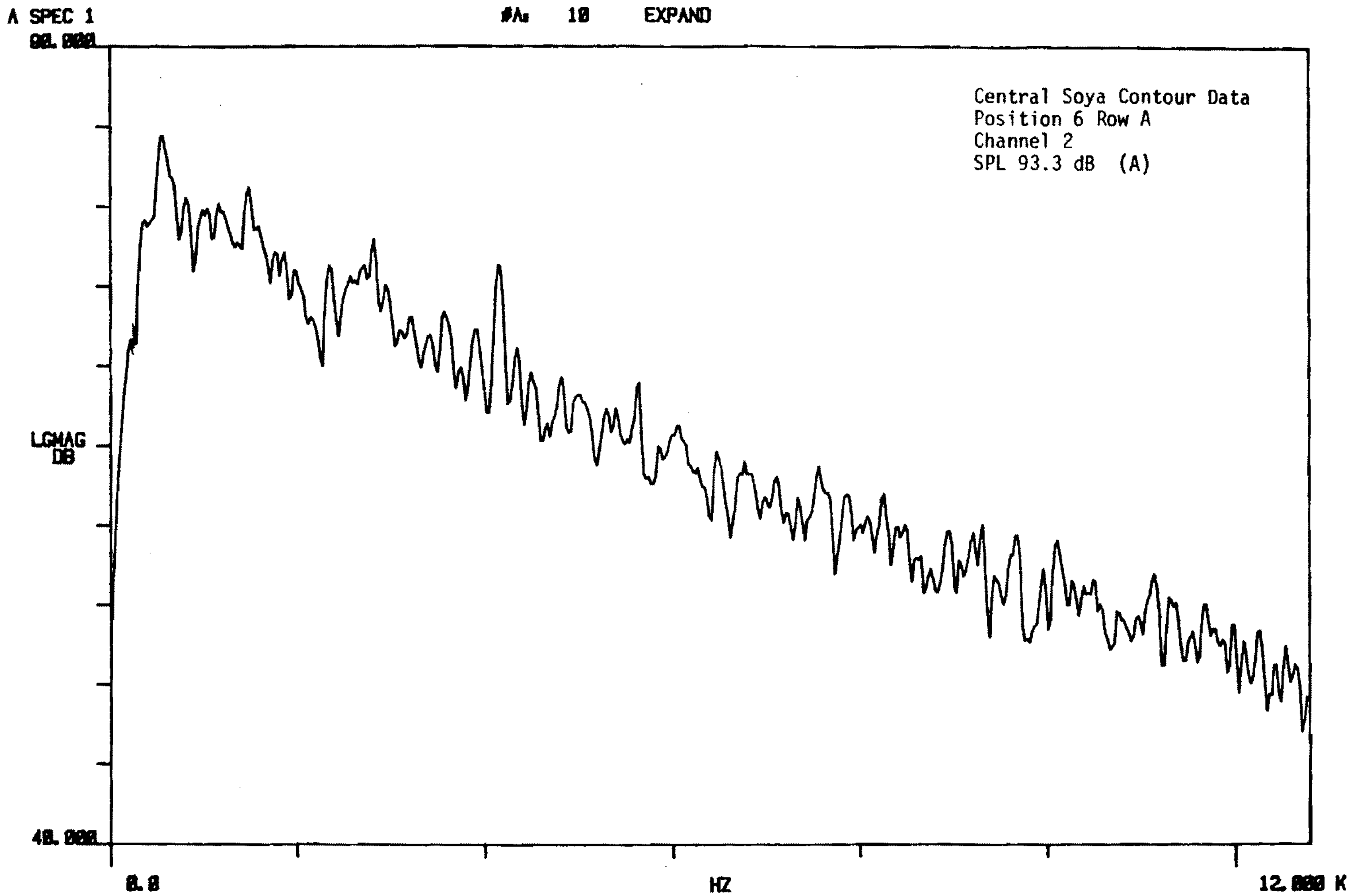

A SPEC 1
#A: 10
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 6 Row A
Channel 2
SPL 93.3 dB (A)

Figure 5B

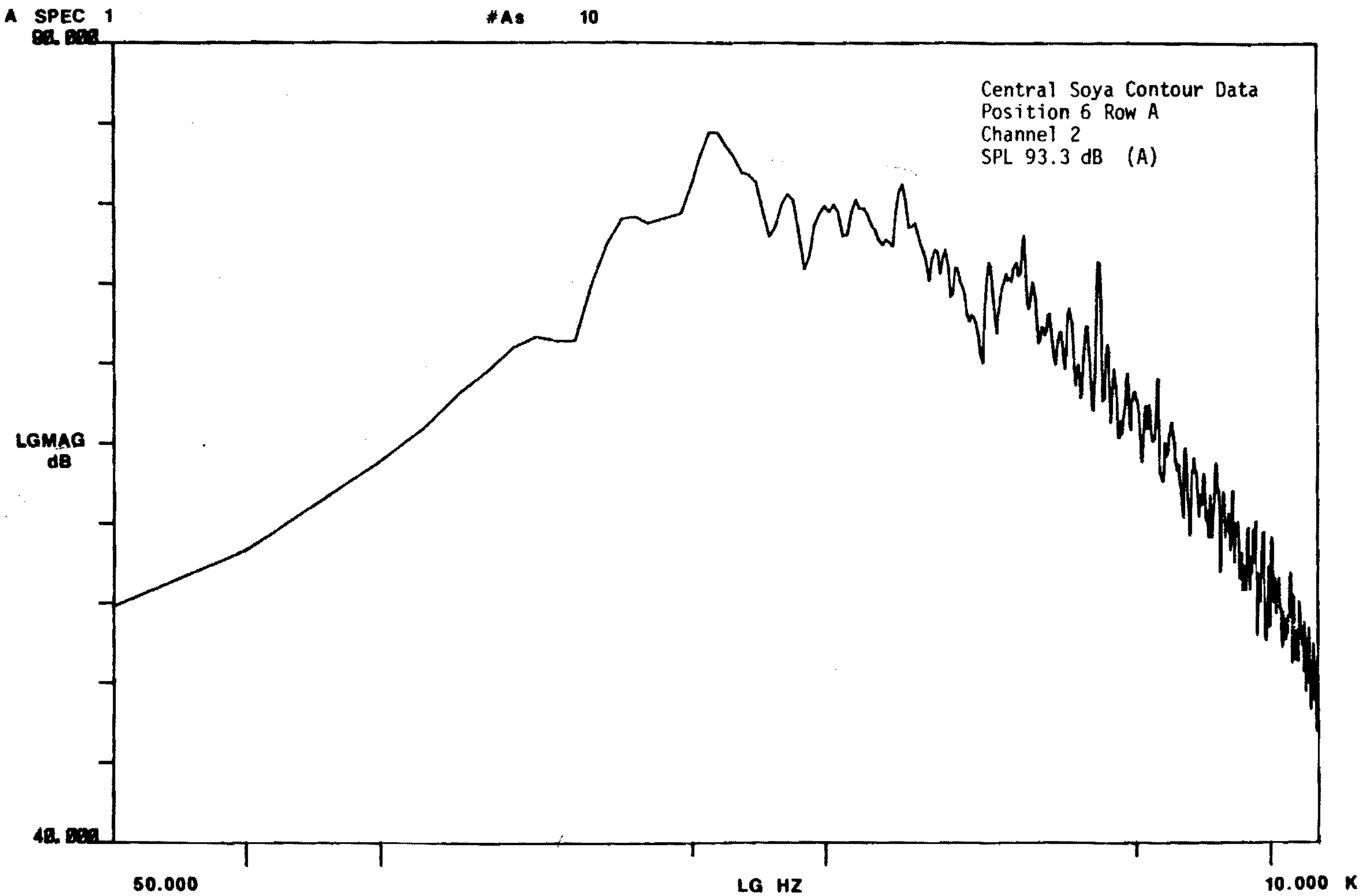

A SPEC 1
#As 10
90.000
LGMAG
dB
40.000
50.000
LG HZ
10.000 K
Central Soya Contour Data
Position 6 Row A
Channel 2
SPL 93.3 dB (A)

Figure 6B

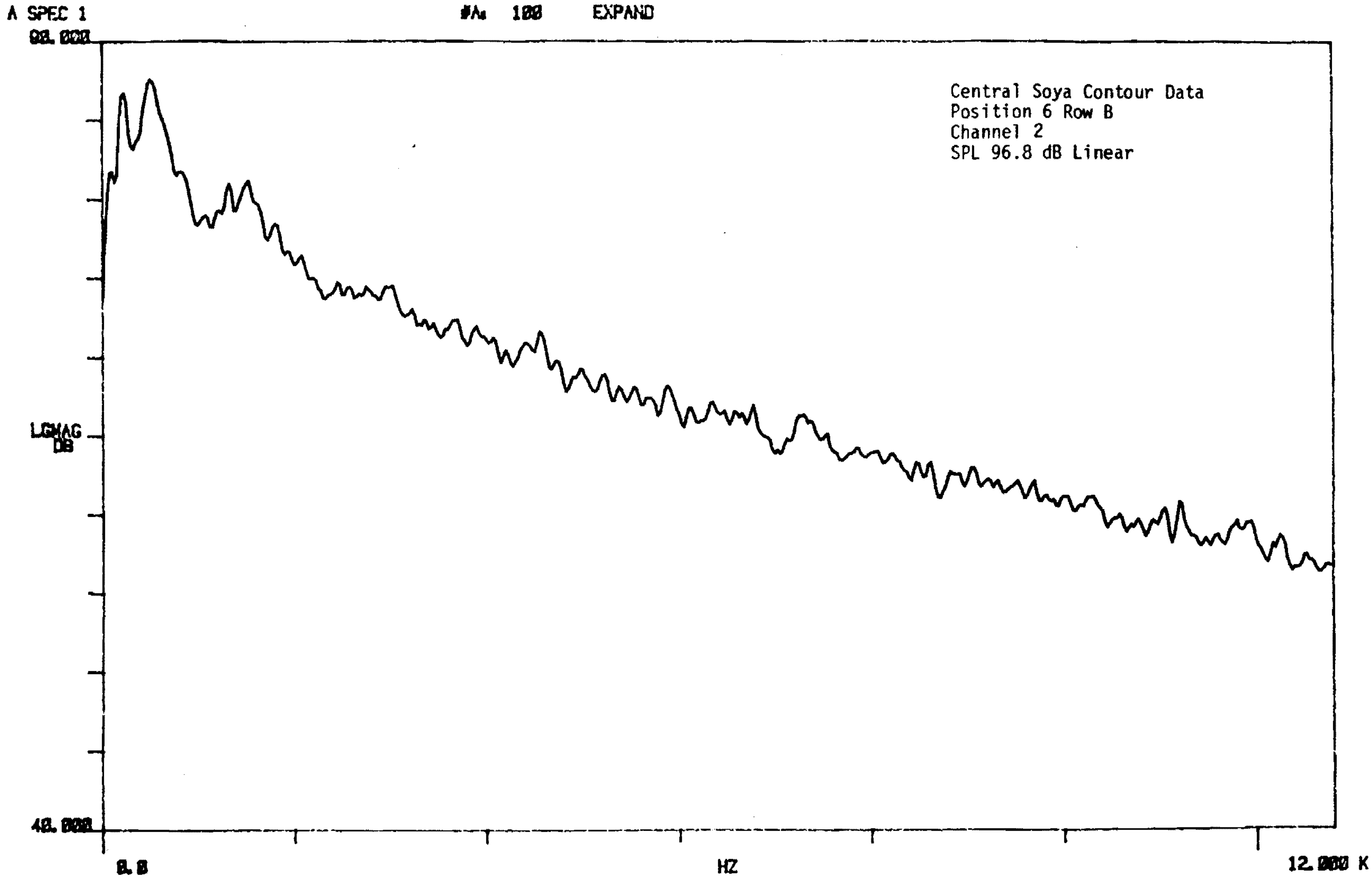

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 6 Row B
Channel 2
SPL 96.8 dB Linear

Figure 7B

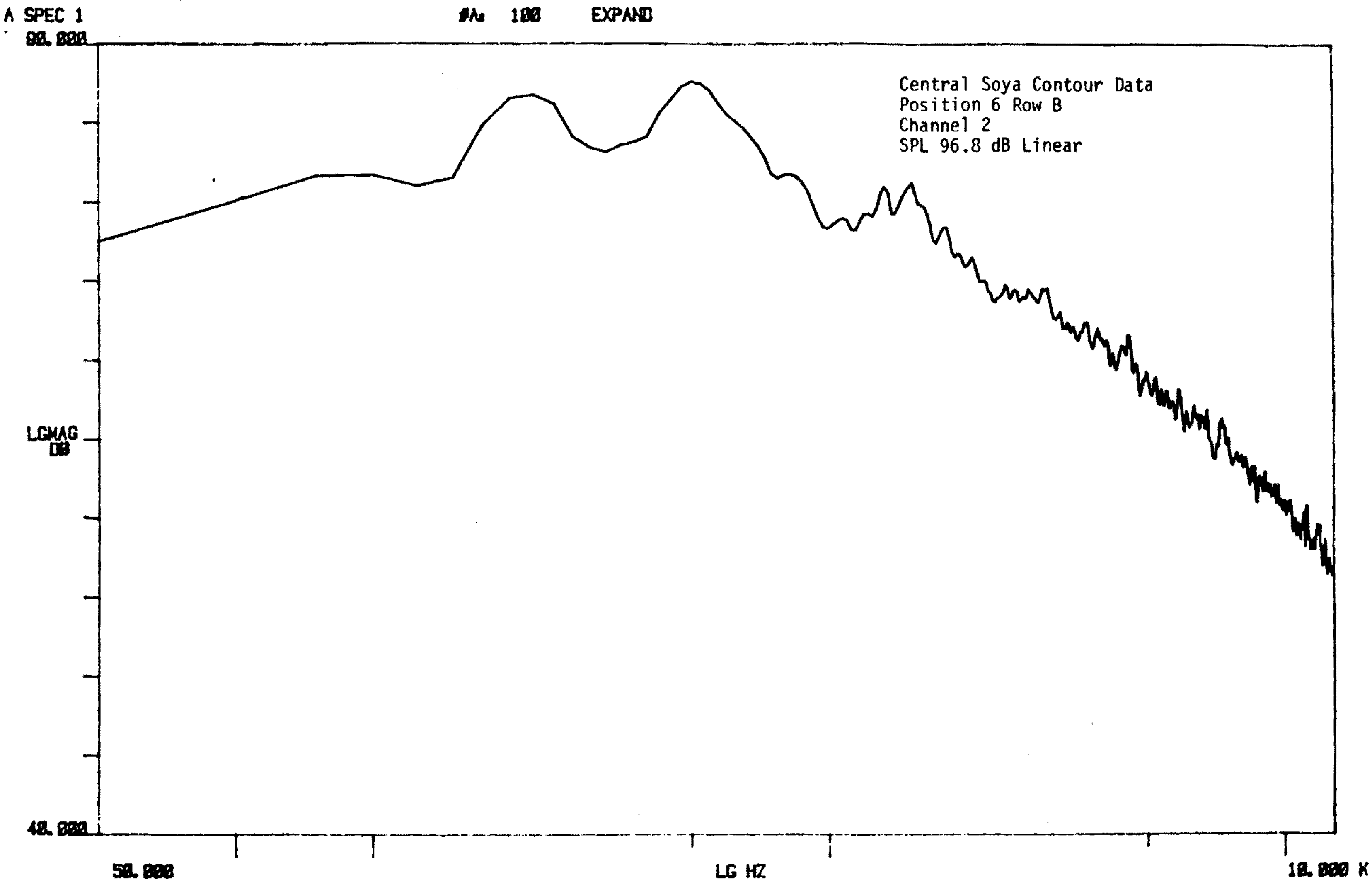
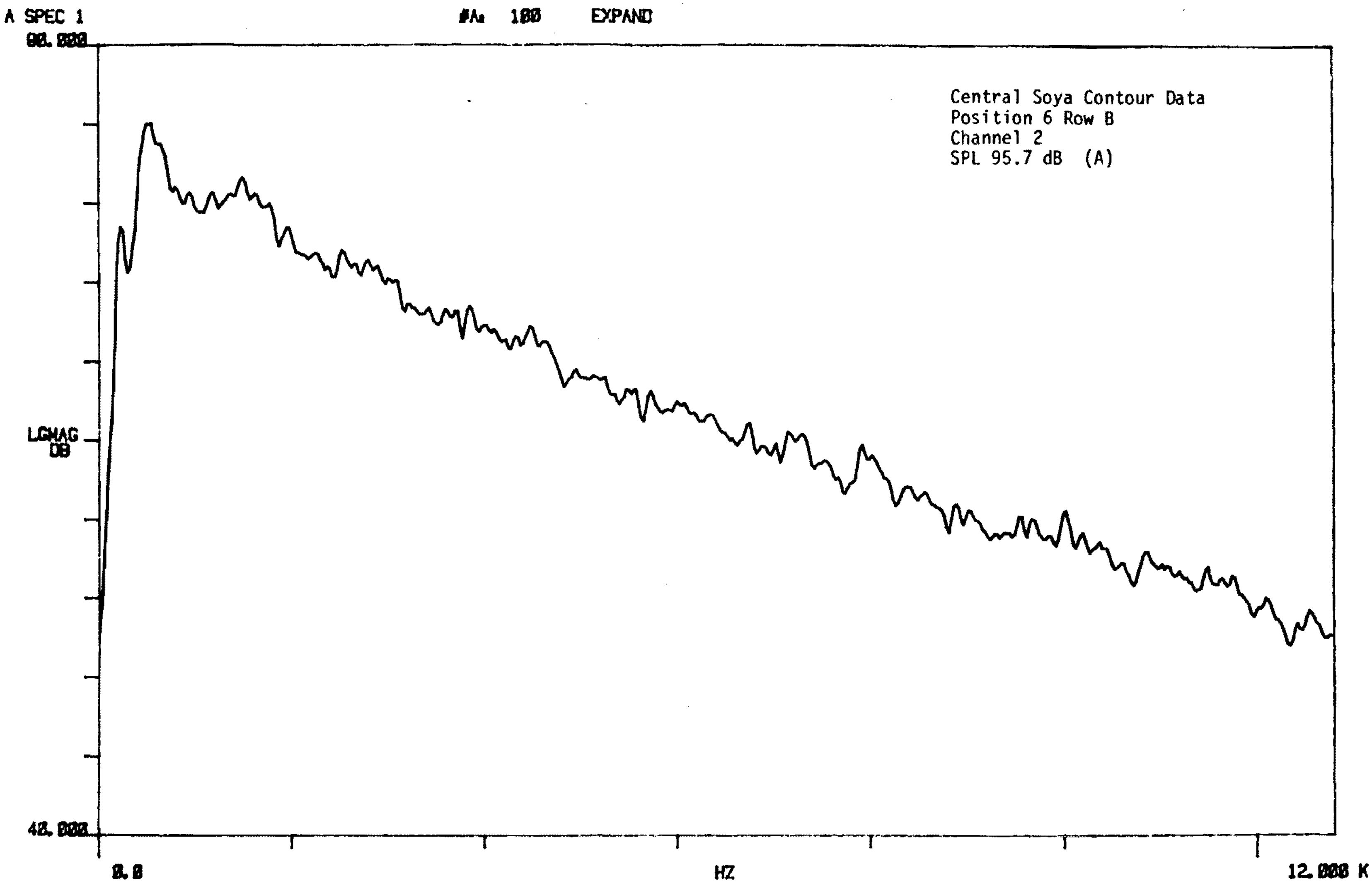

A SPEC 1
#A: 100
EXPAND
90.000
Central Soya Contour Data
Position 6 Row B
Channel 2
SPL 96.8 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 8B



A SPEC 1
#A: 100 EXPAND
90.000
Central Soya Contour Data
Position 6 Row B
Channel 2
SPL 95.7 dB (A)
LGMAG DB
40.000
0.0
HZ
12.000 K

Figure 9B

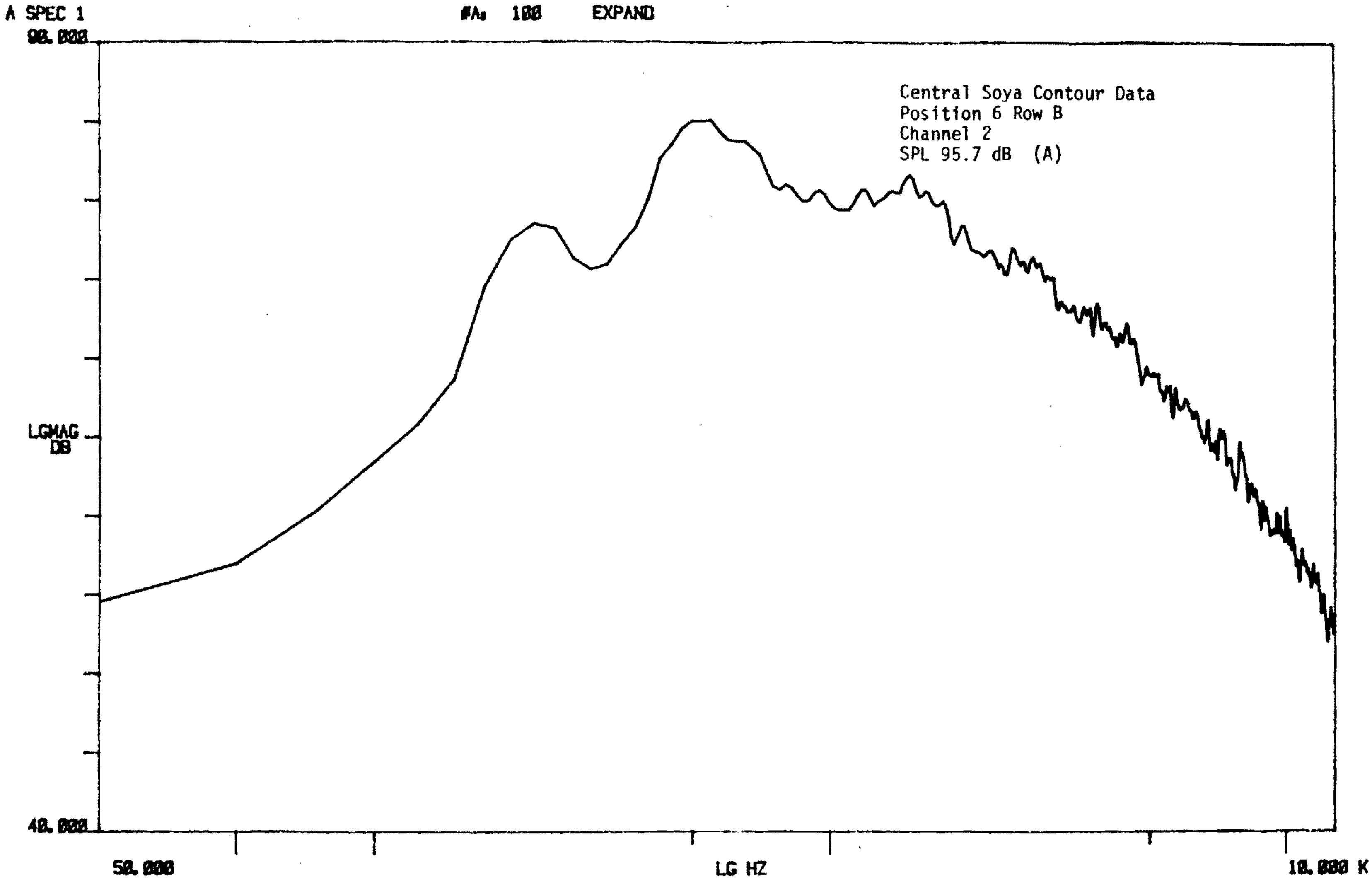
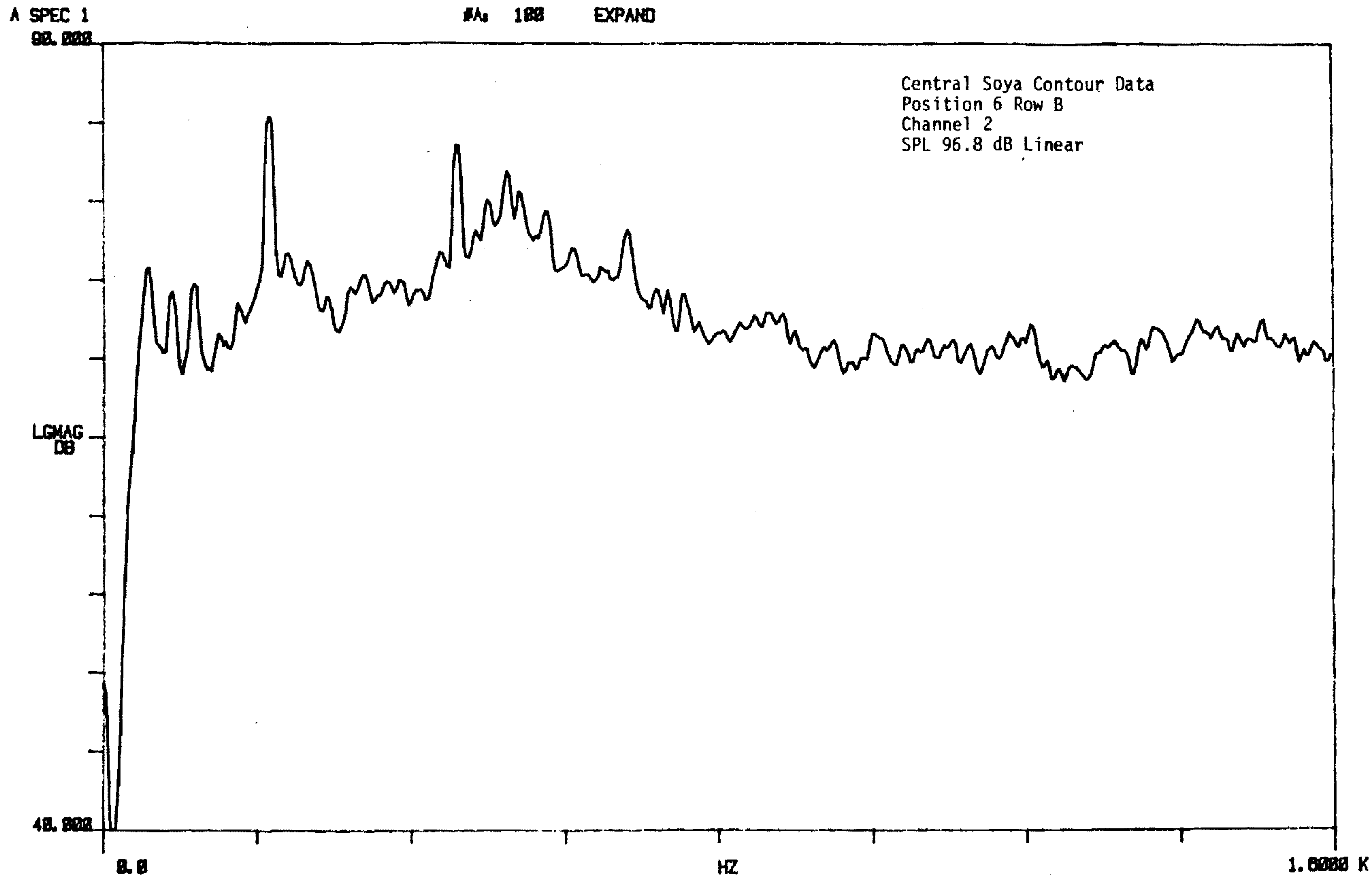

A SPEC 1
#A: 100
EXPAND
90.000
Central Soya Contour Data
Position 6 Row B
Channel 2
SPL 95.7 dB (A)
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 10B


A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
1.0000 K
Central Soya Contour Data
Position 6 Row B
Channel 2
SPL 96.8 dB Linear
Figure 11B

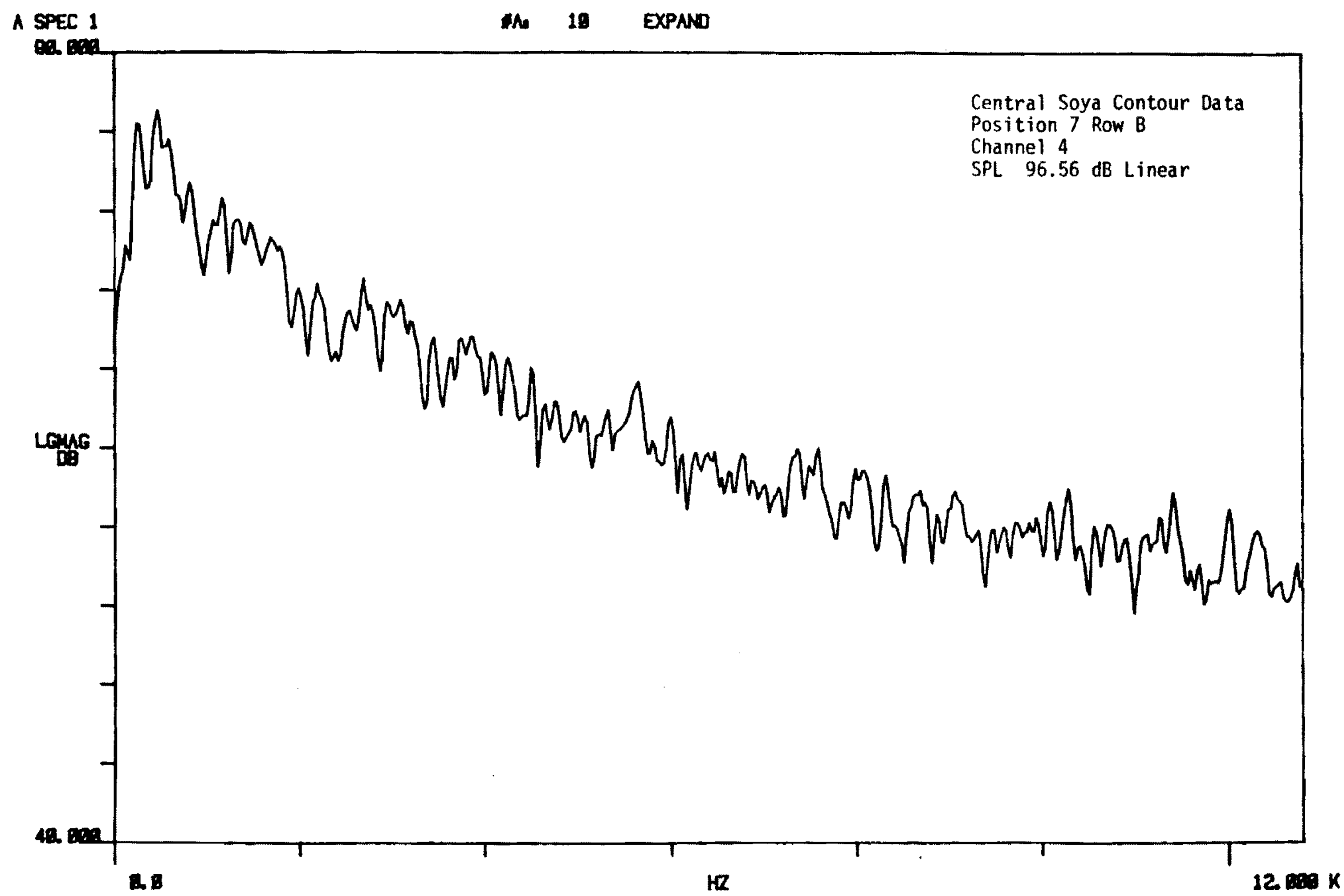

A SPEC 1
#A: 10
EXPAND
90.000
Central Soya Contour Data
Position 7 Row B
Channel 4
SPL 96.56 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 12B

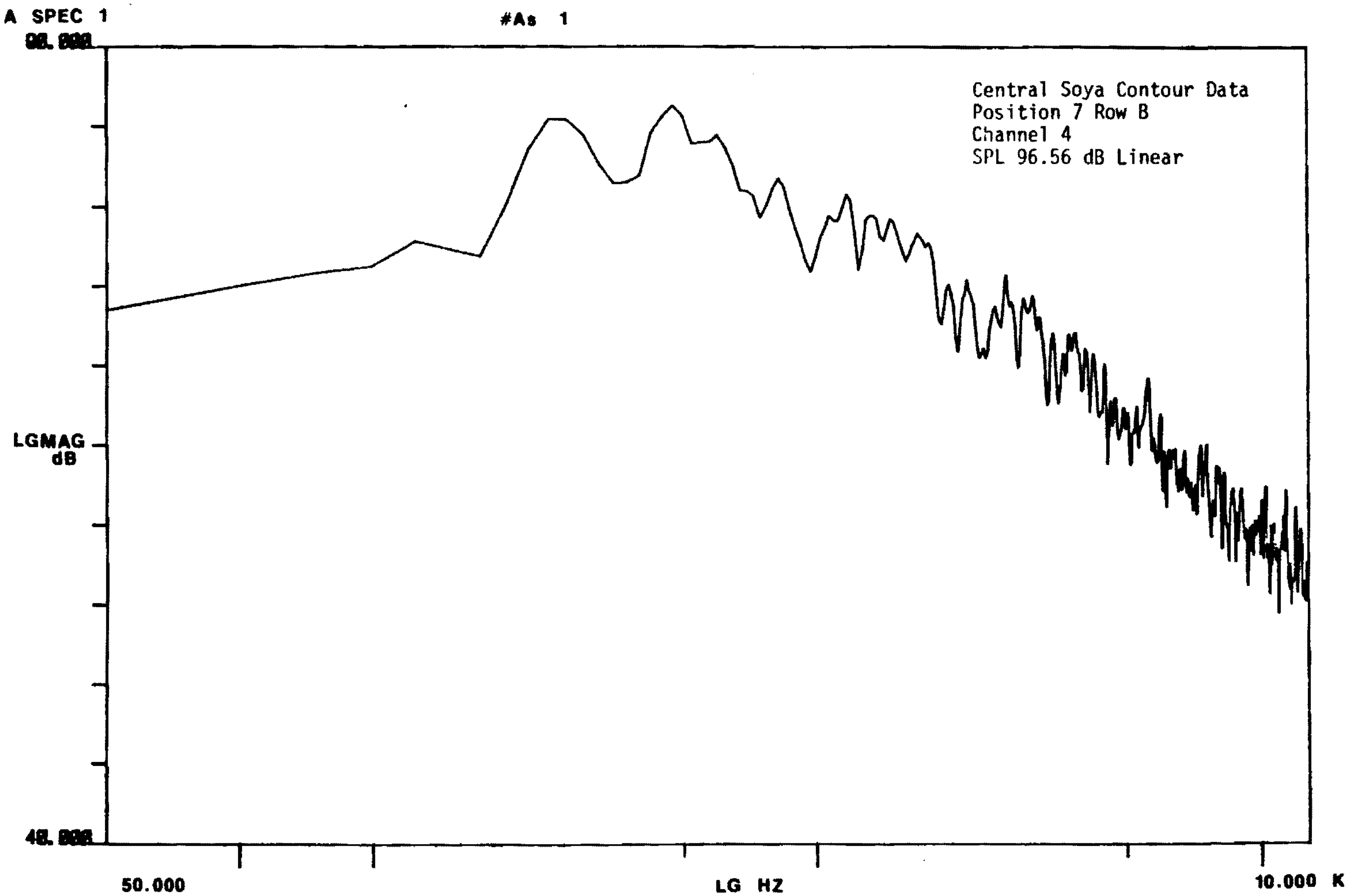
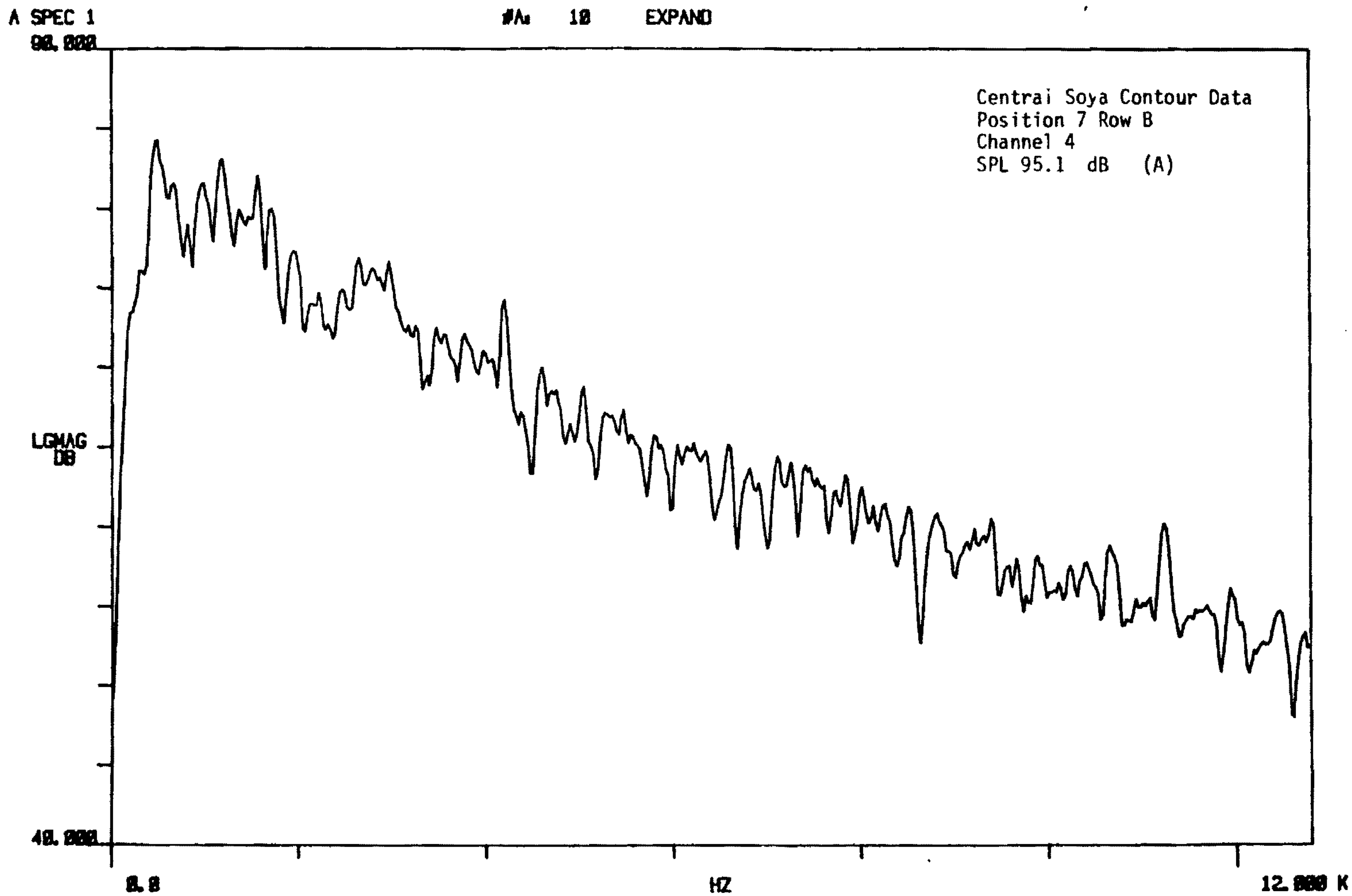

A SPEC 1
#As 1
Central Soya Contour Data
Position 7 Row B
Channel 4
SPL 96.56 dB Linear
LGMAG
dB
50.000
LG HZ
10.000 K

Figure 13B



A SPEC 1
#A: 10
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.800 K
Centrai Soya Contour Data
Position 7 Row B
Channel 4
SPL 95.1 dB (A)

Figure 14B

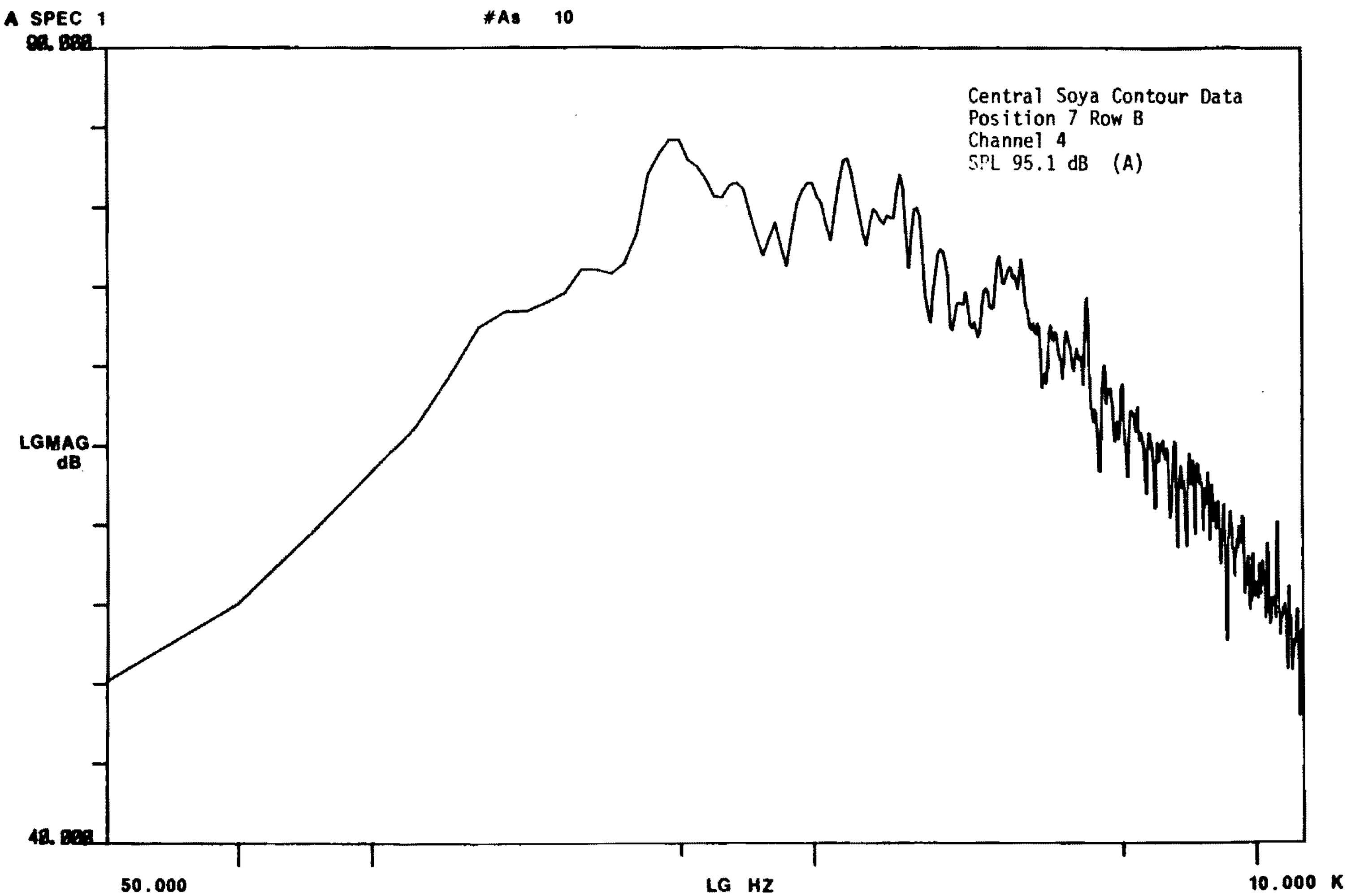

A SPEC 1
#As 10
Central Soya Contour Data
Position 7 Row B
Channel 4
SPL 95.1 dB (A)
LGMAG
dB
50.000
LG HZ
10.000 K

Figure 15B

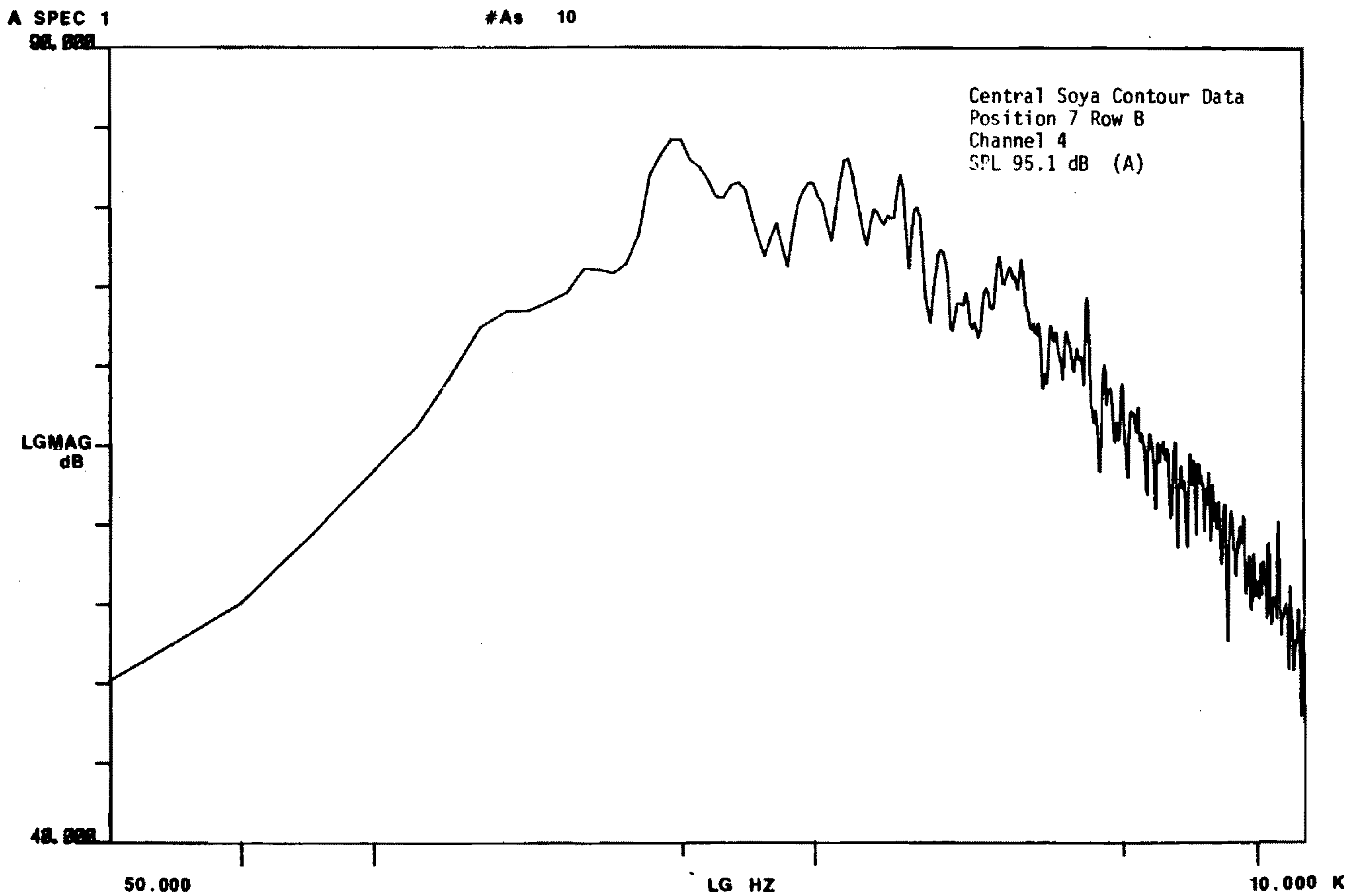
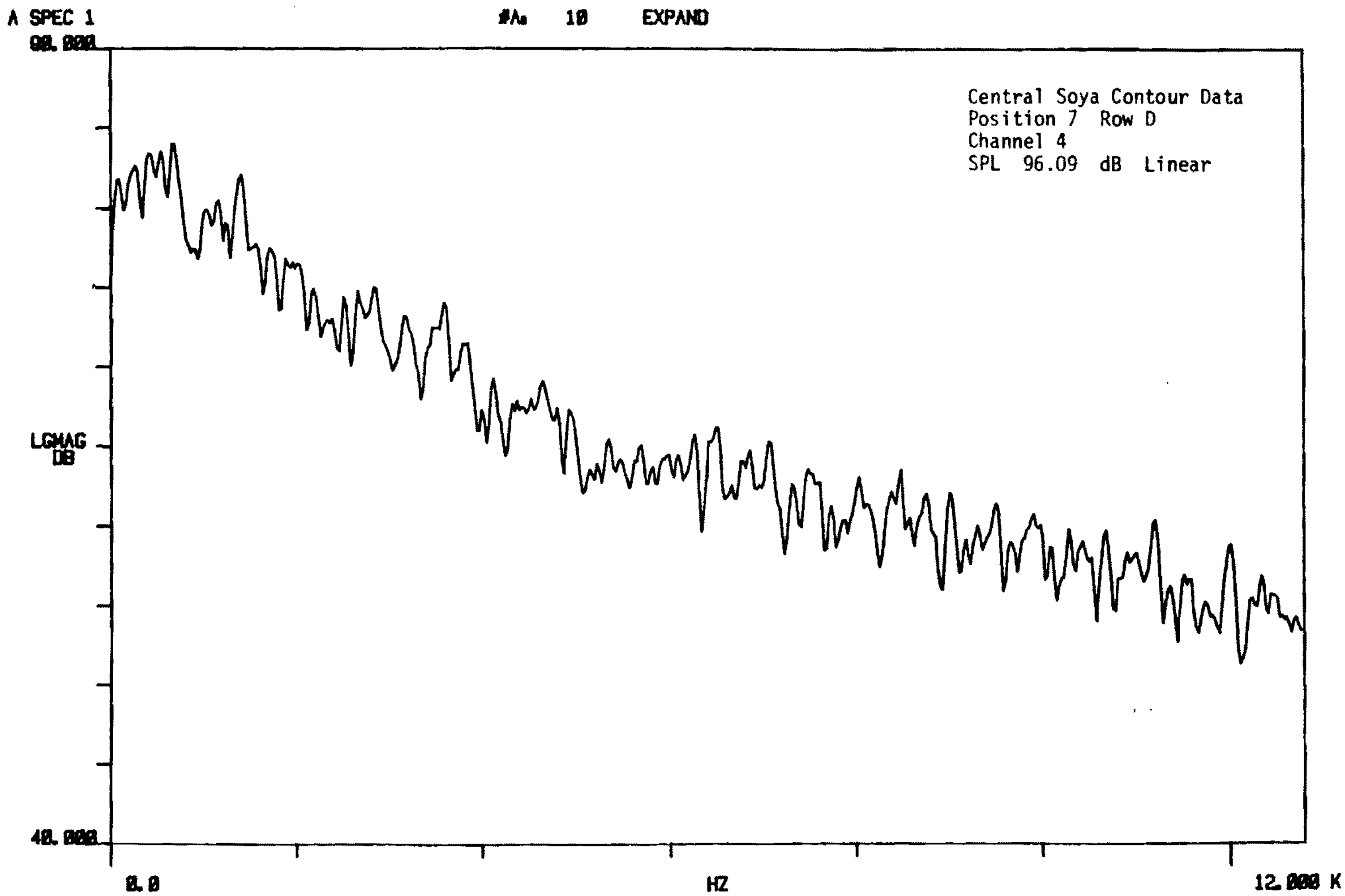

A SPEC 1
#As 10
98.000
LGMAG
dB
48.000
50.000
LG HZ
10.000 K
Central Soya Contour Data
Position 7 Row B
Channel 4
SPL 95.1 dB (A)

Figure 15B



A SPEC 1
#A: 10 EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 7 Row D
Channel 4
SPL 96.09 dB Linear

Figure 16B

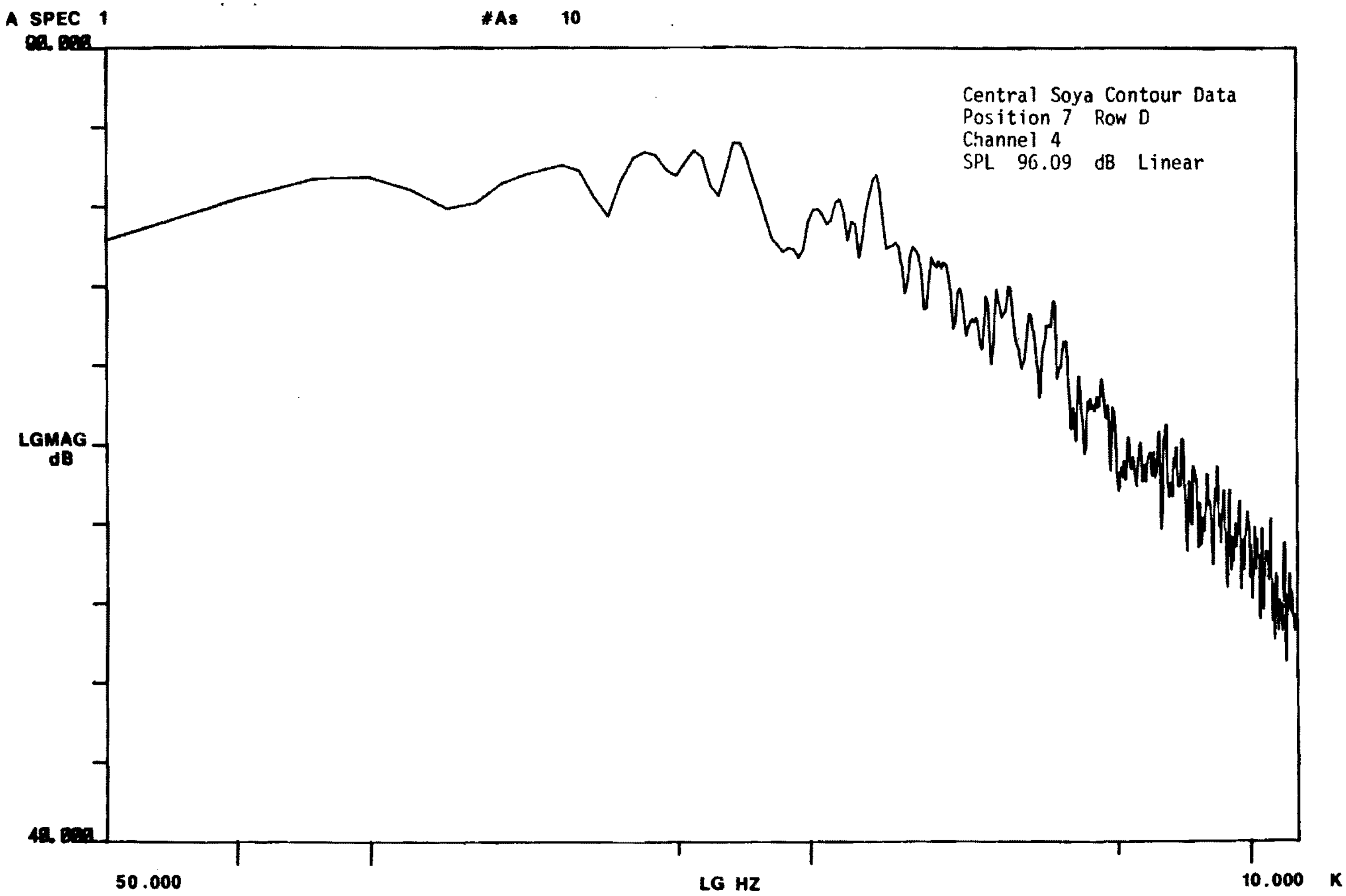

A SPEC 1
#As 10
90.000
Central Soya Contour Data
Position 7 Row D
Channel 4
SPL 96.09 dB Linear
LGMAG
dB
40.000
50.000
LG HZ
10.000 K

Figure 17B

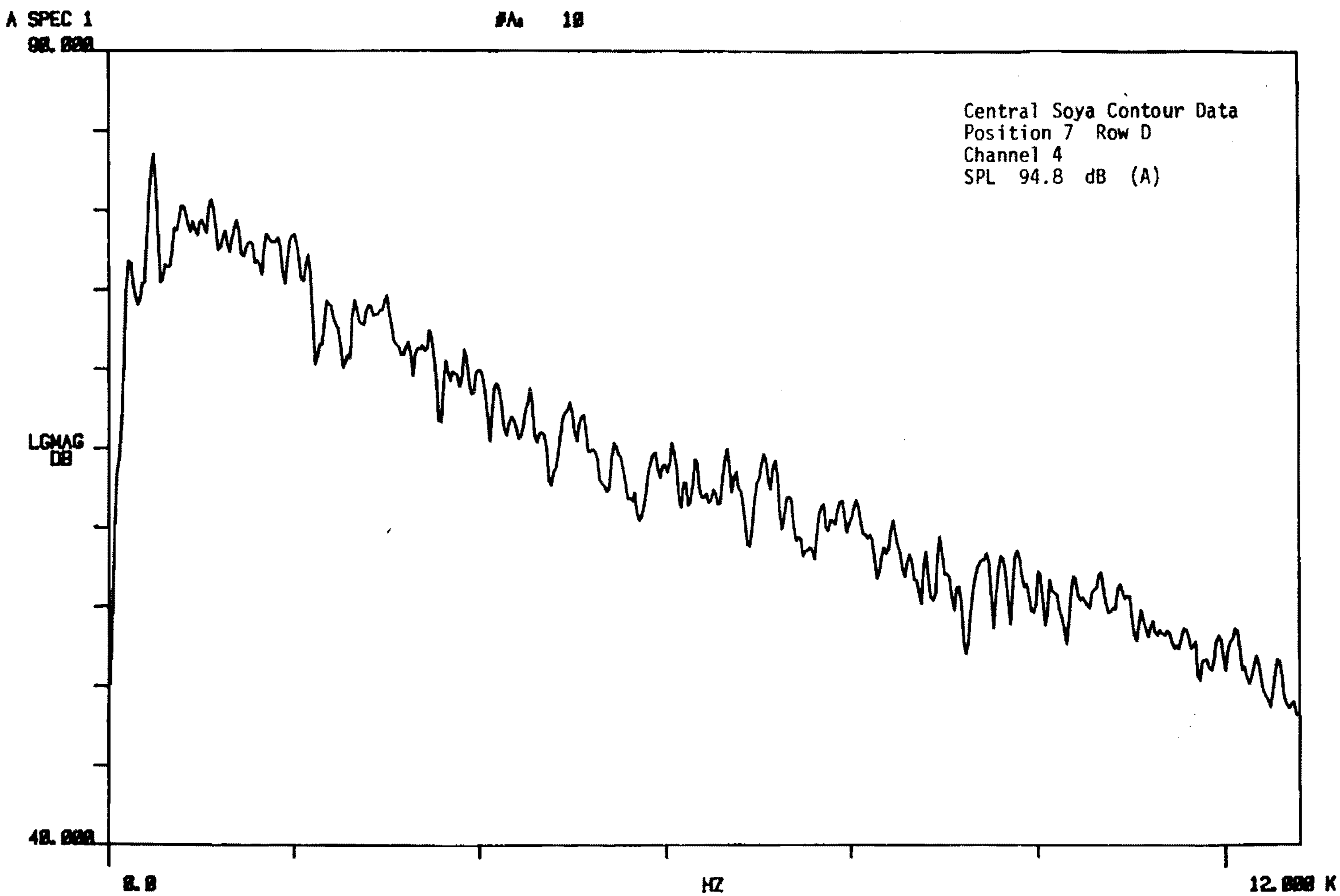
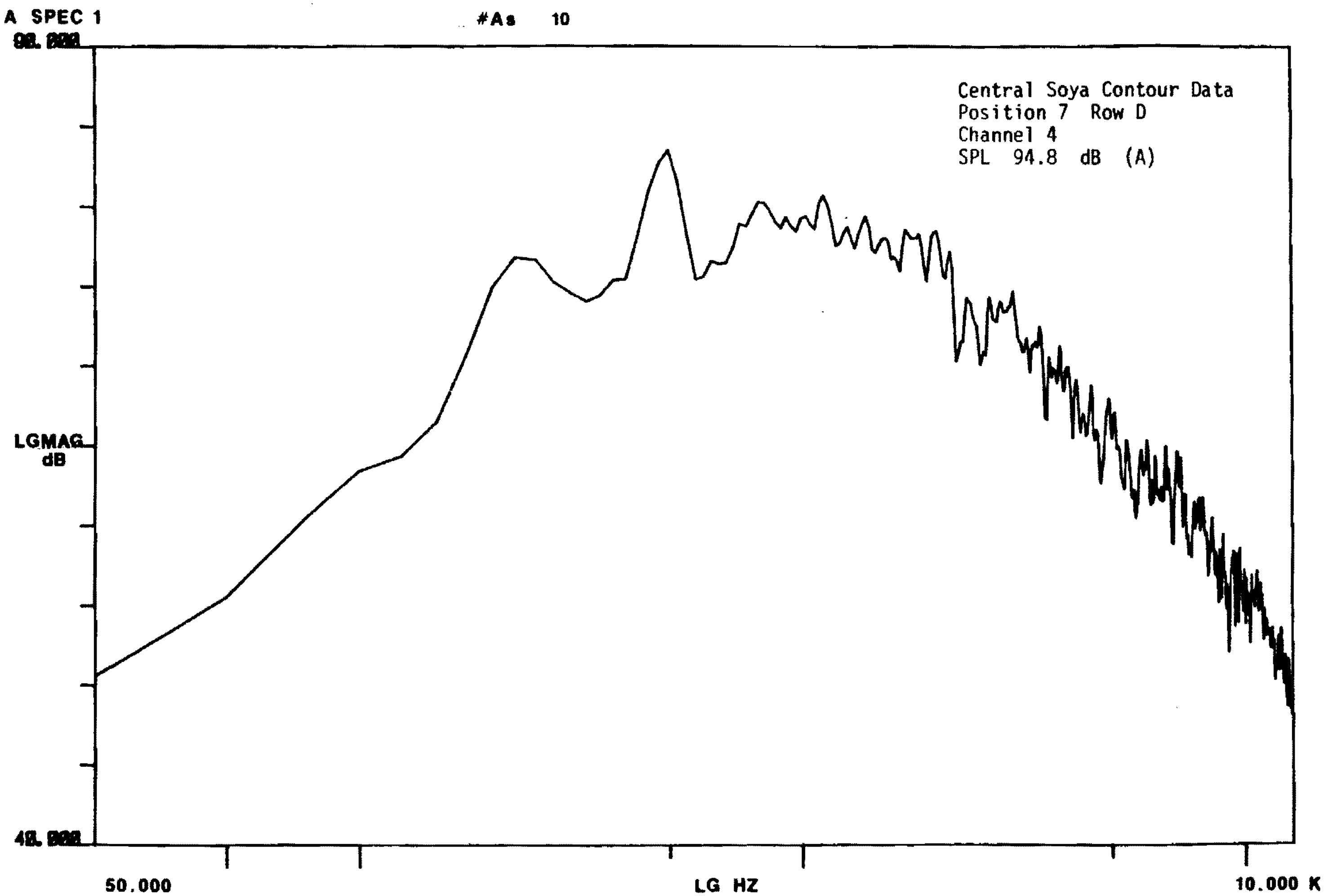

A SPEC 1
#A: 10
90.000
Central Soya Contour Data
Position 7 Row D
Channel 4
SPL 94.8 dB (A)
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 18B



A SPEC 1
#As 10
90.000
Central Soya Contour Data
Position 7 Row D
Channel 4
SPL 94.8 dB (A)
LGMAG
dB
40.000
50.000
LG HZ
10.000 K

Figure 19B

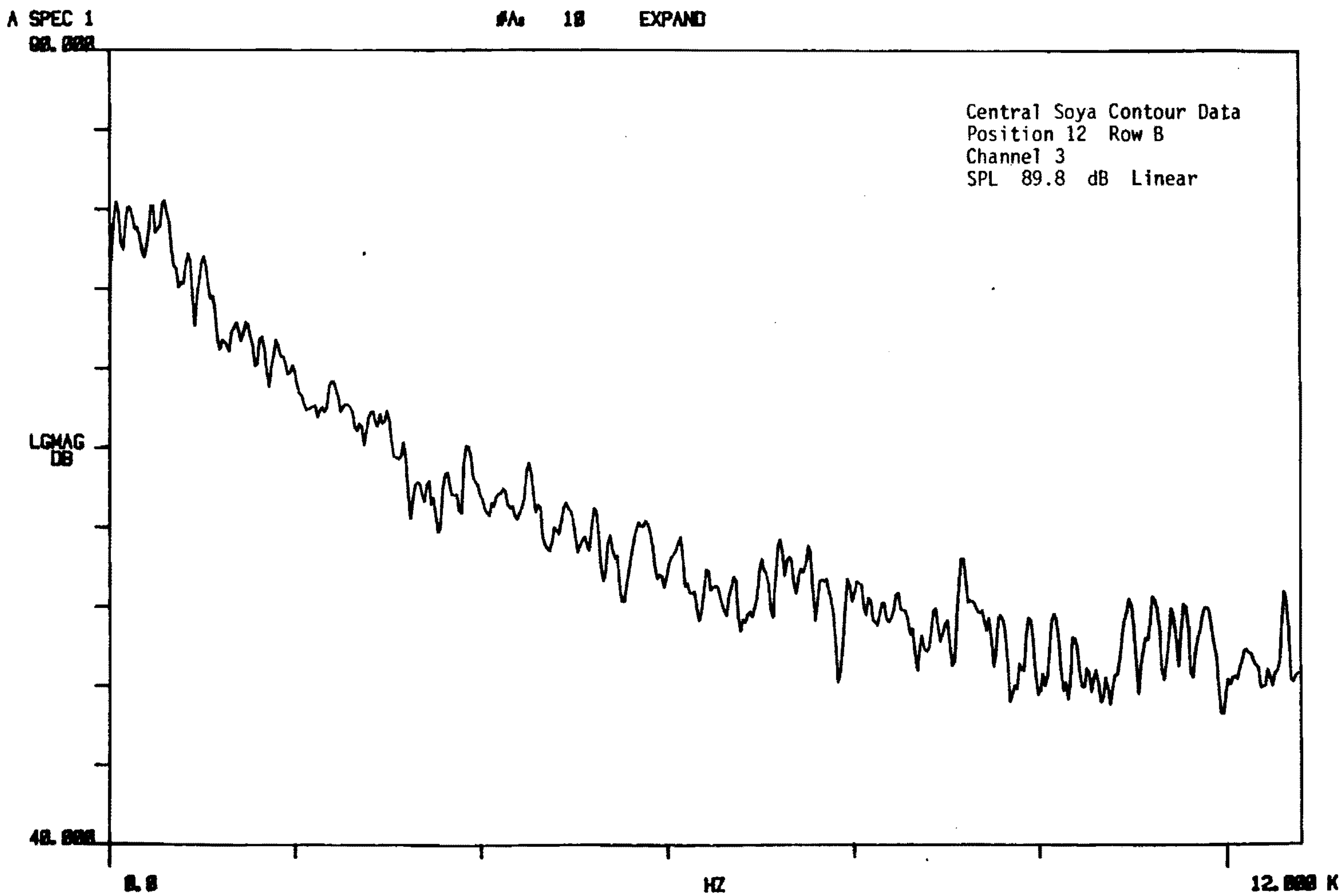

A SPEC 1
#A: 10
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 12 Row B
Channel 3
SPL 89.8 dB Linear

Figure 20B

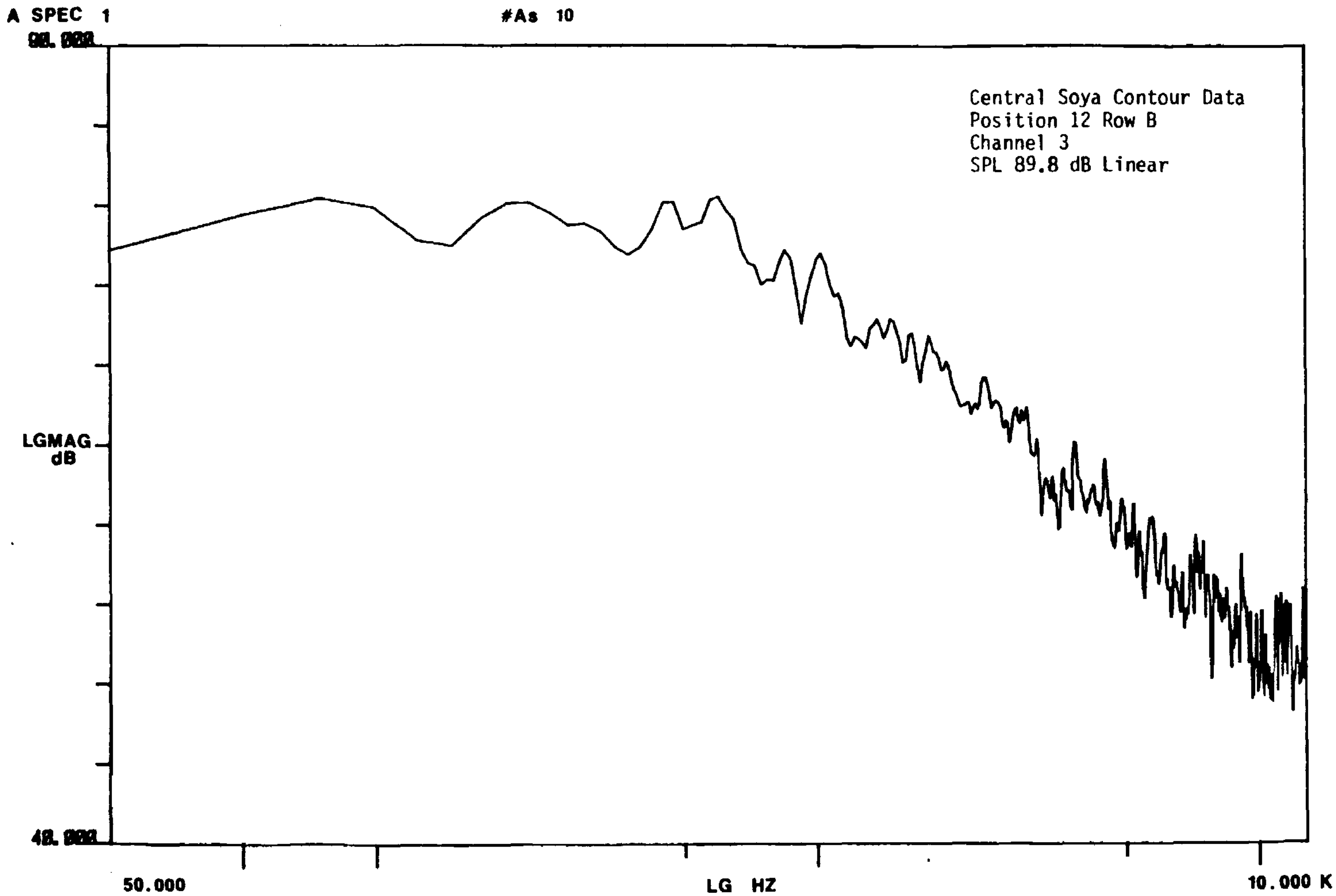
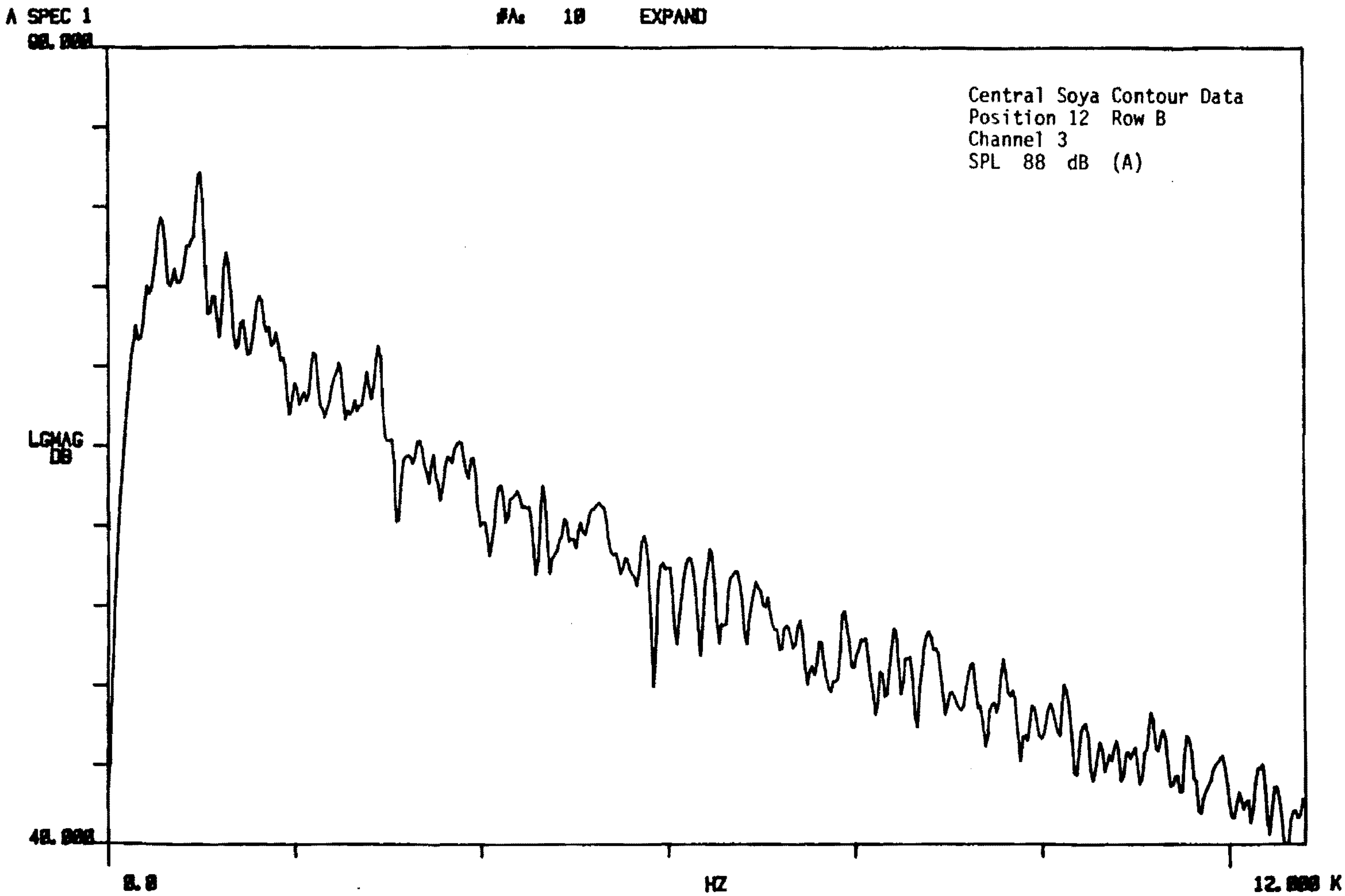

A SPEC 1
#As 10
Central Soya Contour Data
Position 12 Row B
Channel 3
SPL 89.8 dB Linear
LGMAG
dB
50.000
LG HZ
10.000 K

Figure 21B



A SPEC 1
#As 10 EXPAND
90.000
LGMAG DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 12 Row B
Channel 3
SPL 88 dB (A)

Figure 22B

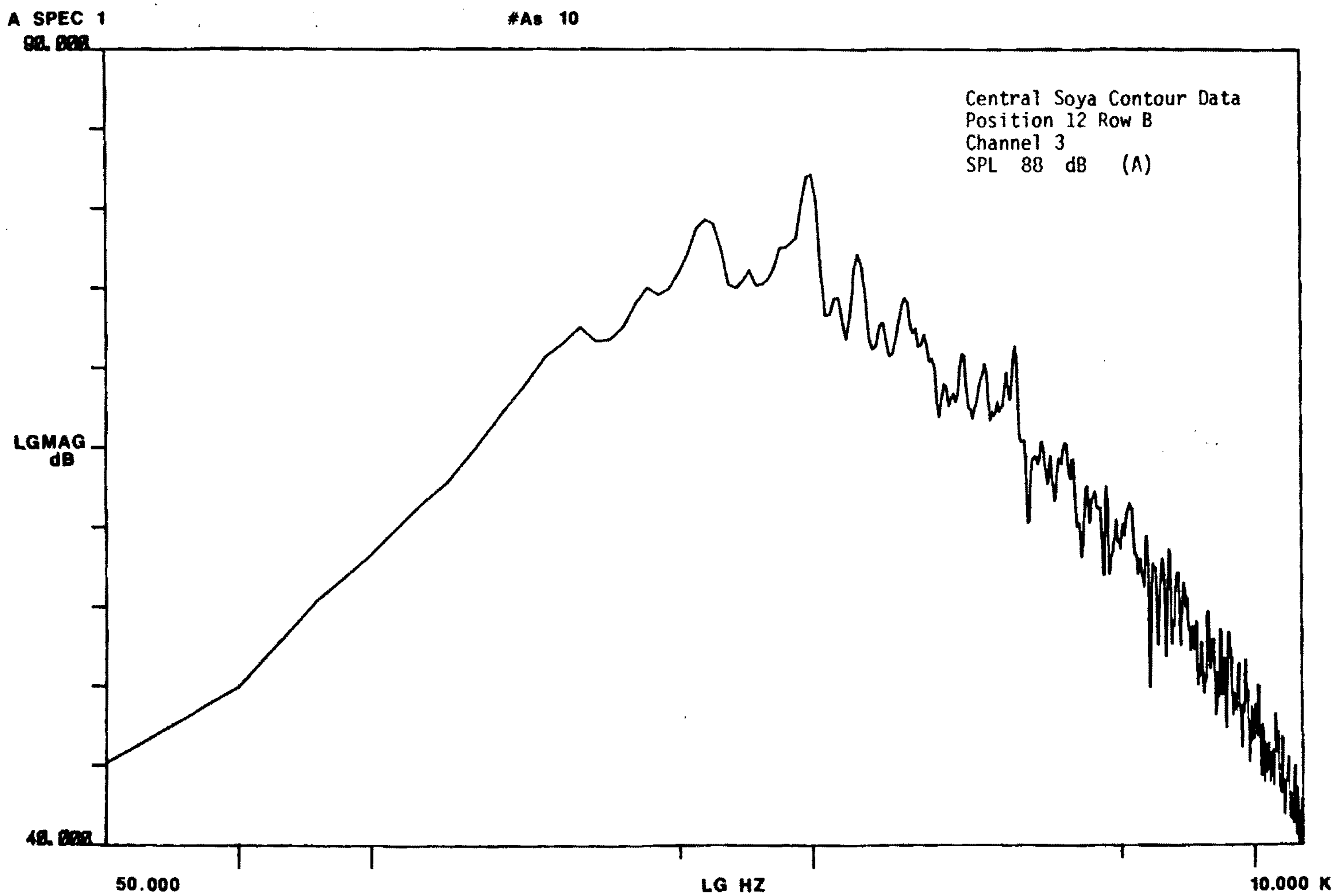

A SPEC 1
#As 10
90.000
LGMAG
dB
40.000
50.000
LG HZ
10.000 K
Central Soya Contour Data
Position 12 Row B
Channel 3
SPL 88 dB (A)

Figure 23B

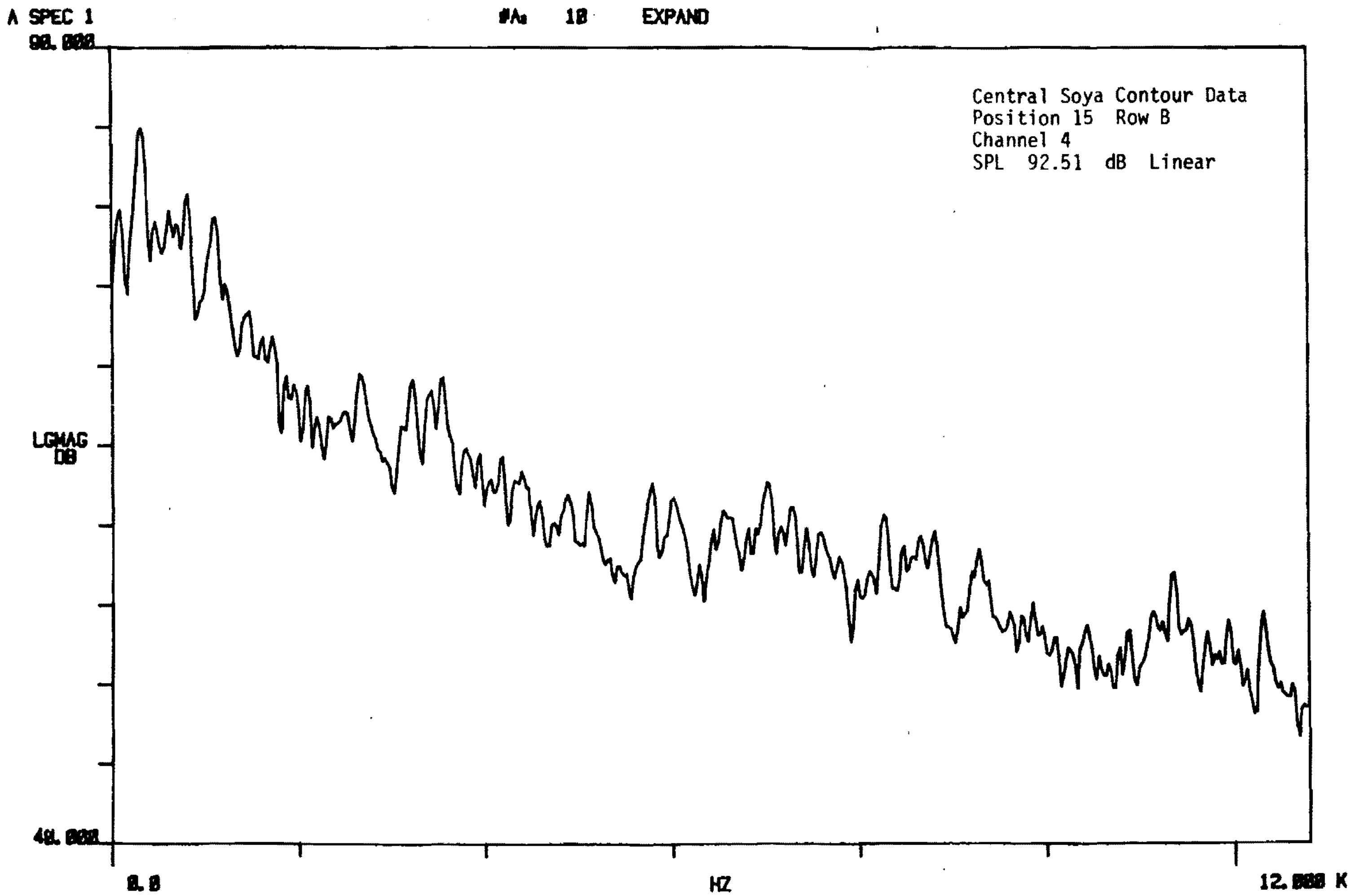

A SPEC 1
#A: 10
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 15 Row B
Channel 4
SPL 92.51 dB Linear

Figure 24B

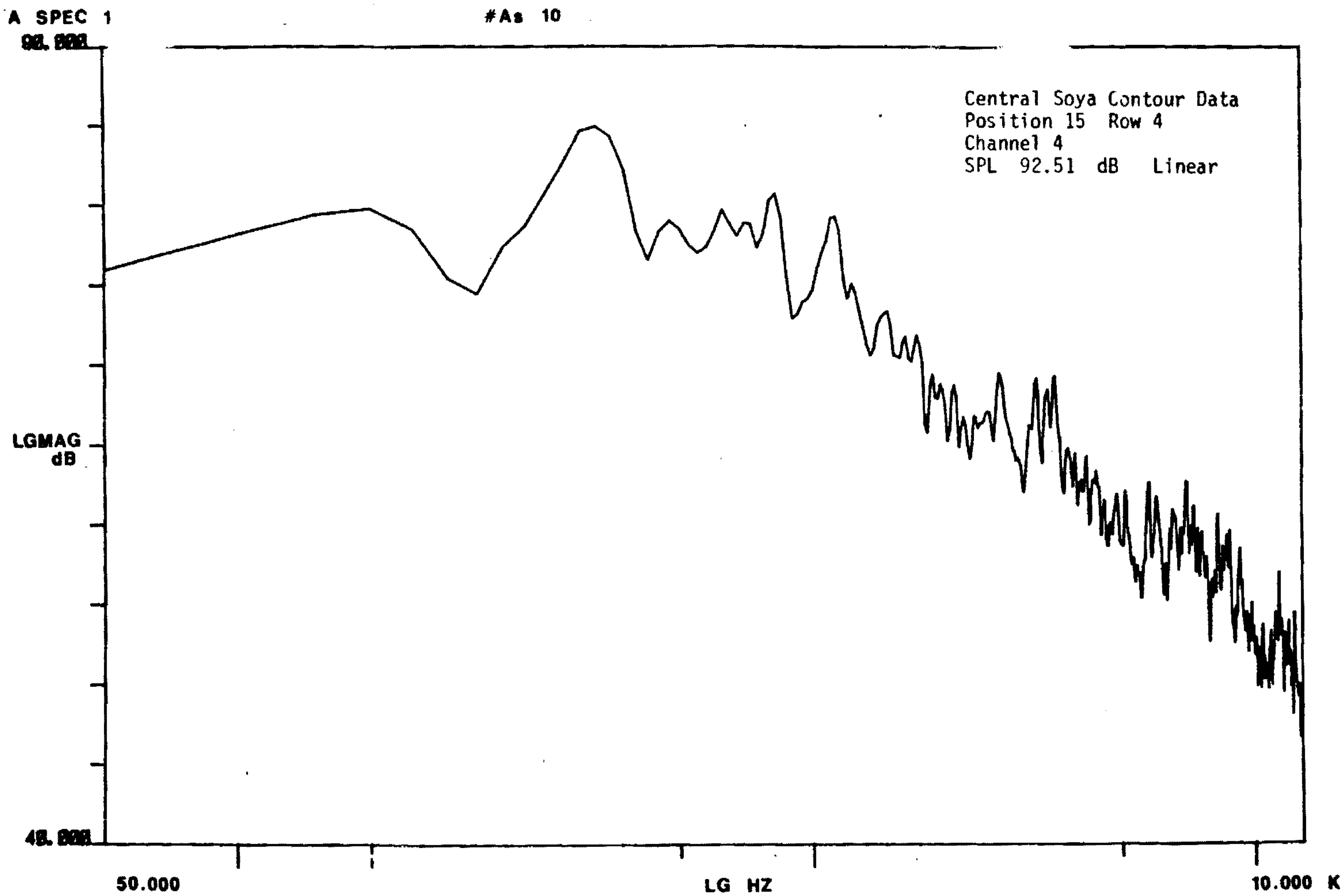

A SPEC 1
#As 10
Central Soya Contour Data
Position 15 Row 4
Channel 4
SPL 92.51 dB Linear
LGMAG
dB
50.000
LG HZ
10.000 K

Figure 25B

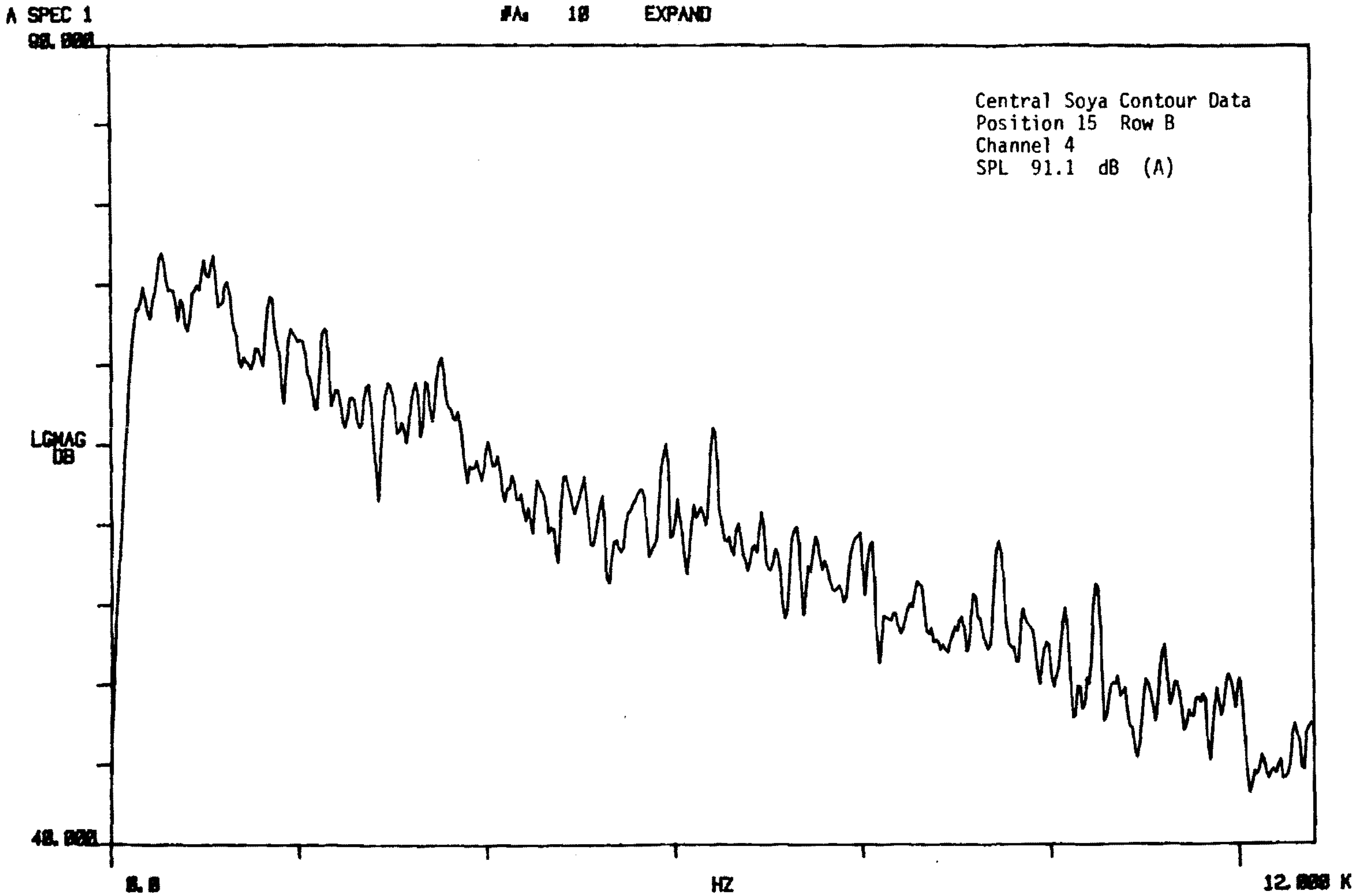

A SPEC 1
#A: 10
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 15 Row B
Channel 4
SPL 91.1 dB (A)

Figure 26B

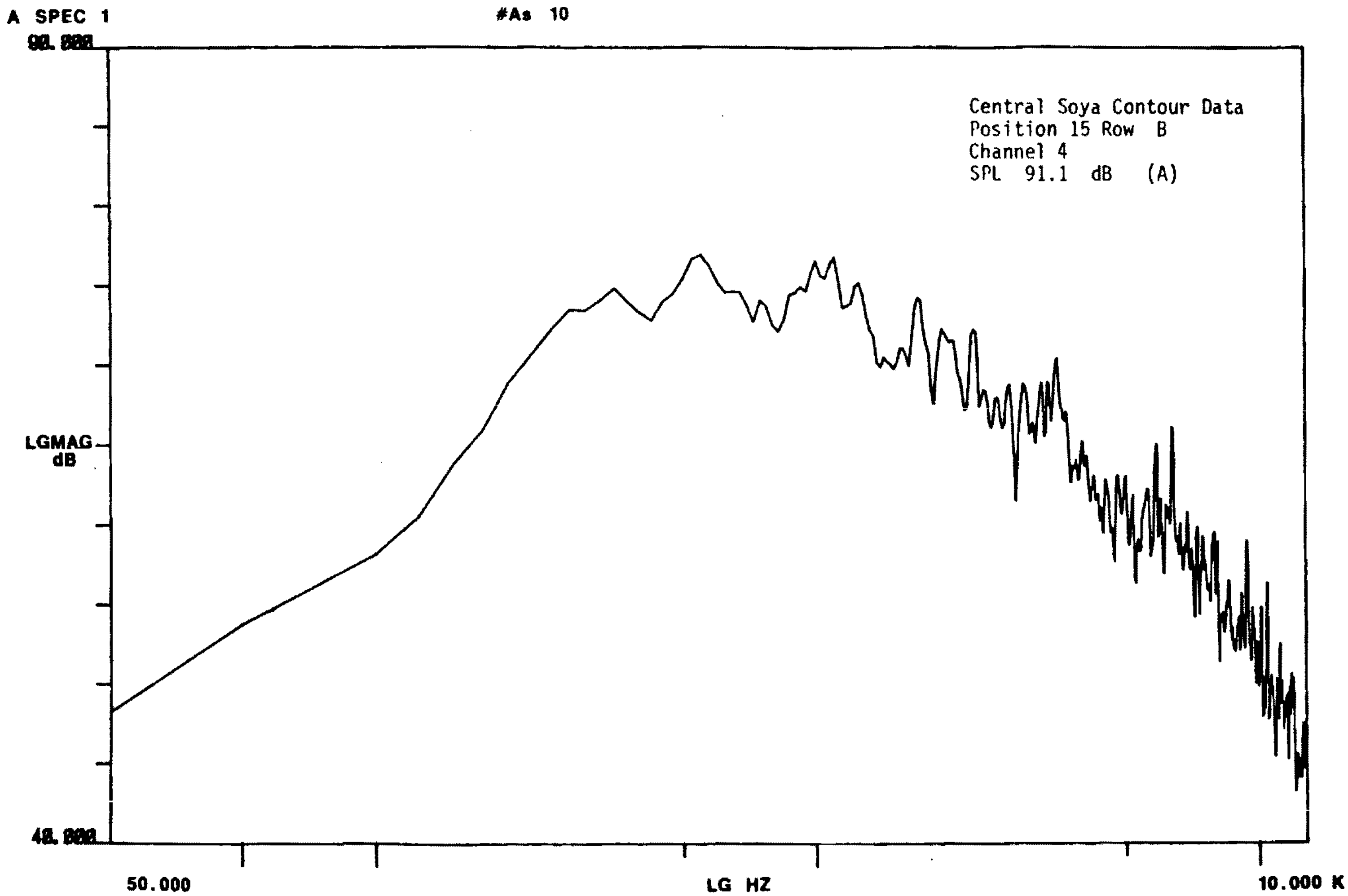

A SPEC 1
#As 10
90.000
LGMAG
dB
40.000
50.000
LG HZ
10.000 K
Central Soya Contour Data
Position 15 Row B
Channel 4
SPL 91.1 dB (A)

Figure 27B

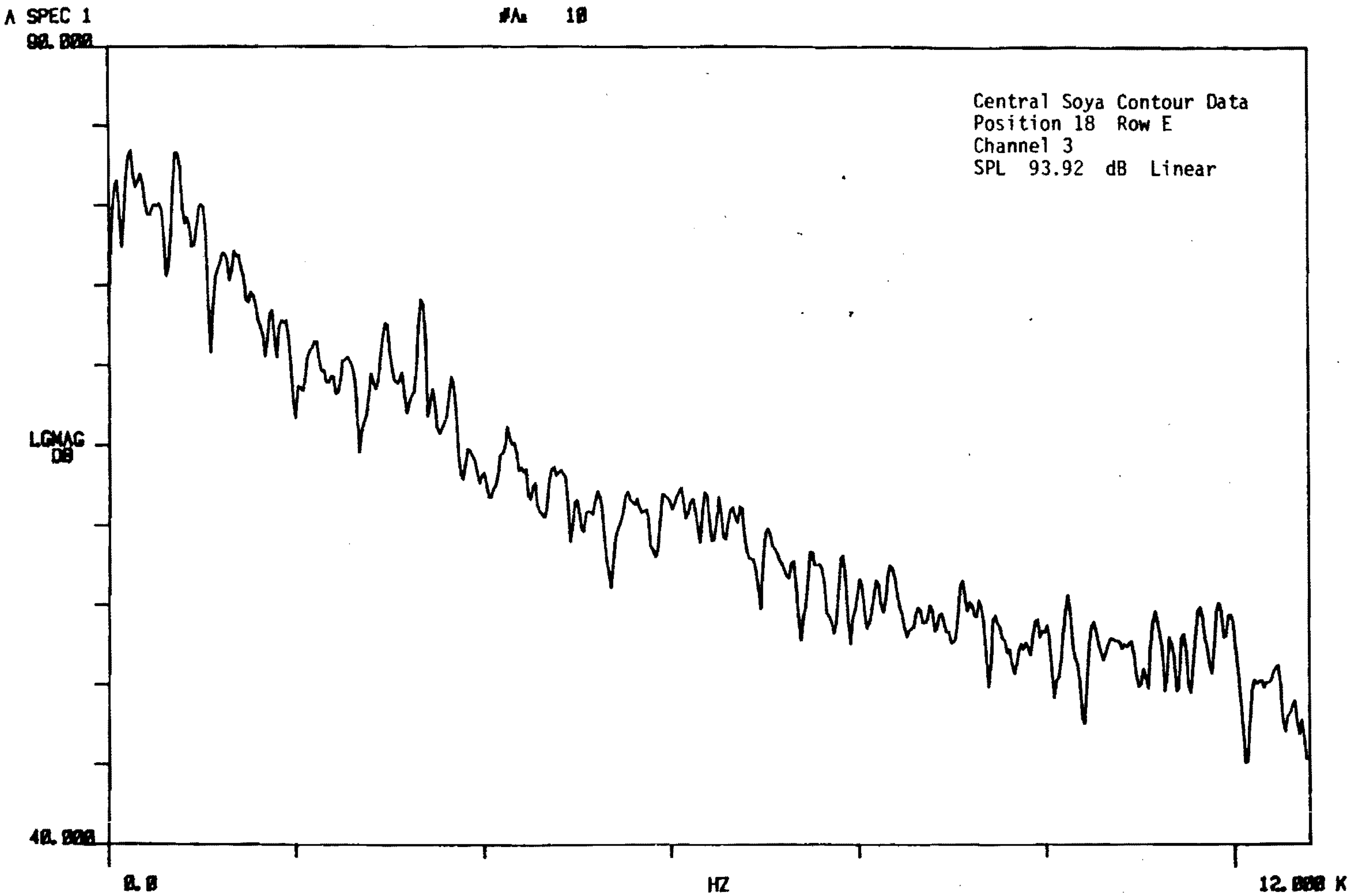

A SPEC 1
#A: 10
90.000
LGMAG DB
40.000
0.0
HZ
12.000 K
Central Soya Contour Data
Position 18 Row E
Channel 3
SPL 93.92 dB Linear

Figure 28B

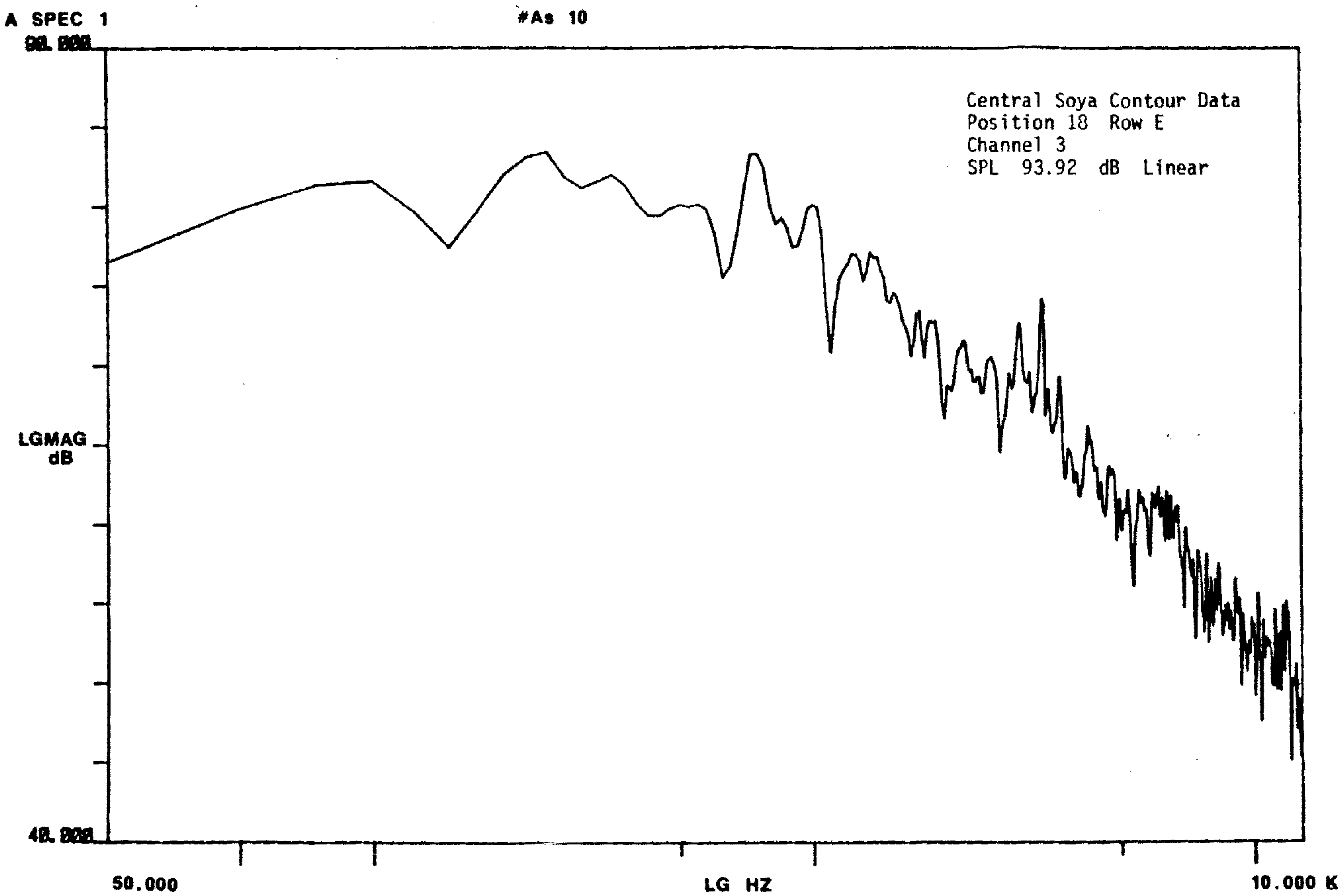

A SPEC 1
#As 10
90.000
LGMAG
dB
40.000
Central Soya Contour Data
Position 18 Row E
Channel 3
SPL 93.92 dB Linear
50.000
LG HZ
10.000 K

Figure 29B

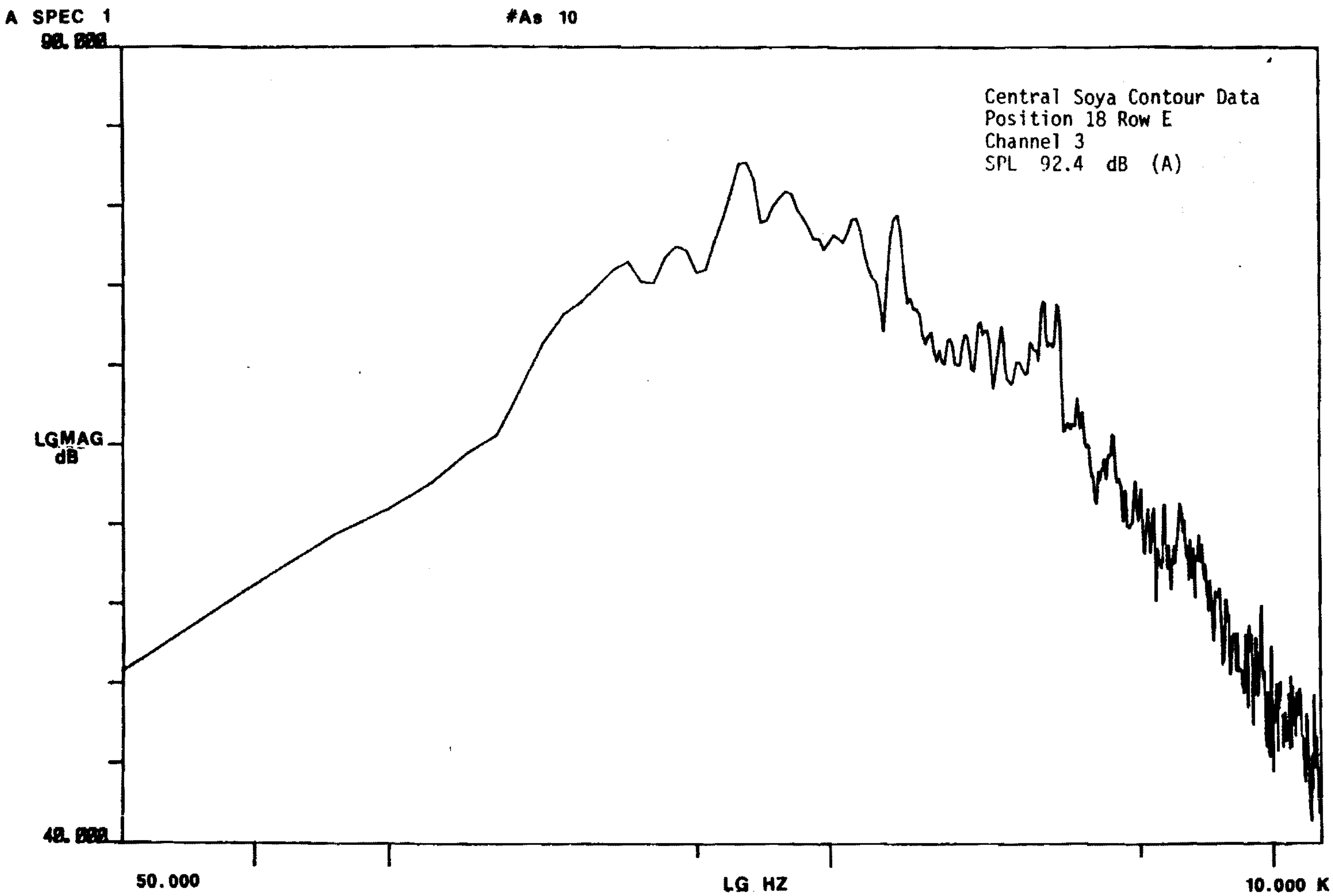

A SPEC 1
#As 10
90.000
LGMAG
dB
40.000
50.000
LG HZ
10.000 K
Central Soya Contour Data
Position 18 Row E
Channel 3
SPL 92.4 dB (A)

Figure 30B

Noise Contour

Data

TIME AVERAGES

CENTRAL SOYA PLANT

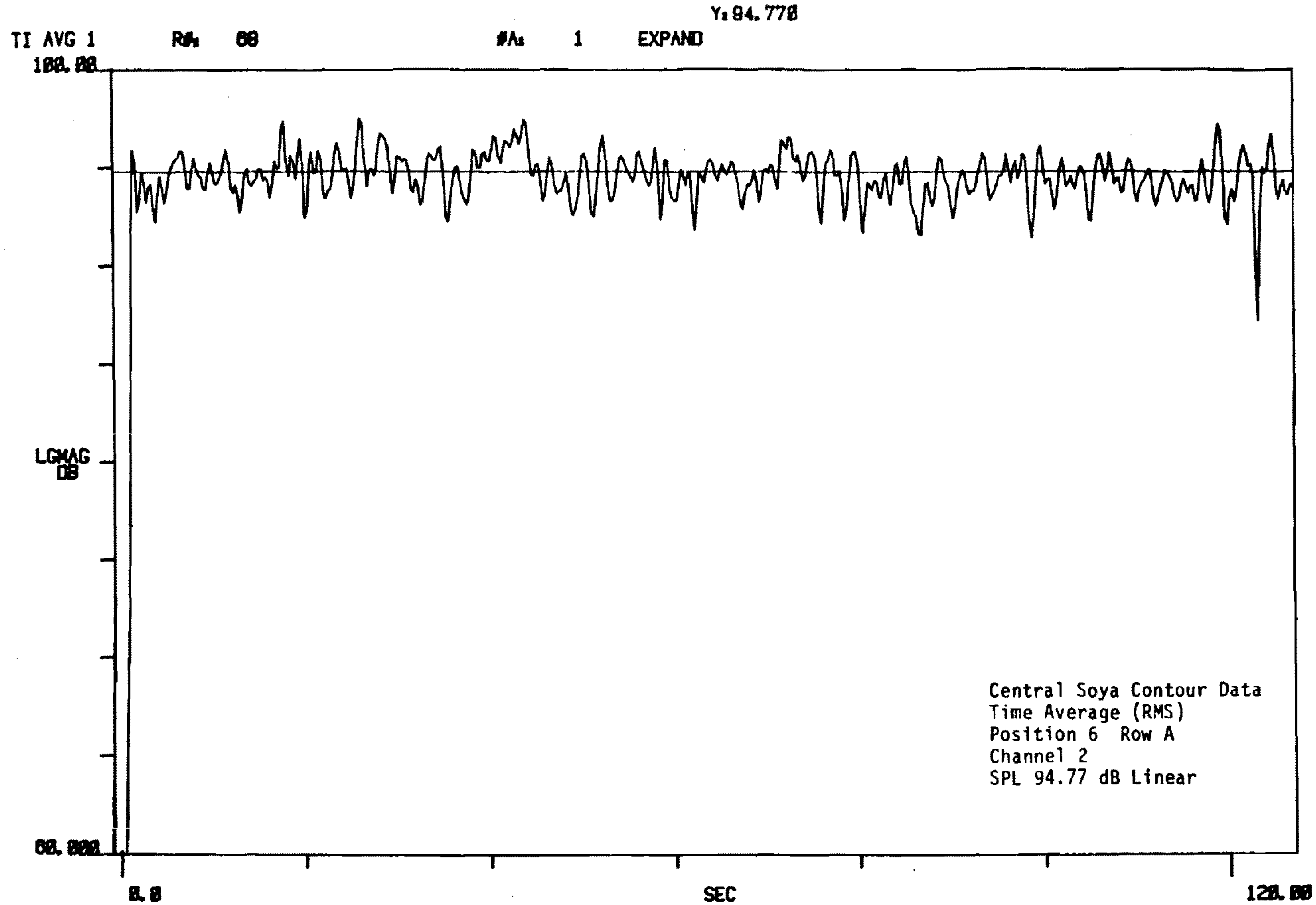

Y: 94.770
TI AVG 1 R#: 60 #A: 1 EXPAND
100.00
LGMAG DB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 6 Row A
Channel 2
SPL 94.77 dB Linear

Figure 31B

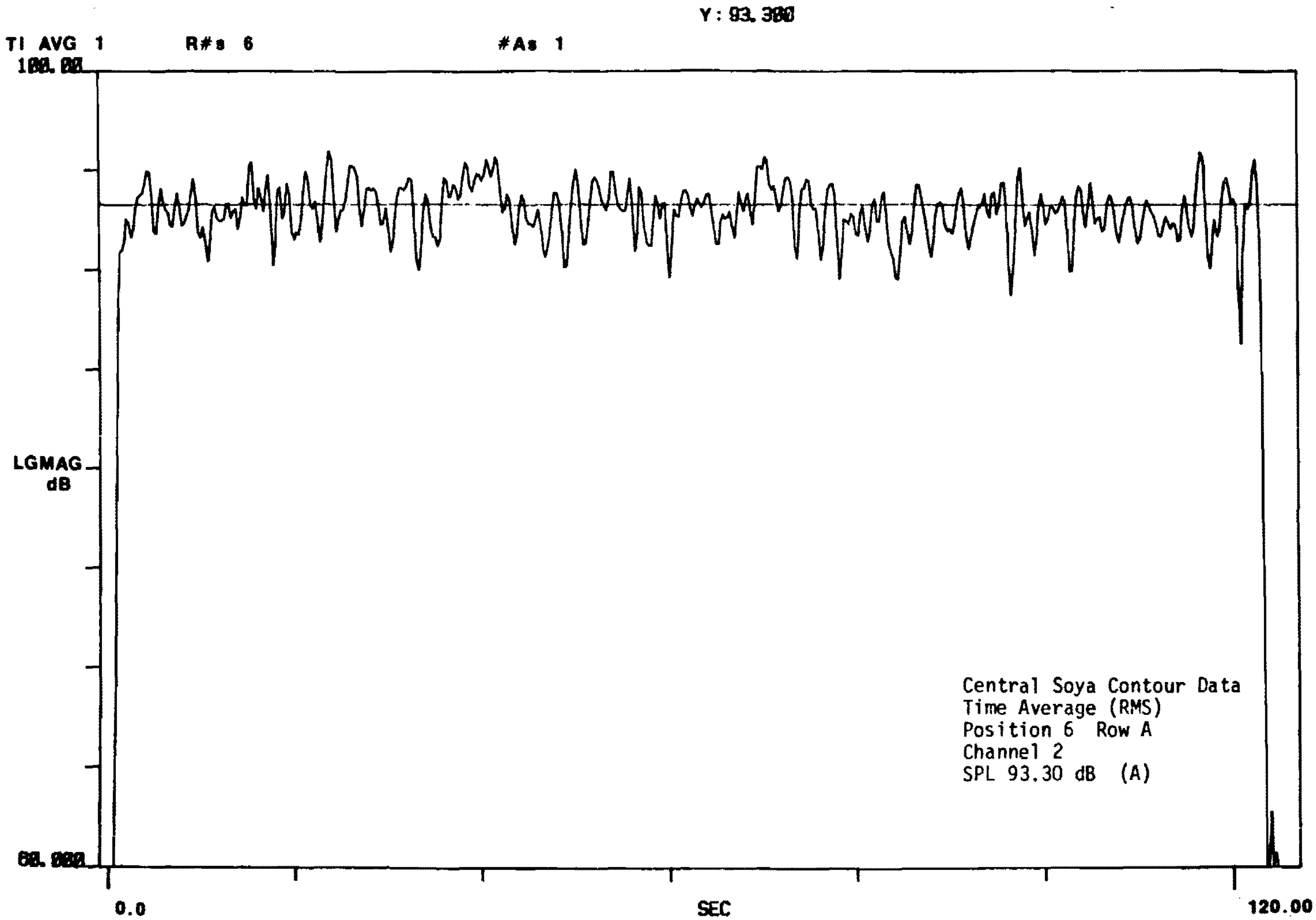
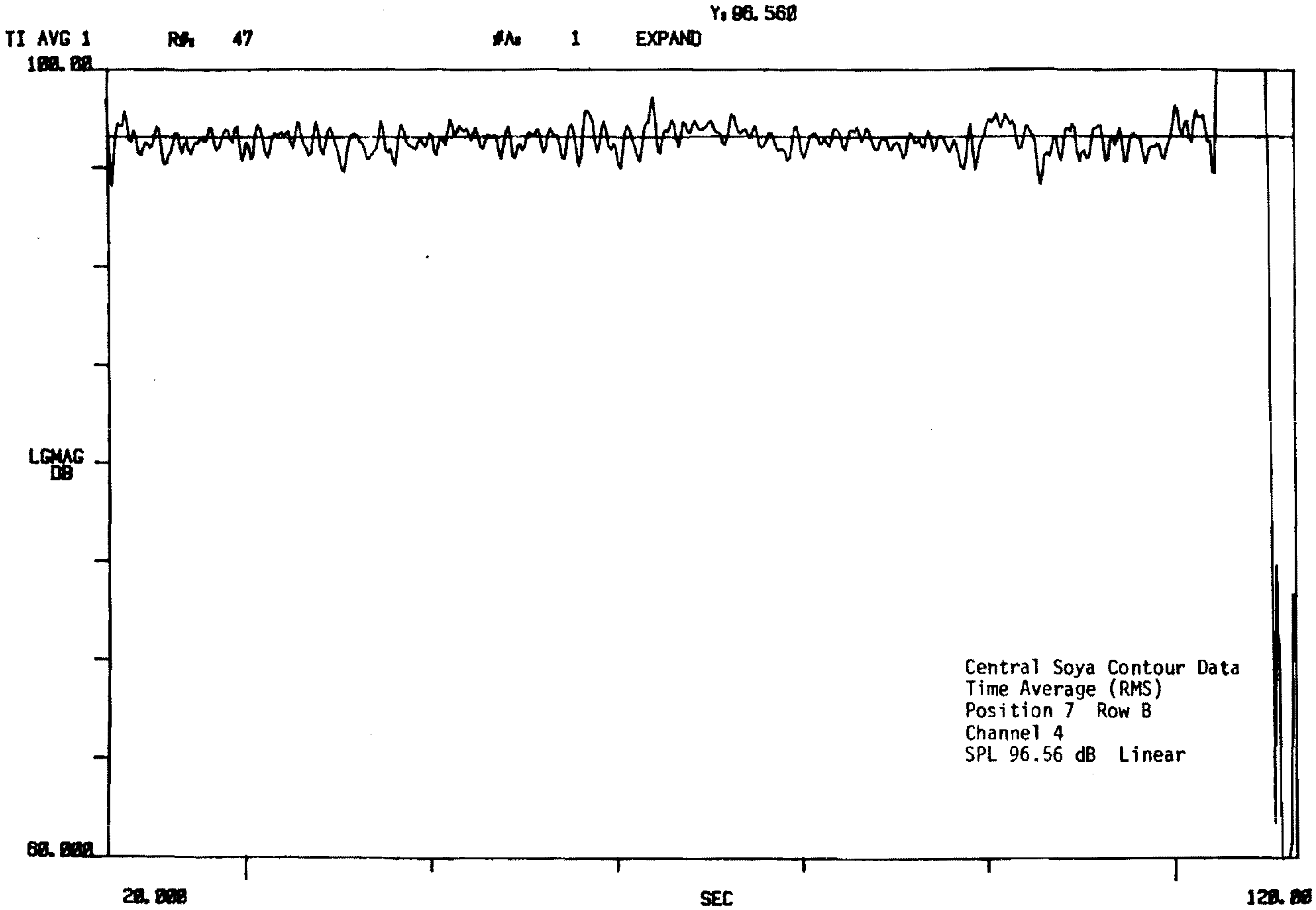

Y: 93.300
TI AVG 1
R#s 6
#As 1
100.00
LGMAG
dB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 6 Row A
Channel 2
SPL 93.30 dB (A)

Figure 32B



Y: 96.560
TI AVG 1
R#: 47
#A: 1
EXPAND
100.00
LGMAG
DB
60.000
20.000
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 7 Row B
Channel 4
SPL 96.56 dB Linear

Figure 33B

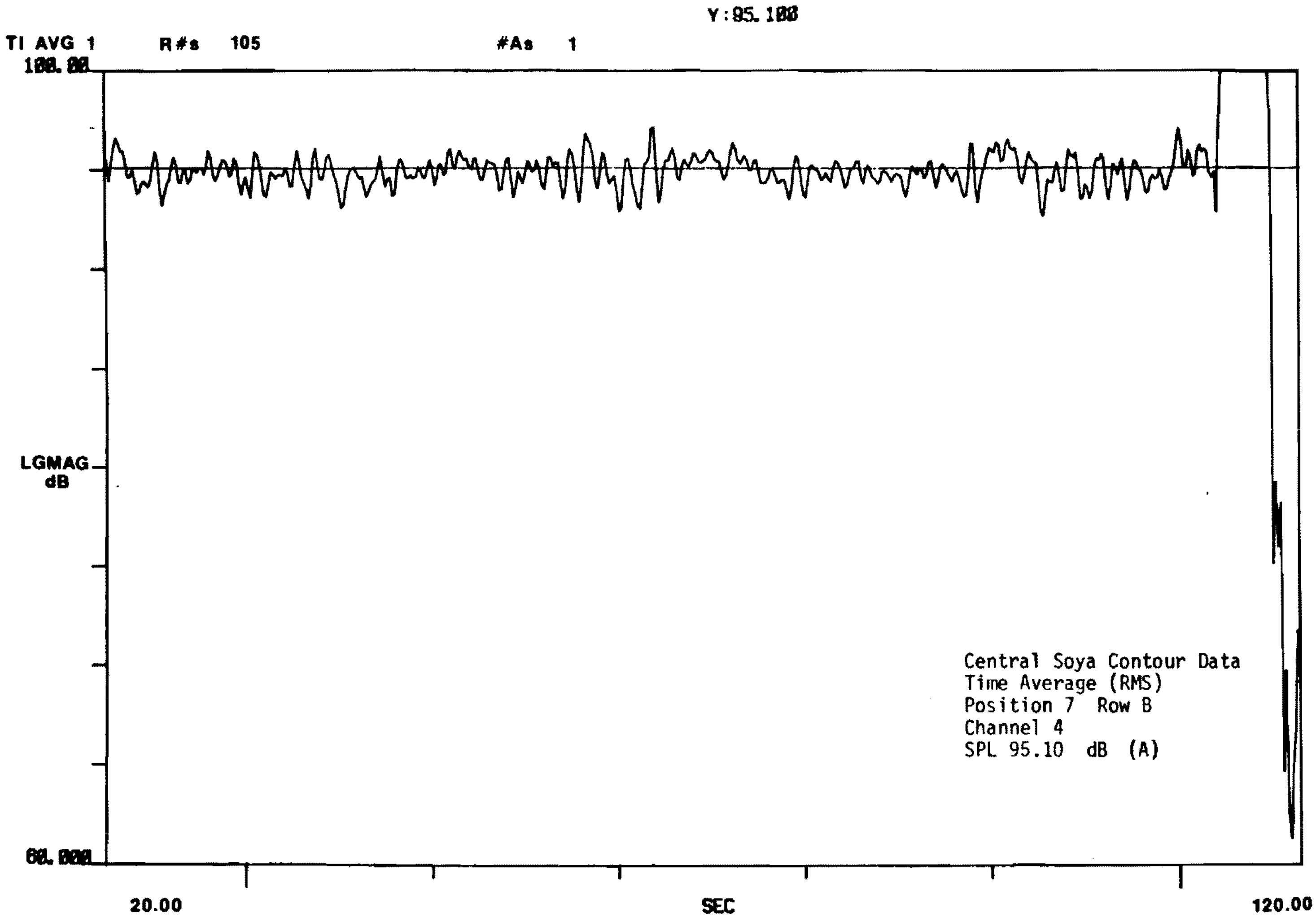

Y:95.100
TI AVG 1
R#s 105
#As 1
100.00
LGMAG
dB
60.000
20.00
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 7 Row B
Channel 4
SPL 95.10 dB (A)

Figure 34B

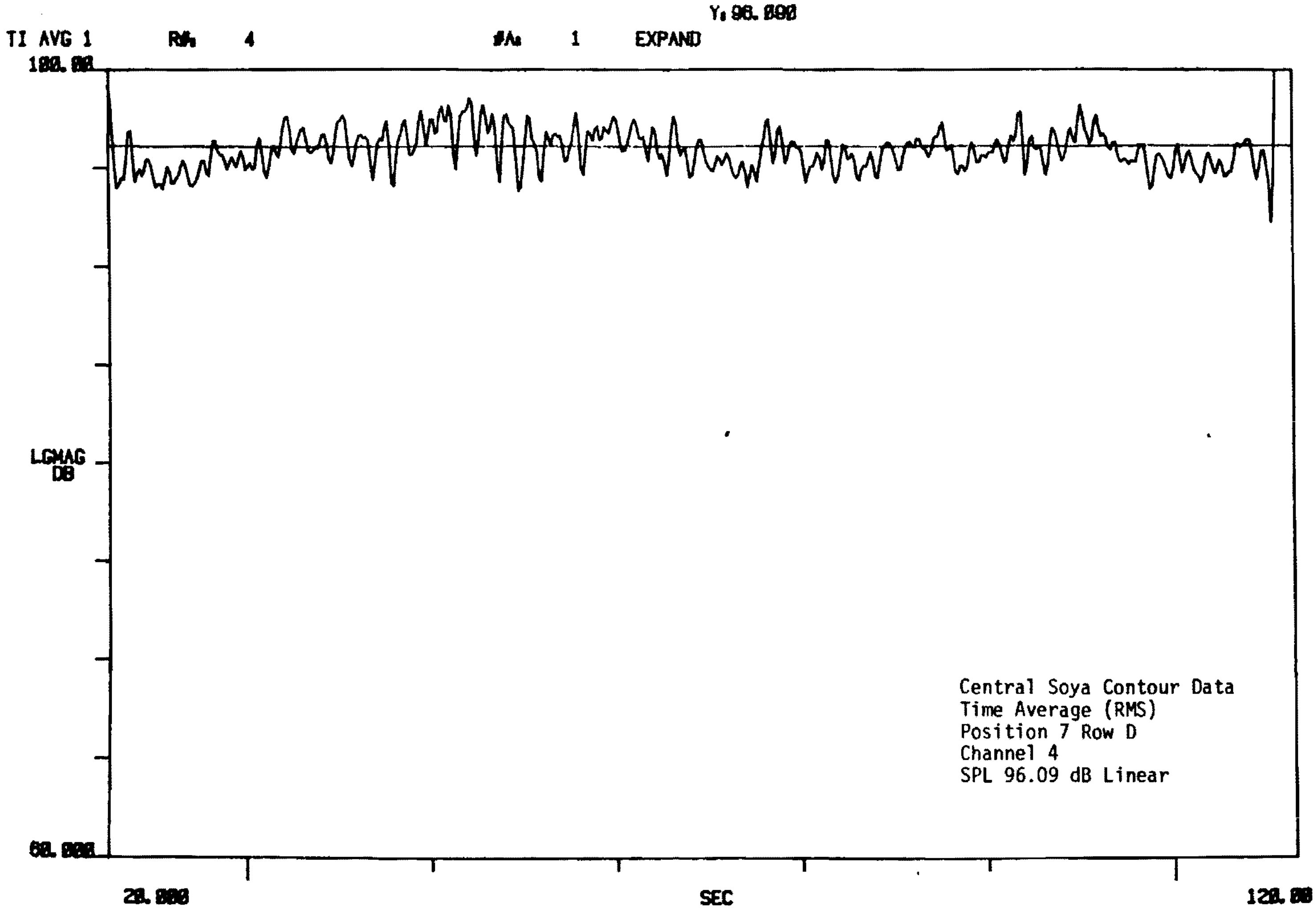

Y: 96.000
TI AVG 1
RM: 4
#A: 1
EXPAND
100.00
LGMAG
DB
60.000
20.000
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 7 Row D
Channel 4
SPL 96.09 dB Linear

Figure 35B

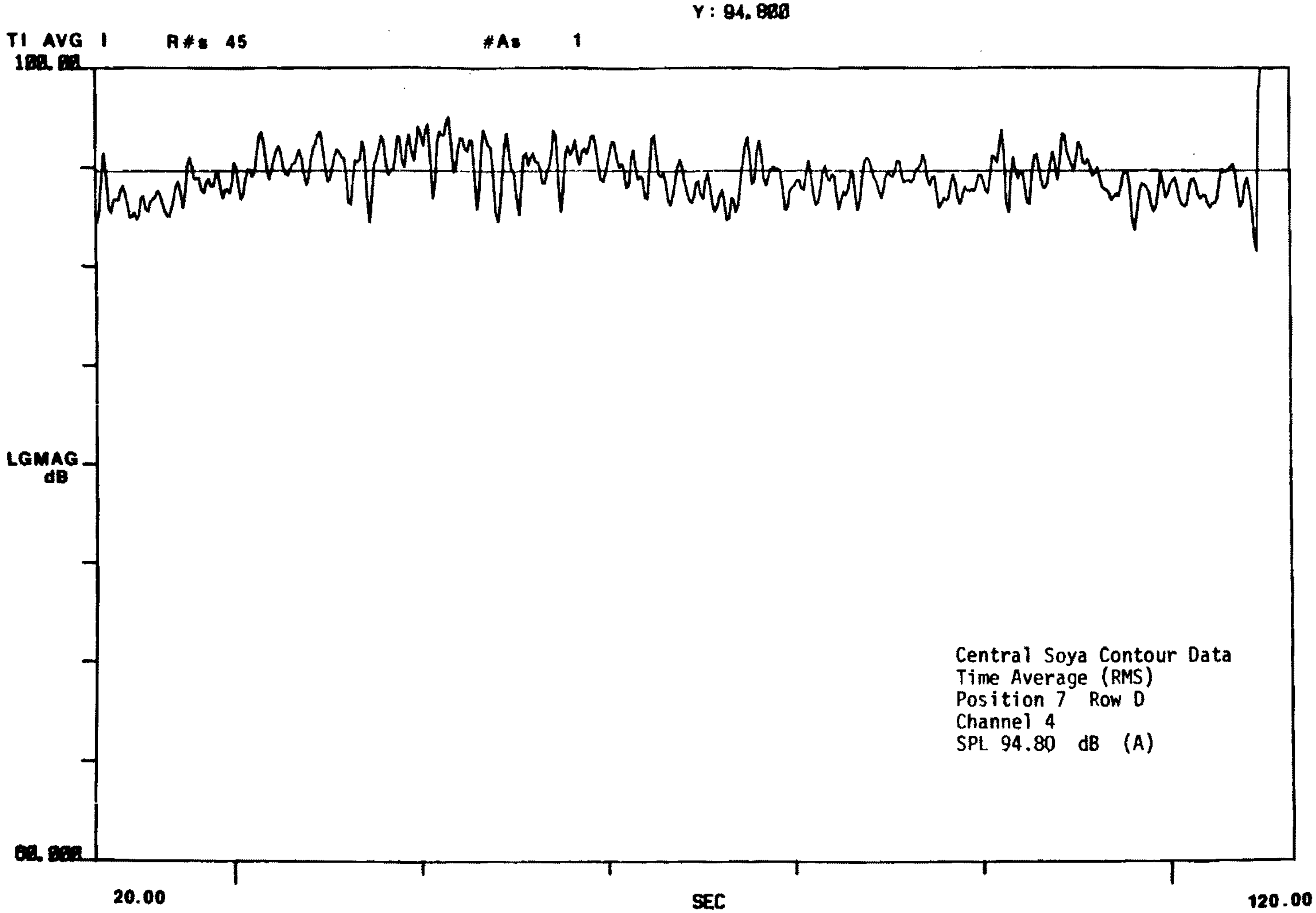
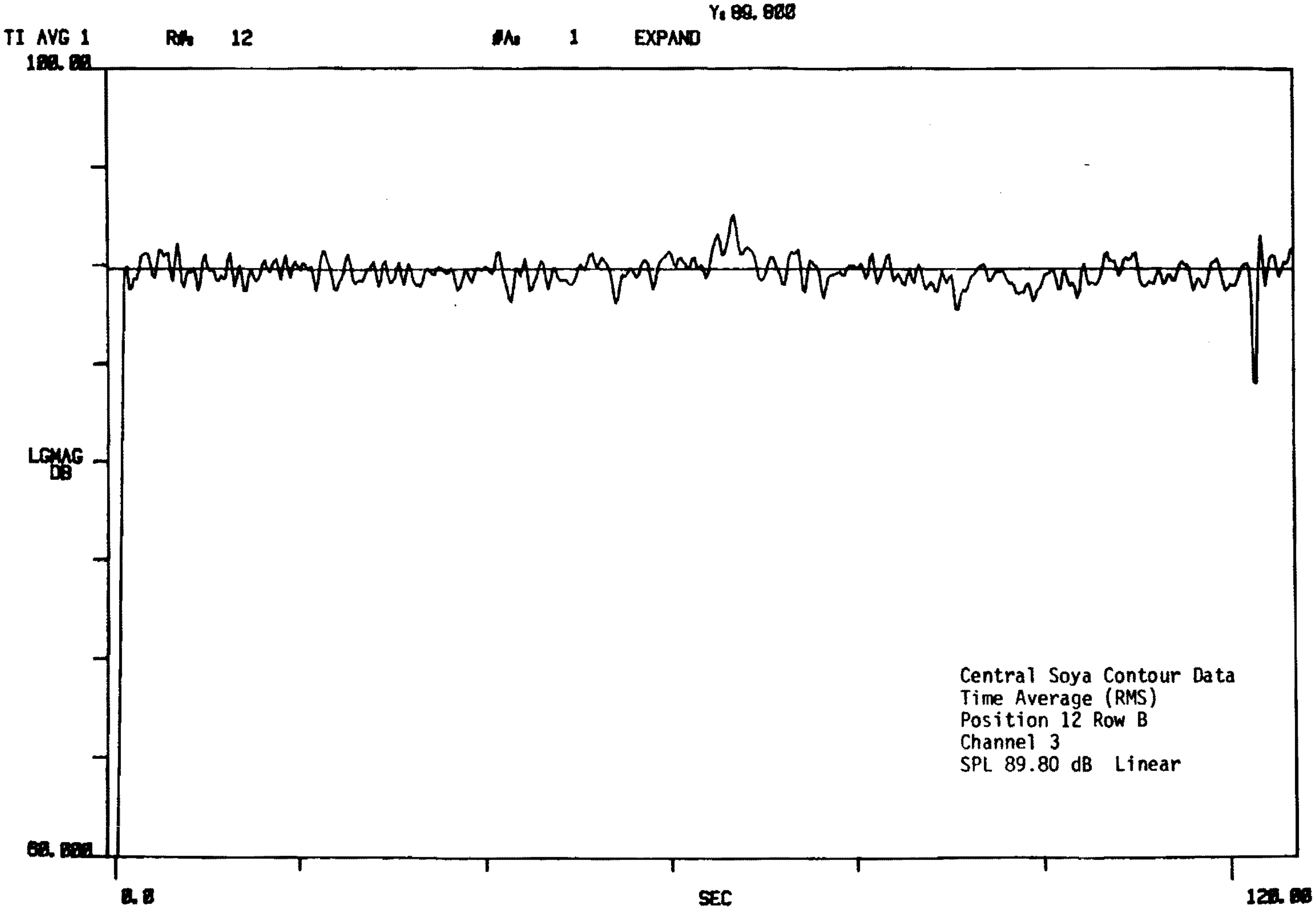

Y: 94.800
TI AVG I R#s 45 #As 1
100.00
LGMAG dB
80.000
20.00
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 7 Row D
Channel 4
SPL 94.80 dB (A)

Figure 36B



TI AVG 1 R#: 12 #A: 1 EXPAND
Y: 89.800
100.00
LGMAG DB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 12 Row B
Channel 3
SPL 89.80 dB Linear

Figure 37B

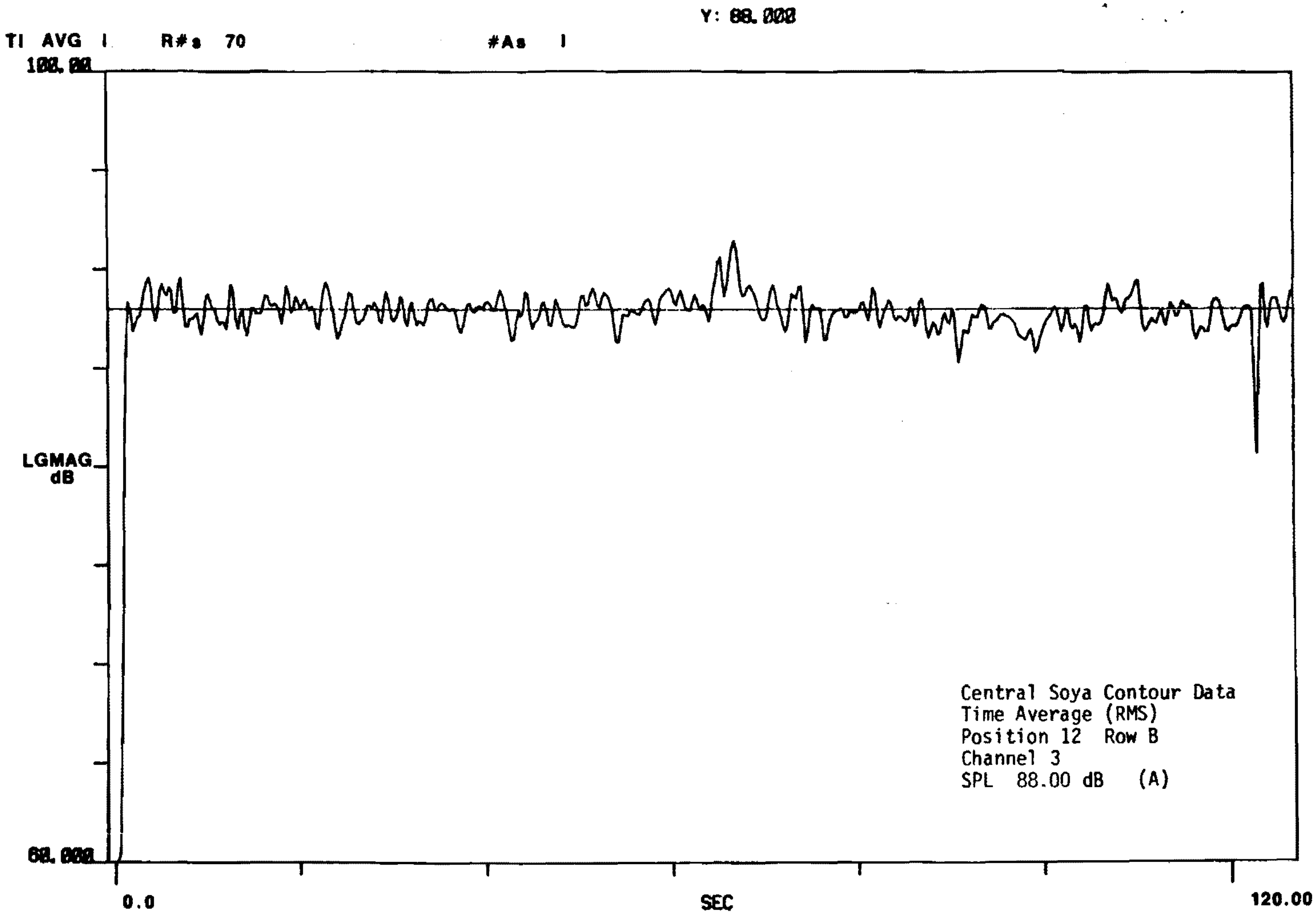

Y: 88.000
TI AVG I R#s 70 #As I
100.00
LGMAG dB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 12 Row B
Channel 3
SPL 88.00 dB (A)

Figure 38B

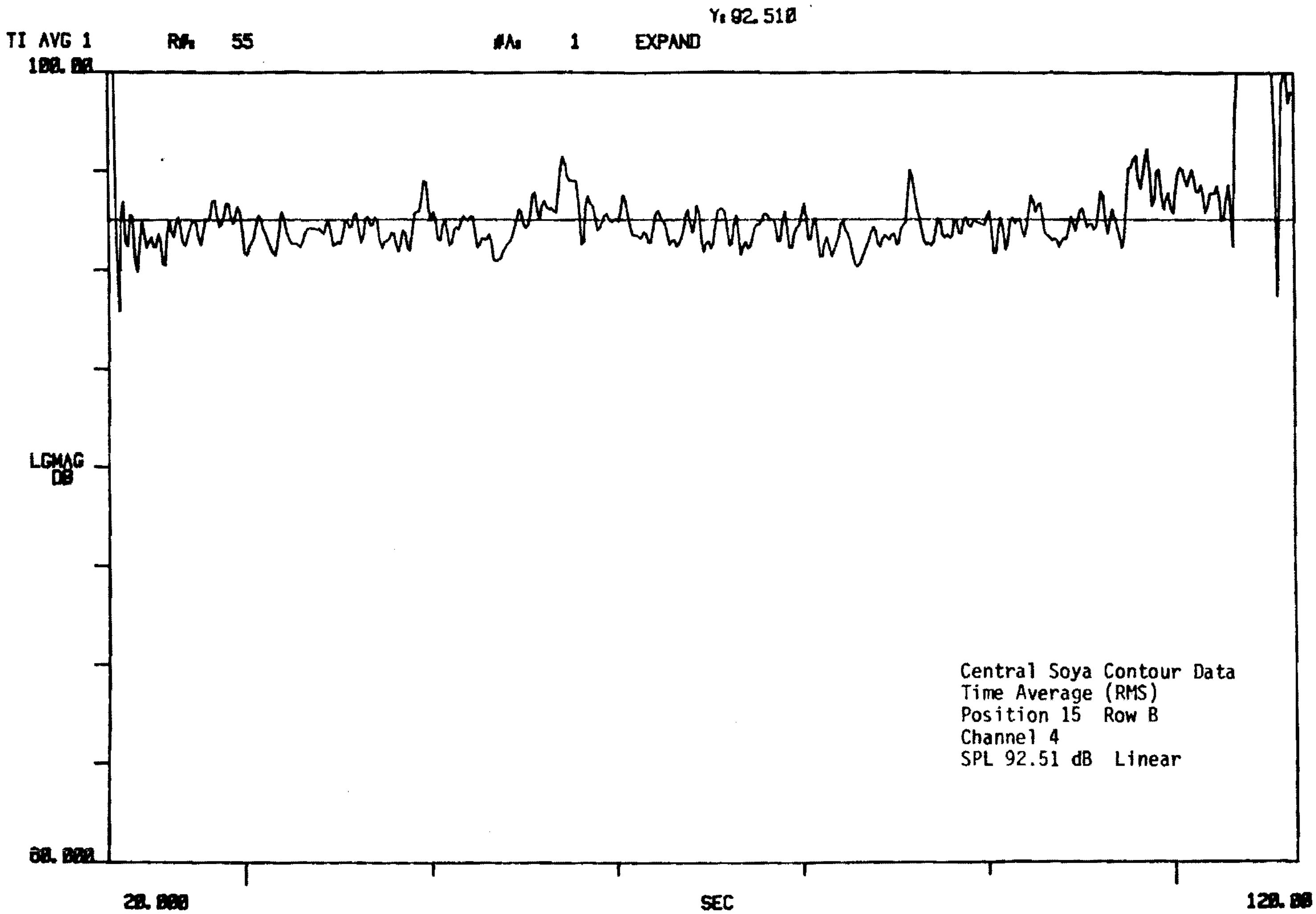

Y: 92.510
TI AVG 1
R#: 55
#A: 1
EXPAND
100.00
LGMAG
DB
60.000
20.000
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 15 Row B
Channel 4
SPL 92.51 dB Linear

Figure 39B

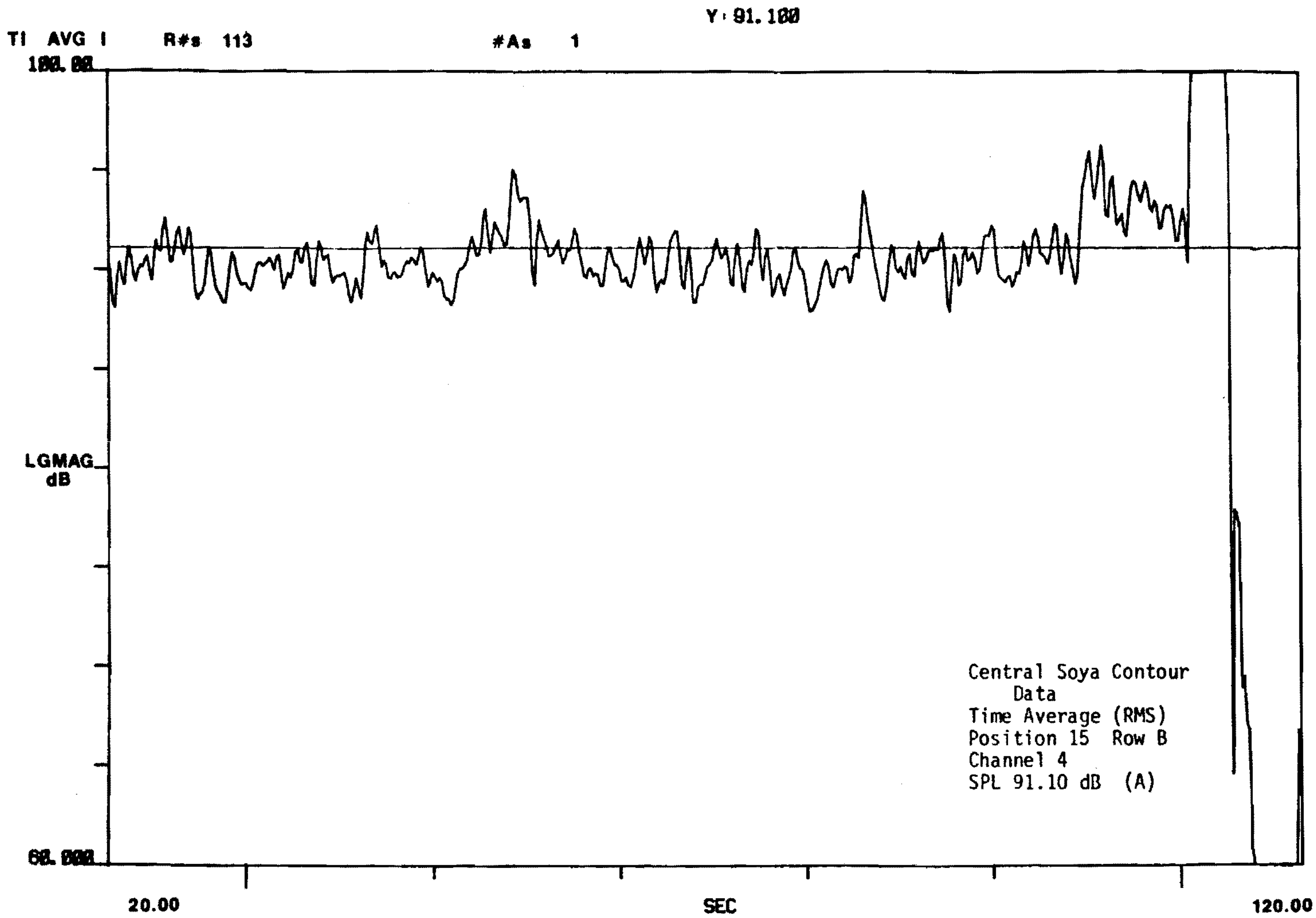

Y: 91.100
TI AVG I R#s 113 #As 1
100.00
LGMAG dB
60.000
20.00
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 15 Row B
Channel 4
SPL 91.10 dB (A)

Figure 40B

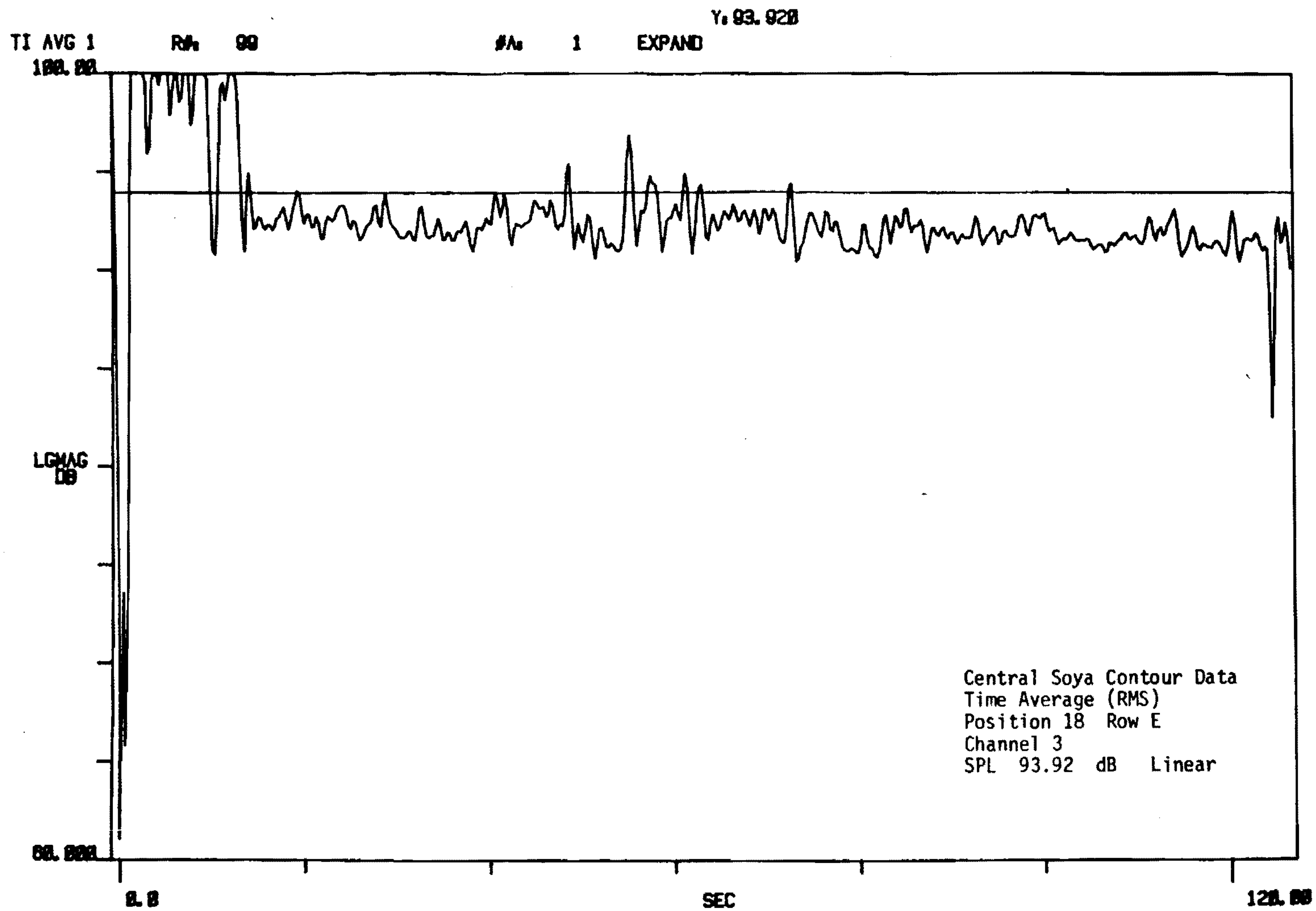

Y: 93.920
TI AVG 1
R#: 99
#A: 1
EXPAND
100.00
LGMAG DB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 18 Row E
Channel 3
SPL 93.92 dB Linear

Figure 41B

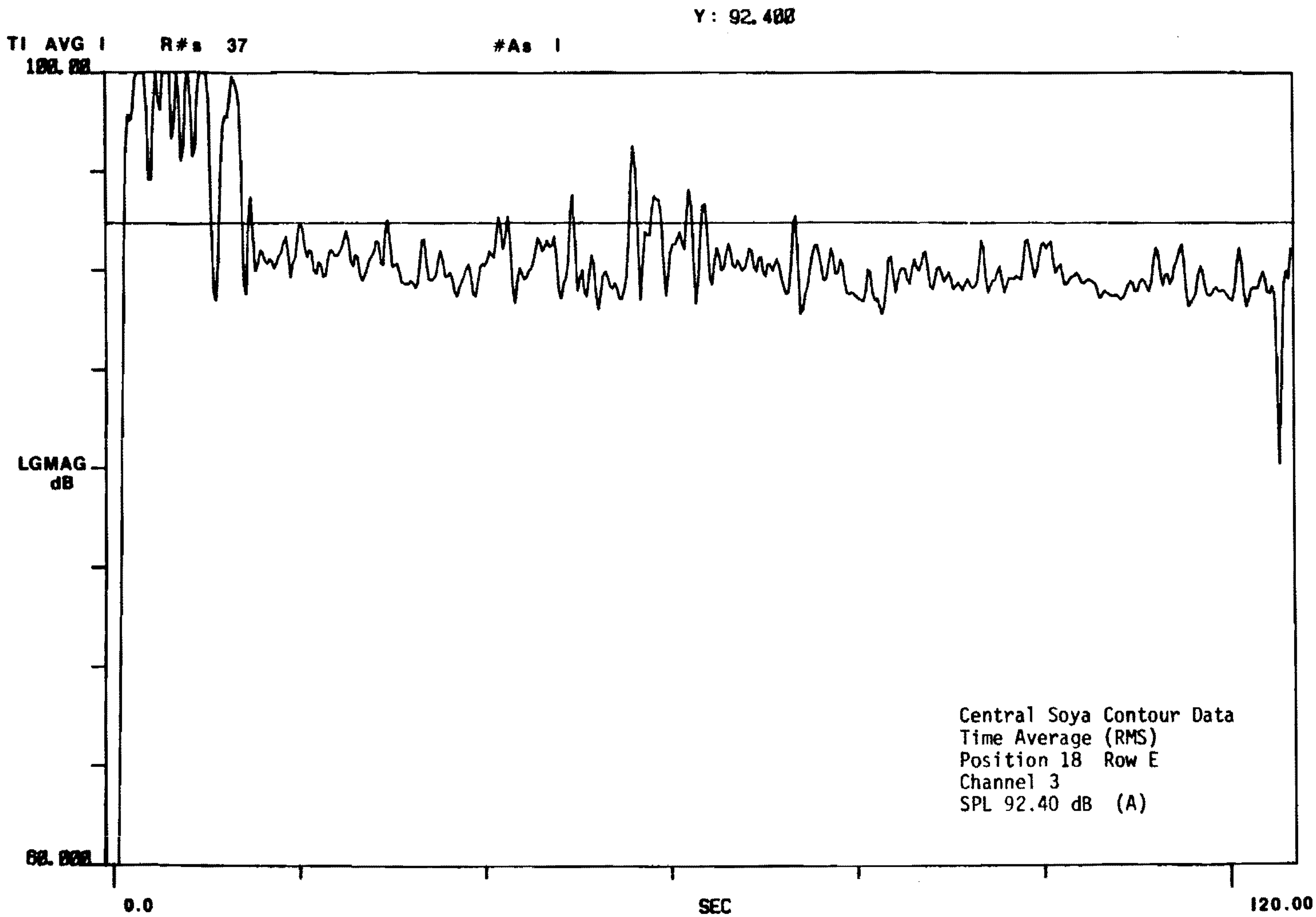

Y: 92.400
TI AVG I
R#s 37
#As I
100.00
LGMAG
dB
60.000
0.0
SEC
120.00
Central Soya Contour Data
Time Average (RMS)
Position 18 Row E
Channel 3
SPL 92.40 dB (A)

Figure 42B

Noise Contour

Data

FREQUENCY SPECTRA TIP TOP PLANT

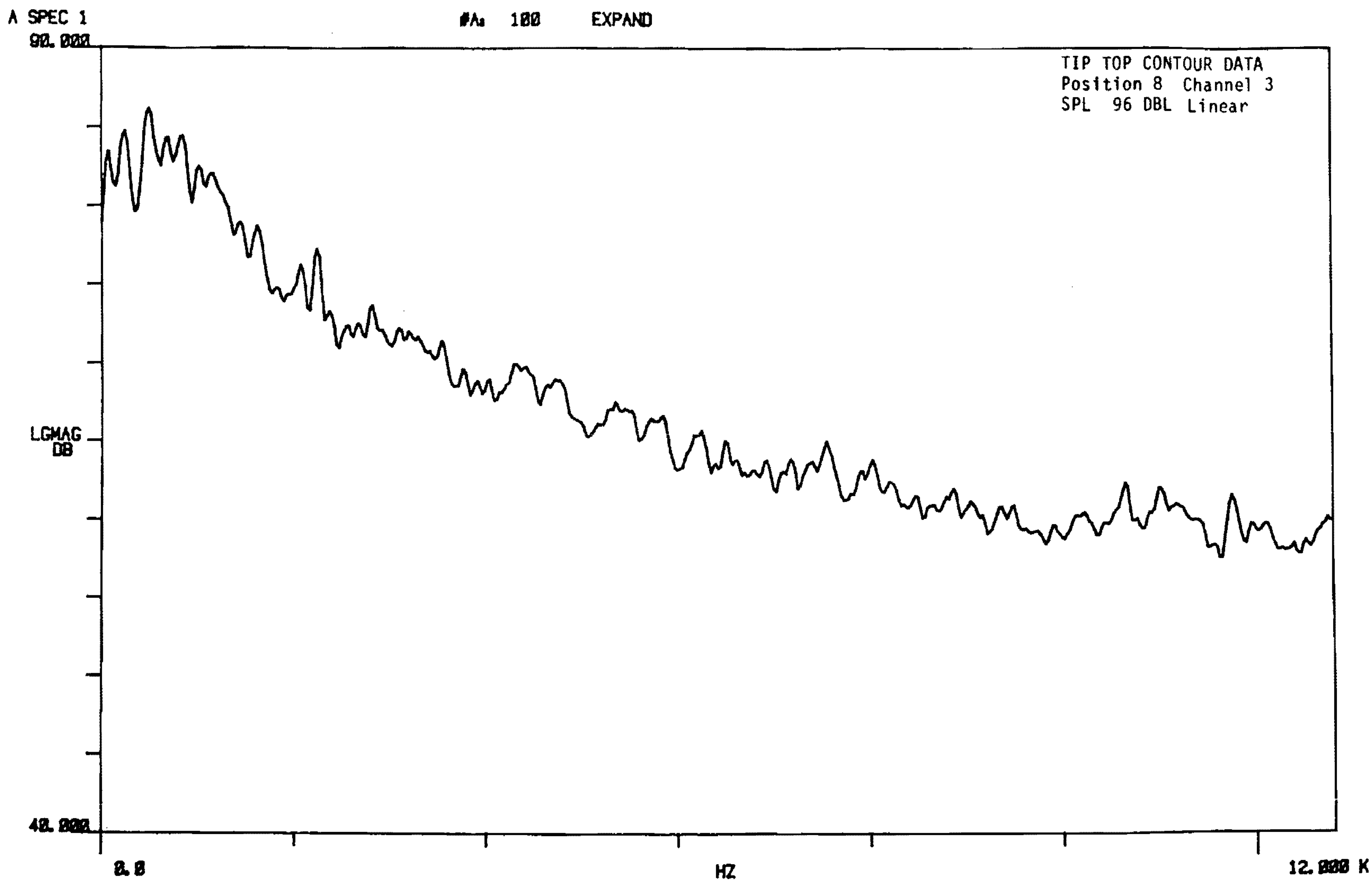

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 8 Channel 3
SPL 96 DBL Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 43B

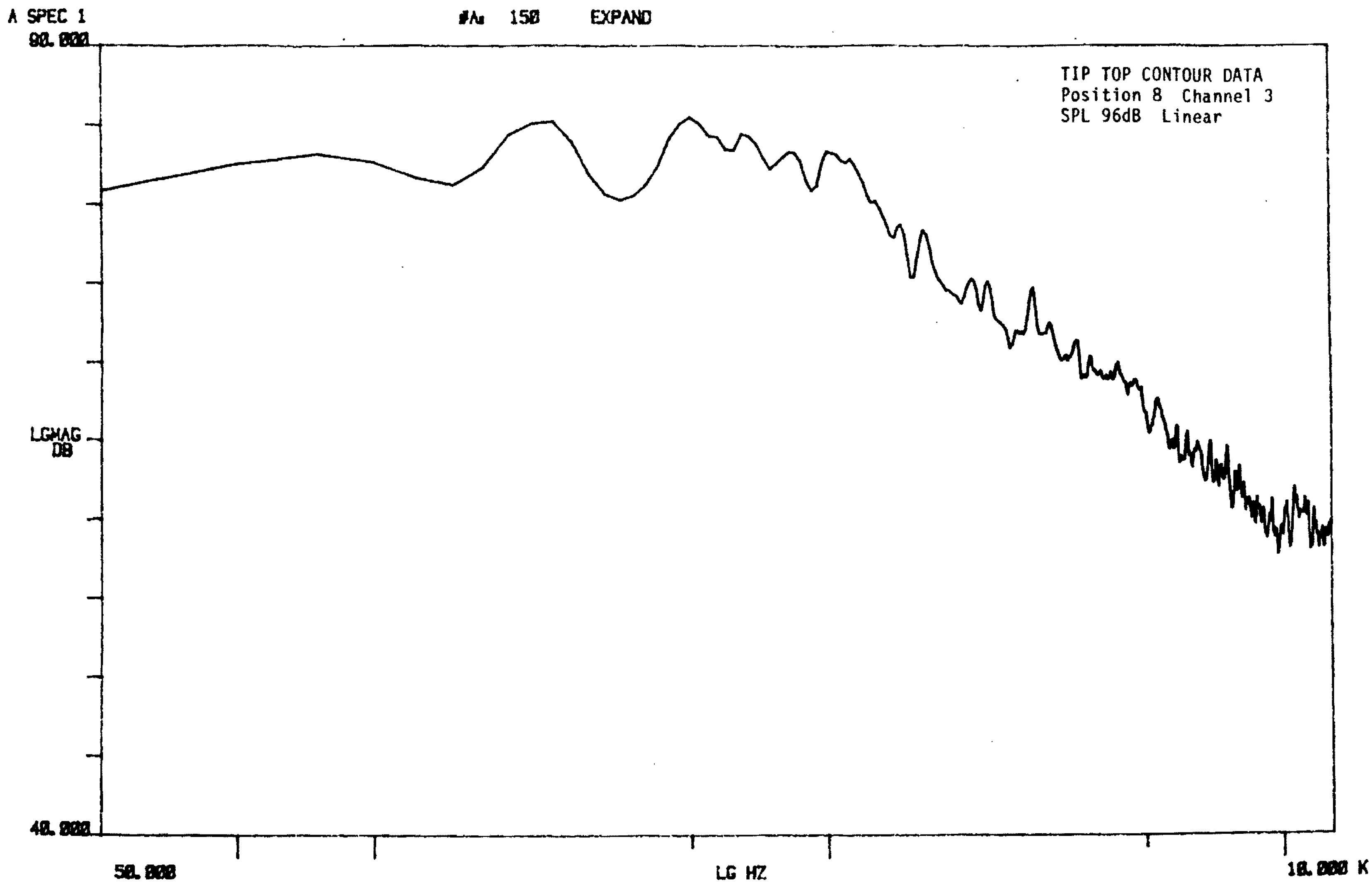
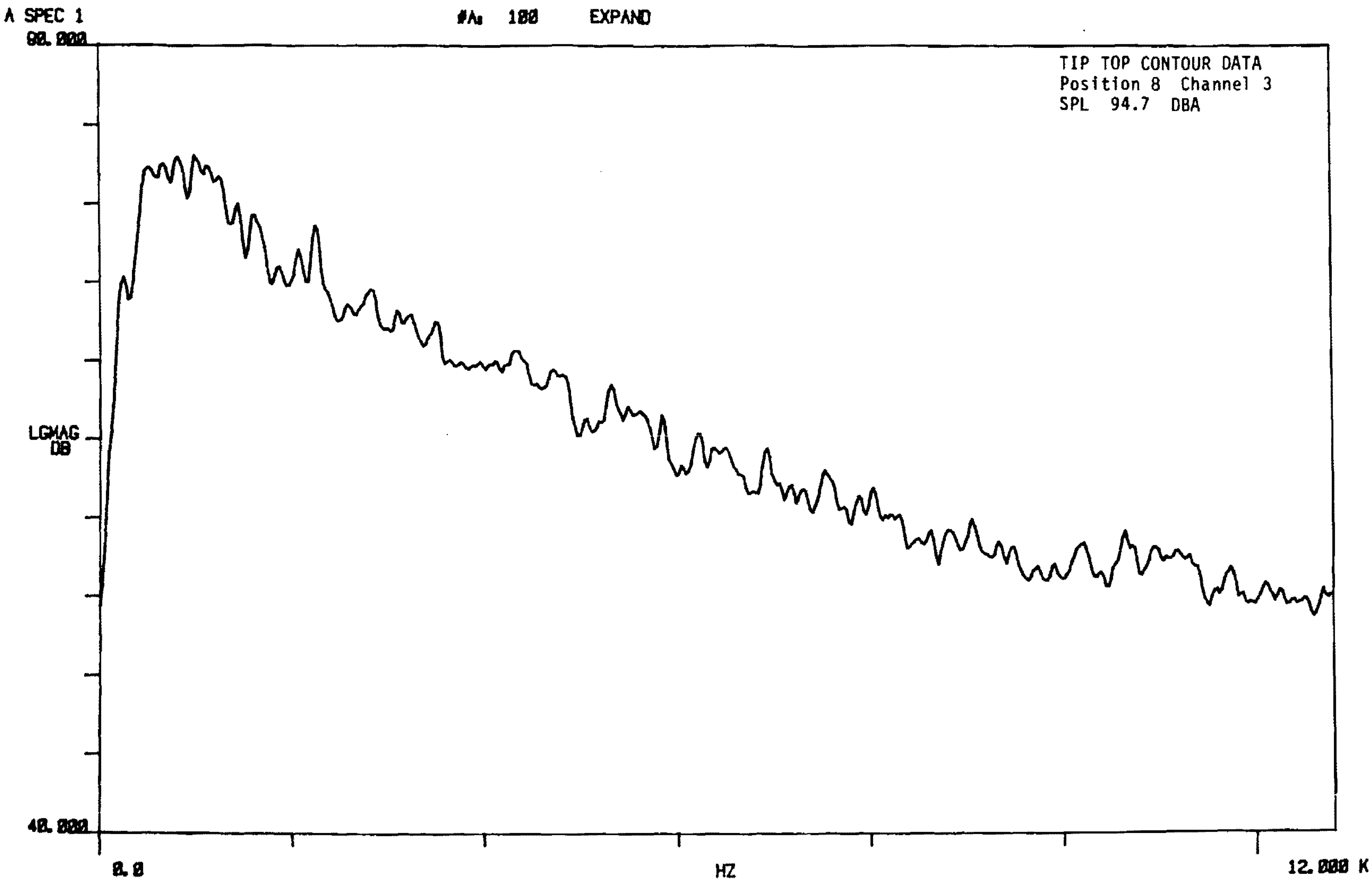

A SPEC 1
#A: 150
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 8 Channel 3
SPL 96dB Linear
LGMAG DB
40.000
50.000
LG HZ
10.000 K

Figure 44B



A SPEC 1
#A: 100 EXPAND
TIP TOP CONTOUR DATA
Position 8 Channel 3
SPL 94.7 DBA
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 45B

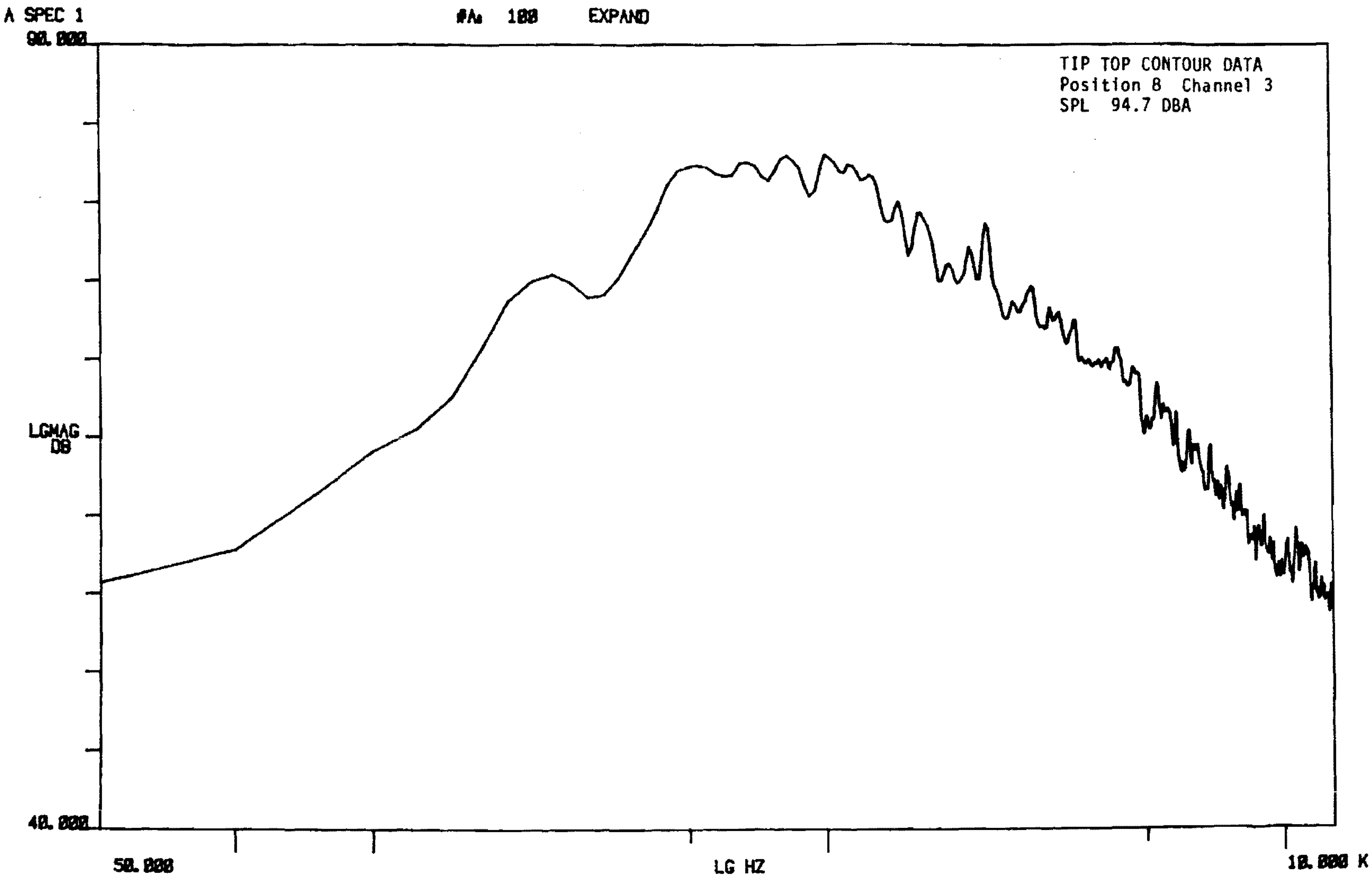

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 8 Channel 3
SPL 94.7 DBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 46B

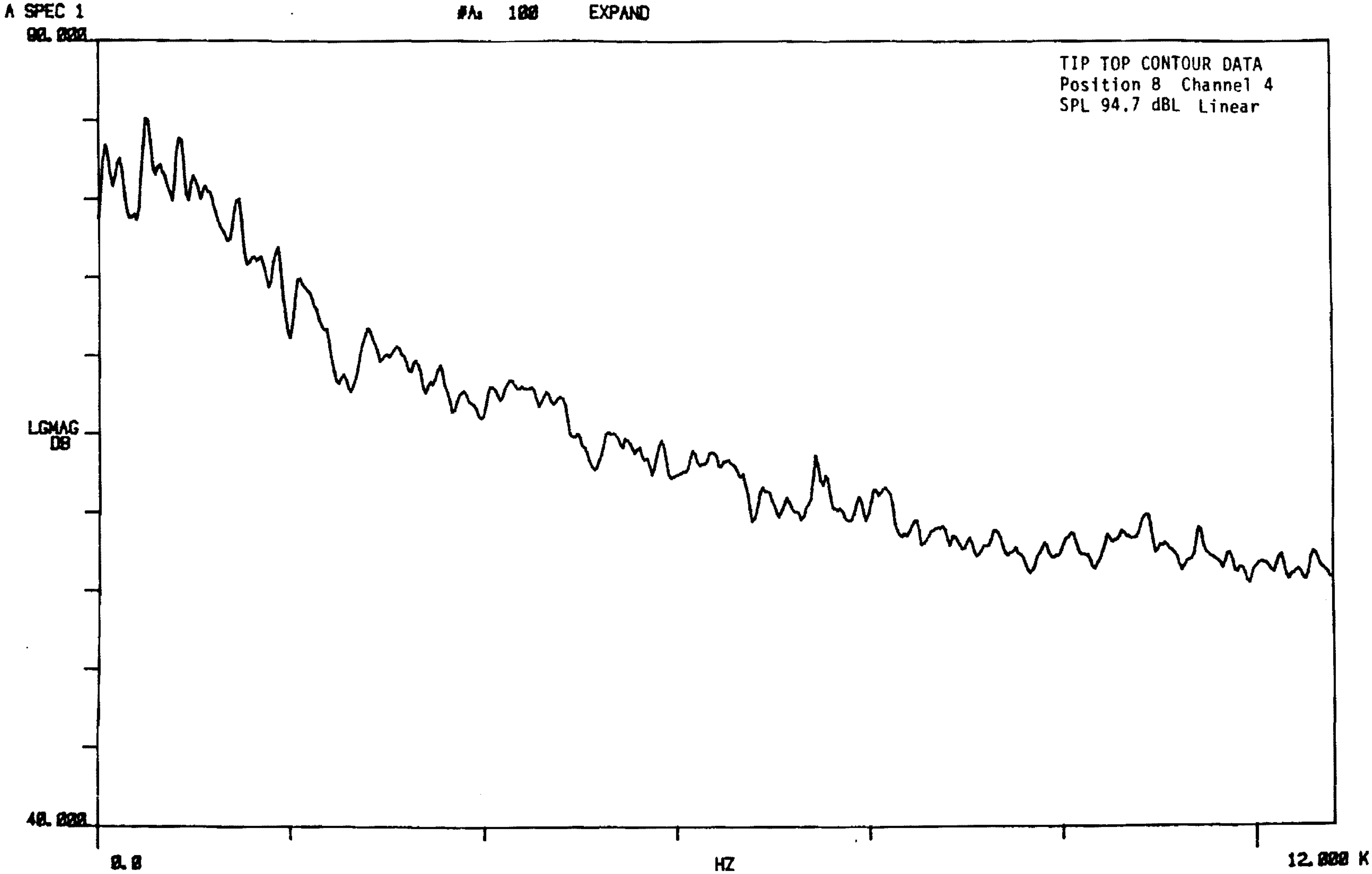
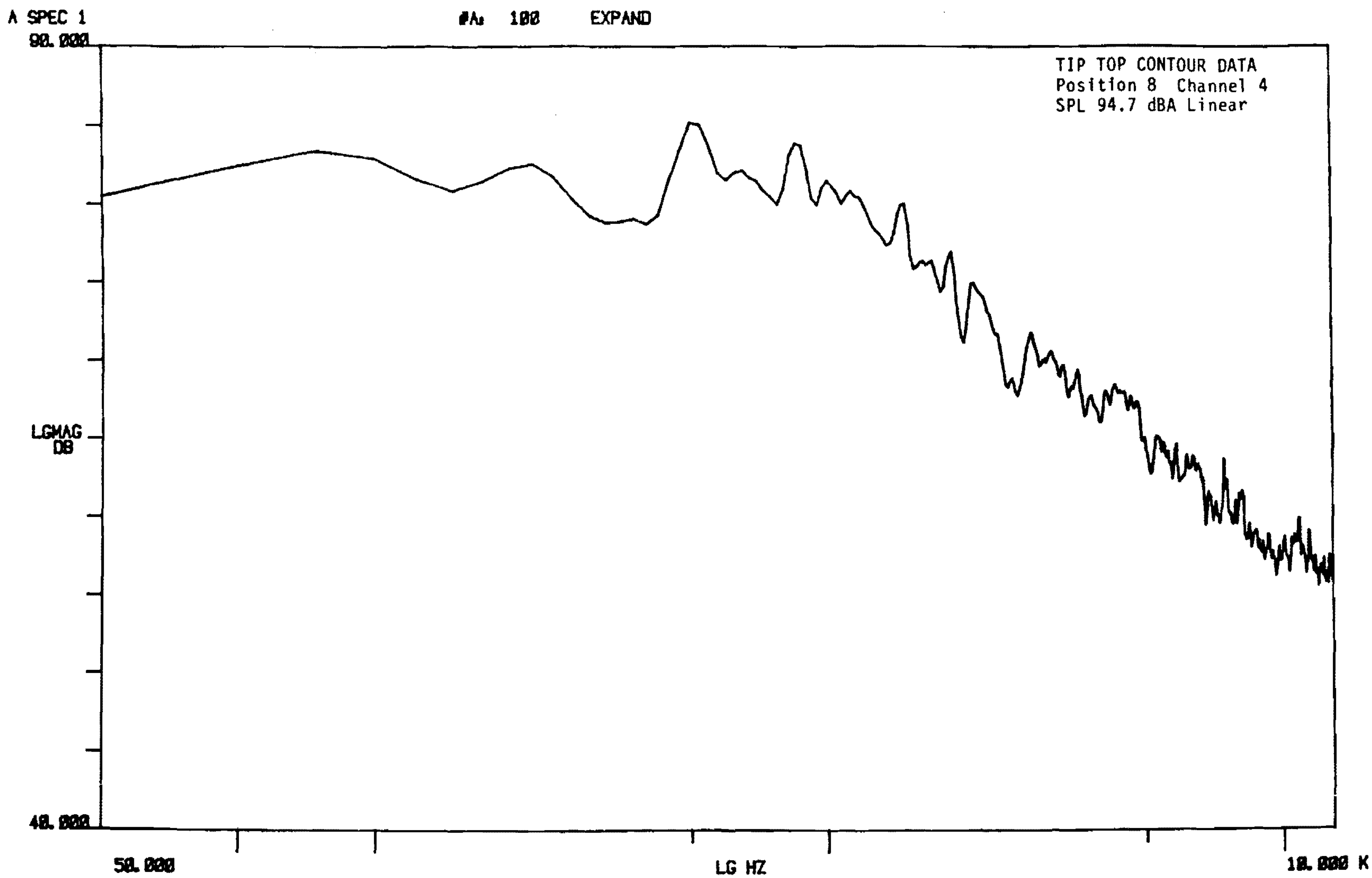

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 8 Channel 4
SPL 94.7 dBL Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 47B



A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 8 Channel 4
SPL 94.7 dBA Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 48B

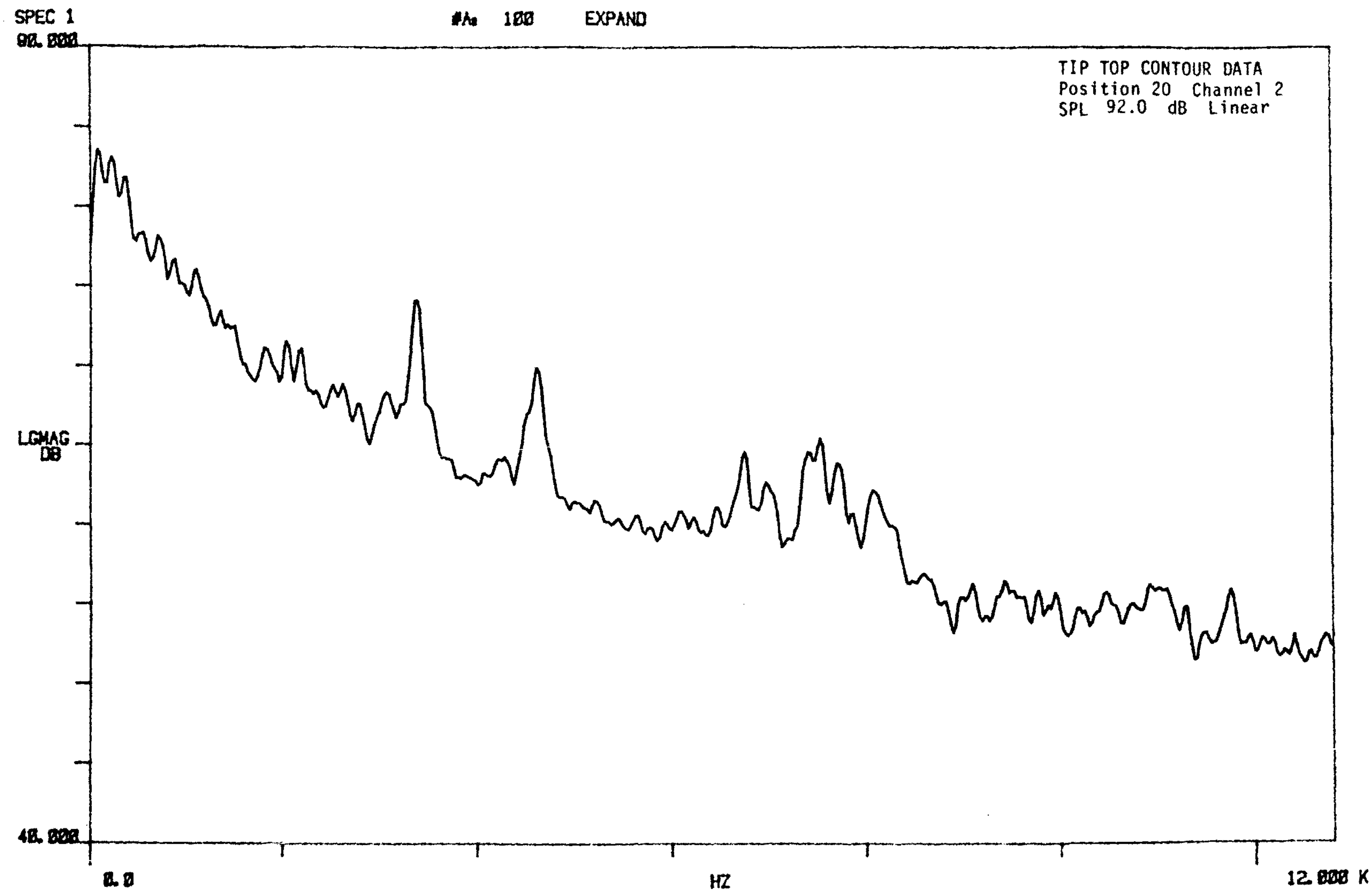

SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 20 Channel 2
SPL 92.0 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 49B

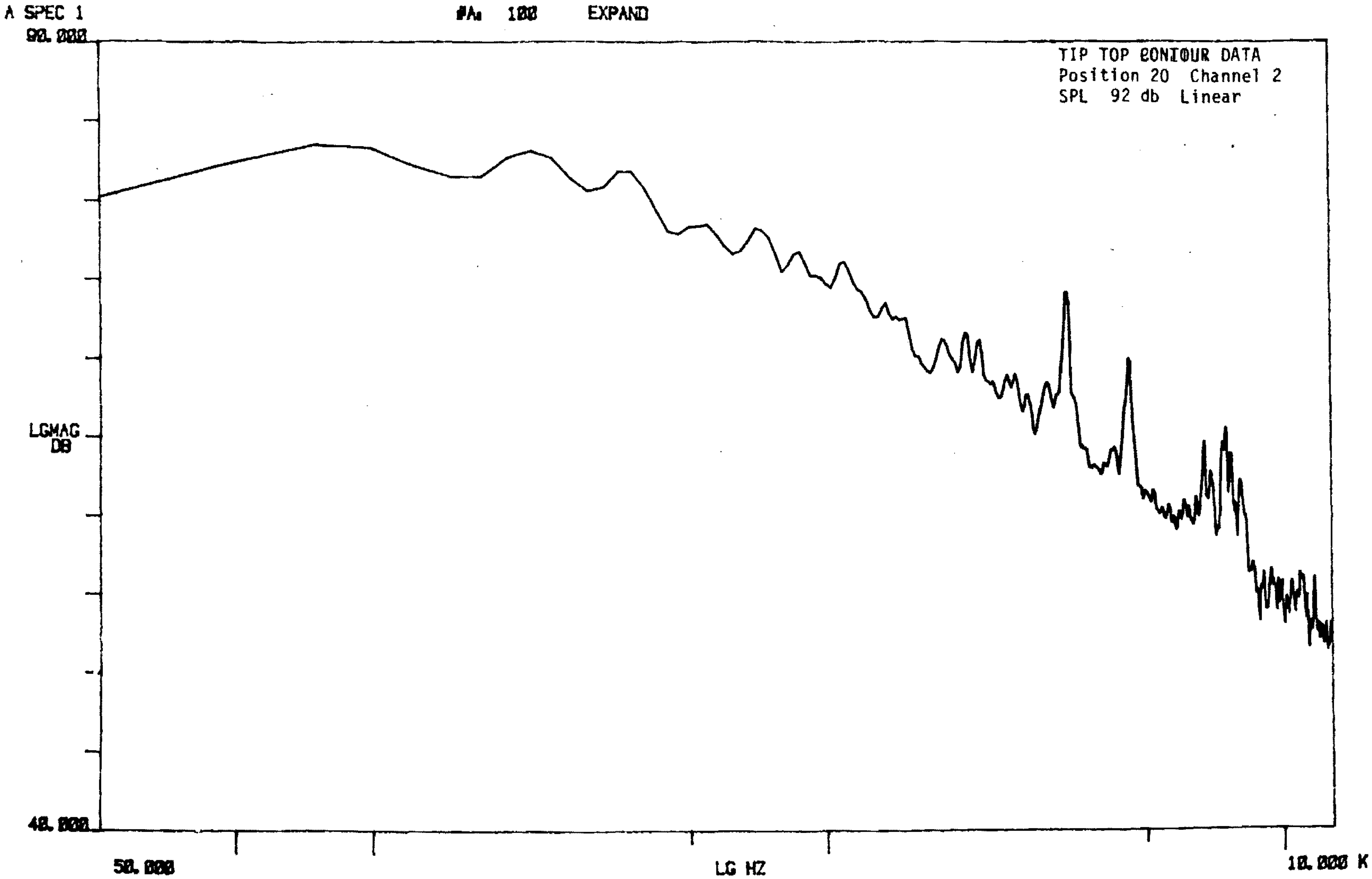

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 20 Channel 2
SPL 92 db Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 50B

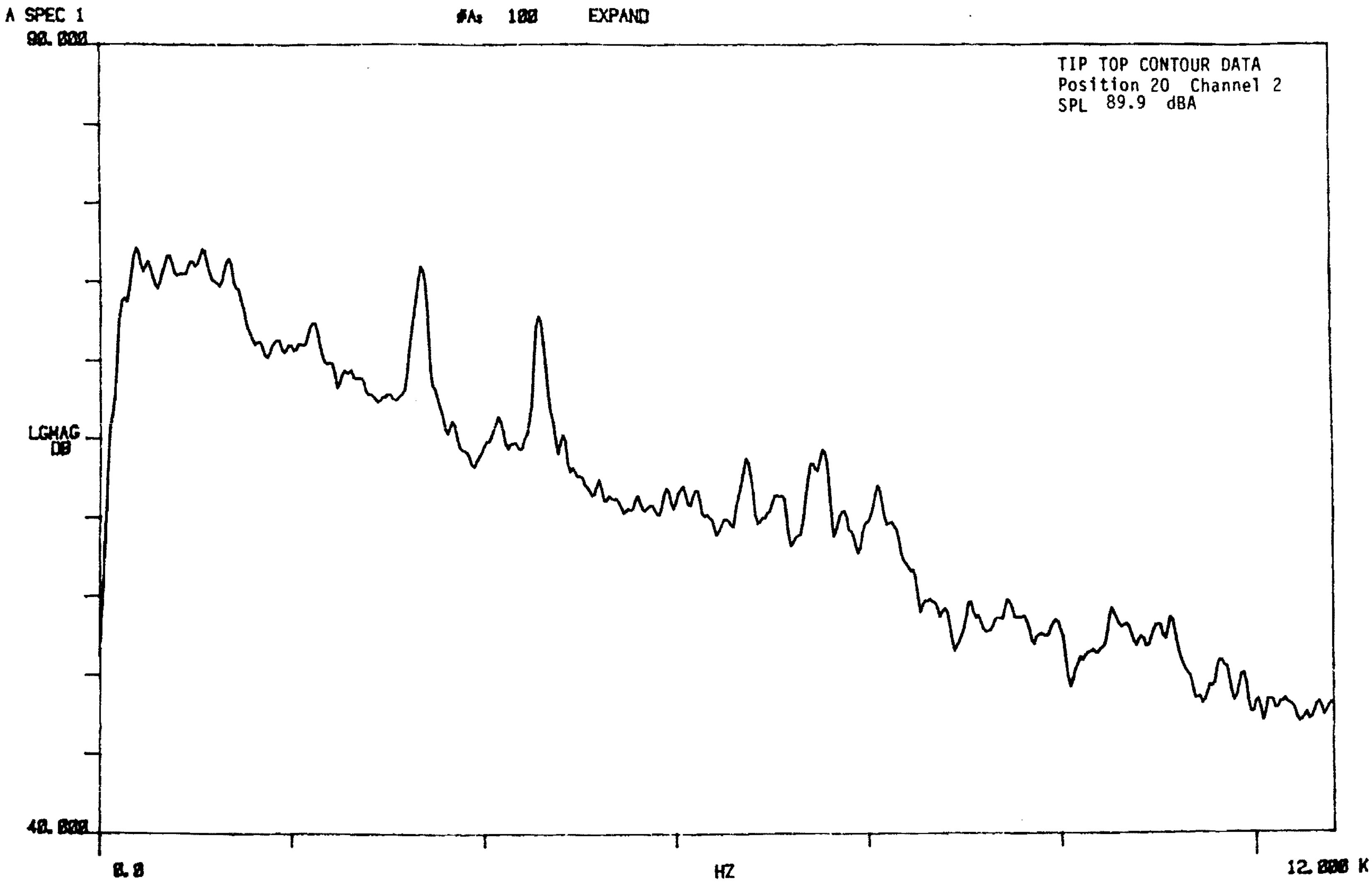

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 20 Channel 2
SPL 89.9 dBA

Figure 51B

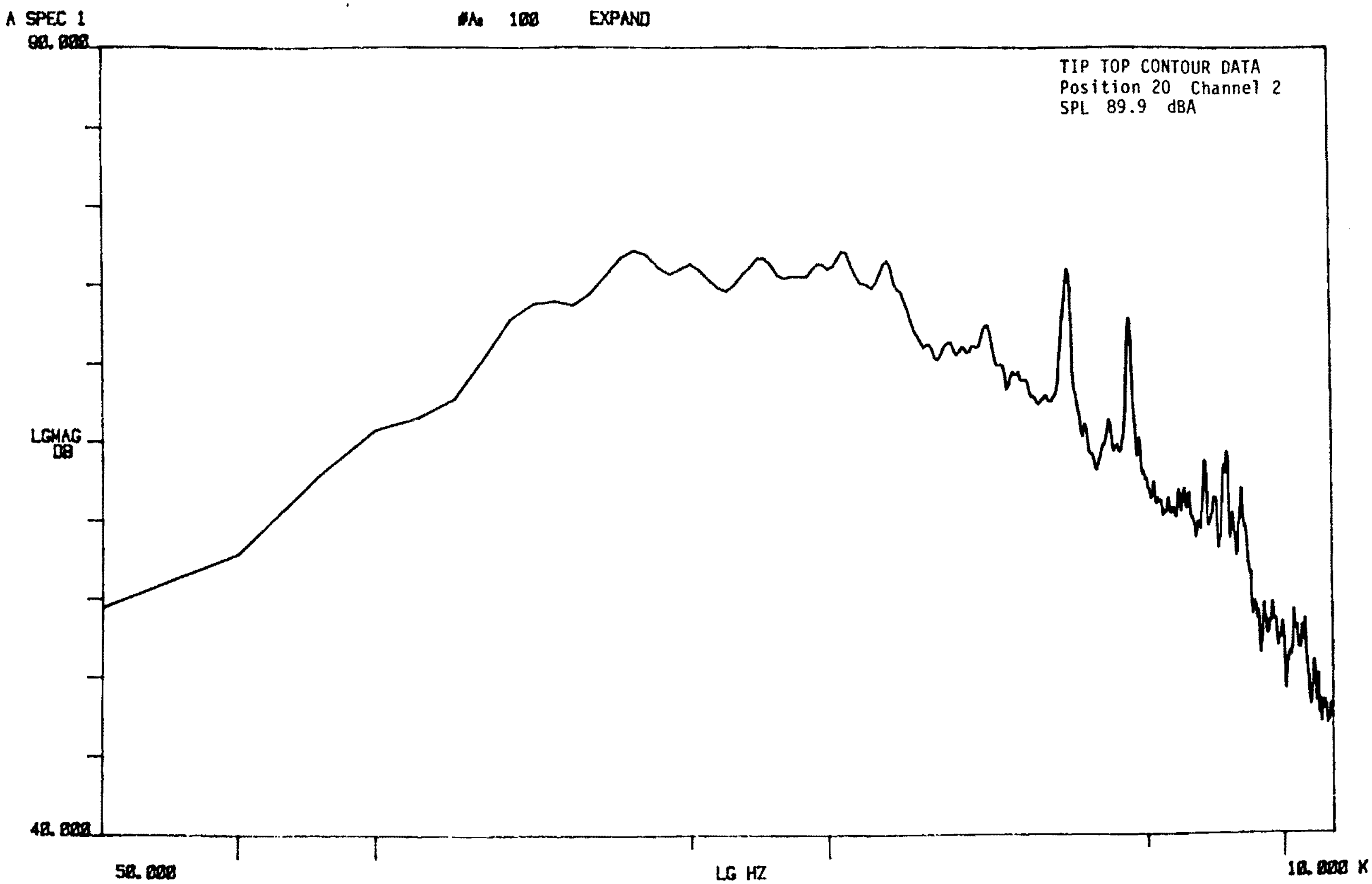

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG DB
40.000
50.000
LG HZ
10.000 K
TIP TOP CONTOUR DATA
Position 20 Channel 2
SPL 89.9 dBA

Figure 52B

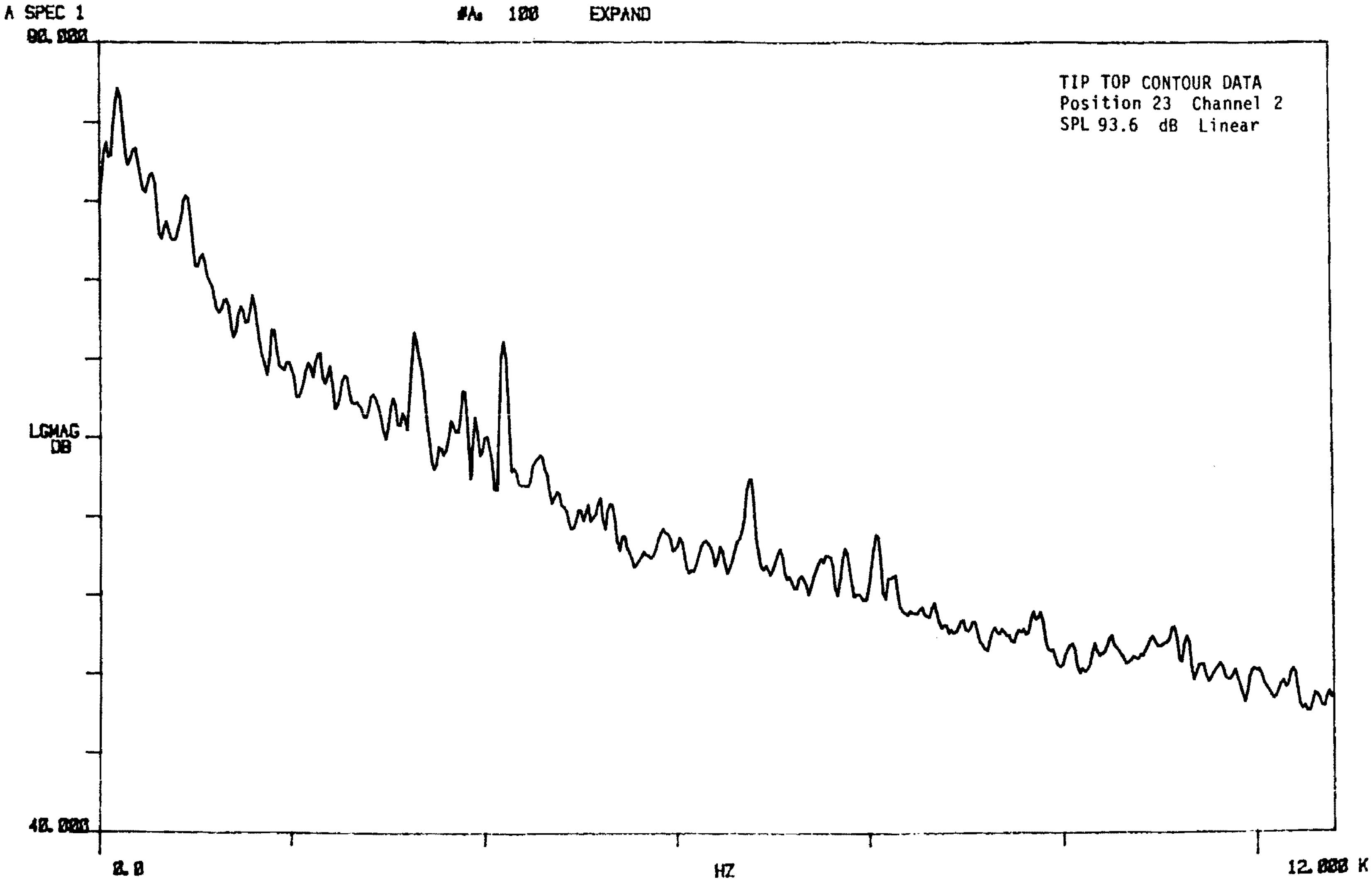

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 23 Channel 2
SPL 93.6 dB Linear

Figure 53B

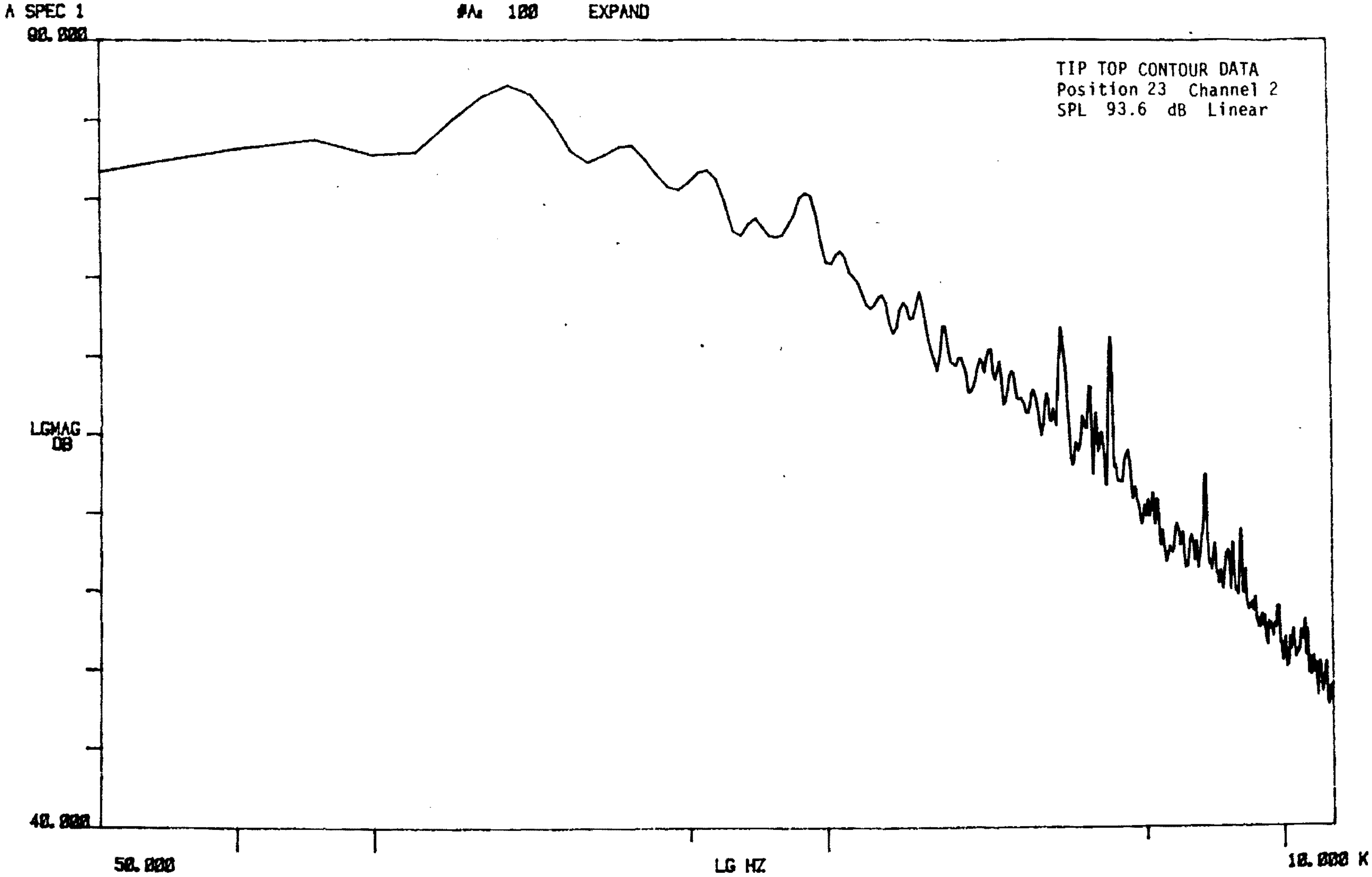

A SPEC 1
#A: 100
EXPAND
80.000
TIP TOP CONTOUR DATA
Position 23 Channel 2
SPL 93.6 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 54B

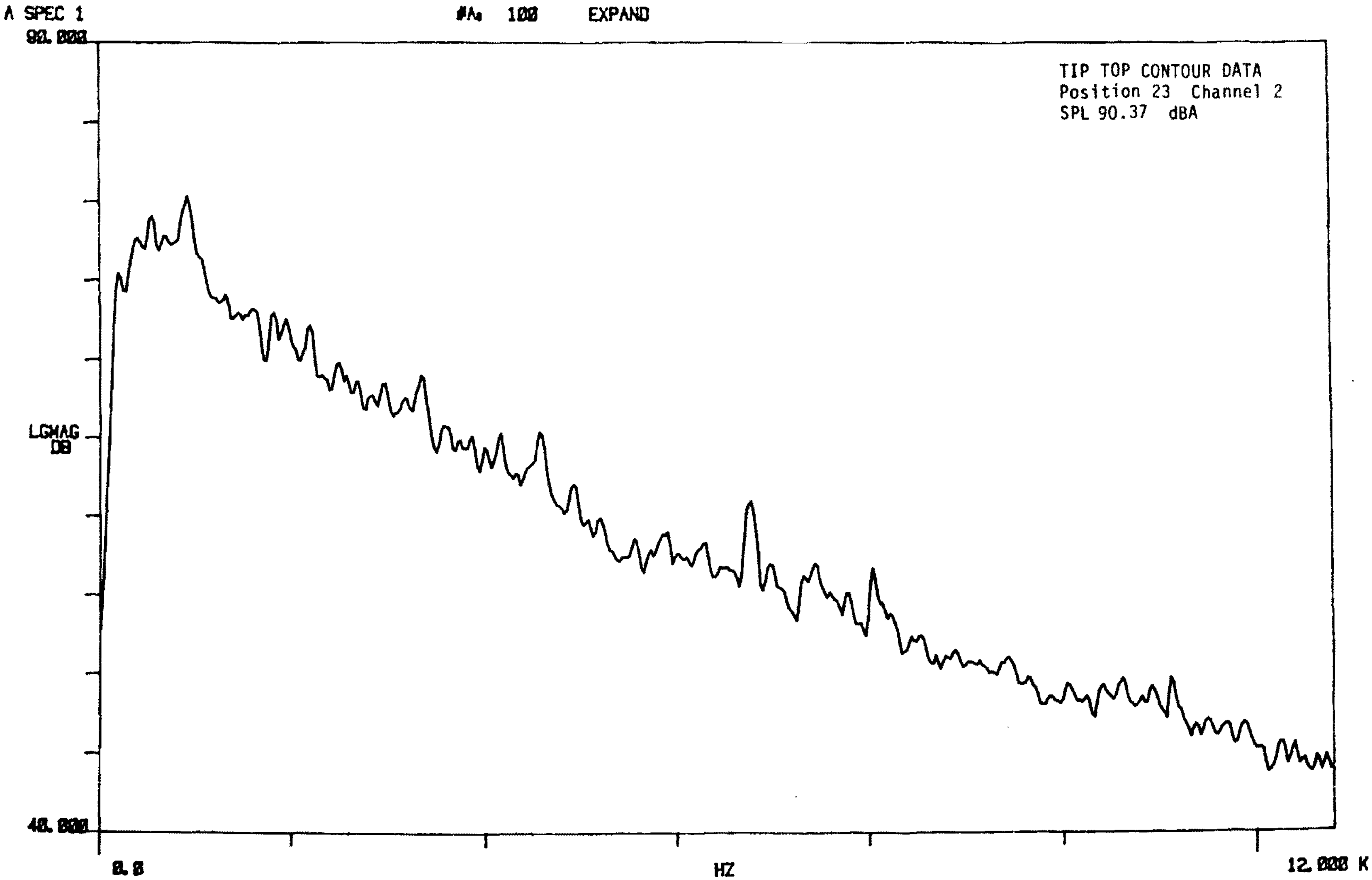

A SPEC 1
#A: 100 EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 23 Channel 2
SPL 90.37 dBA

Figure 55B

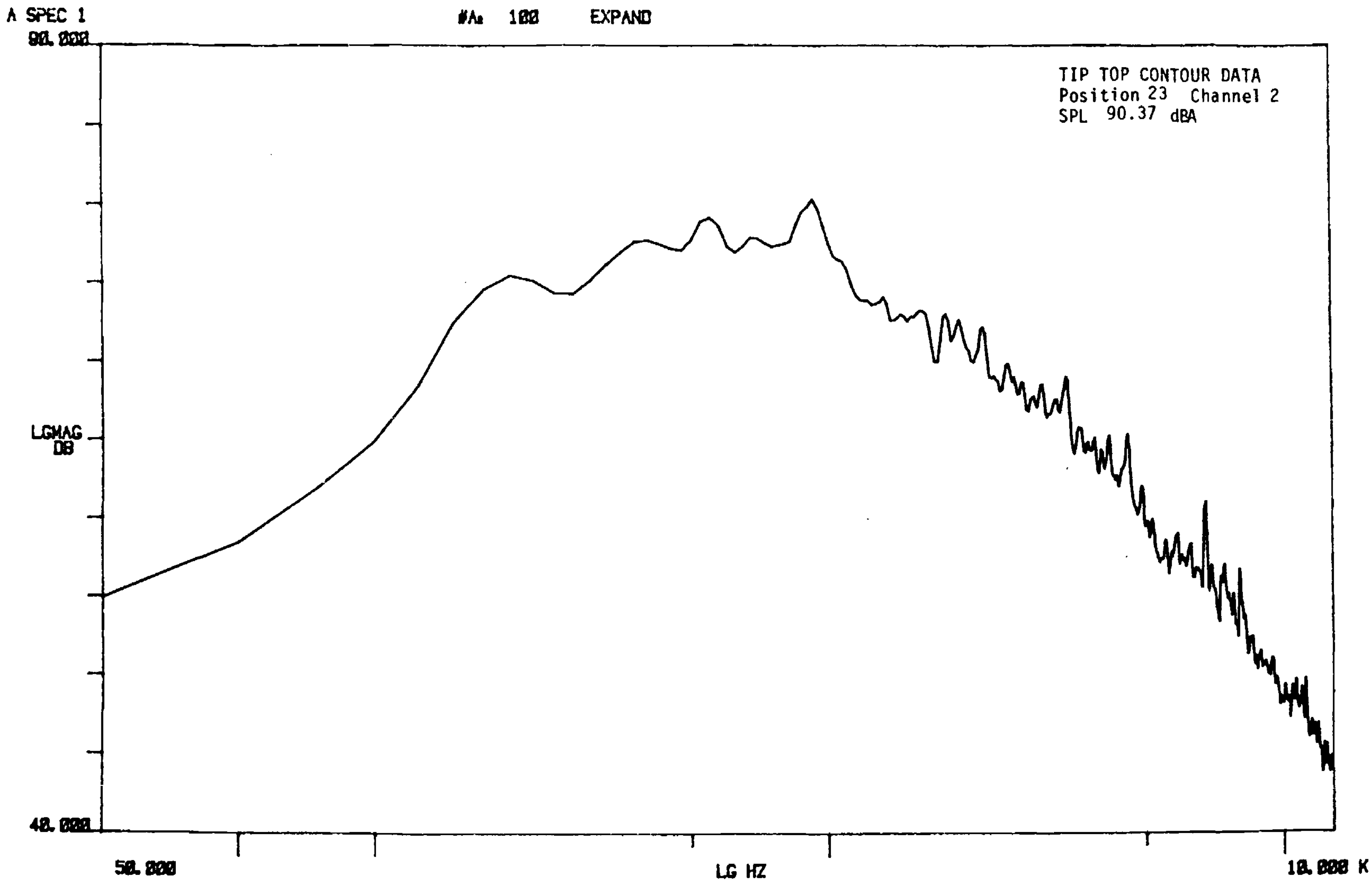
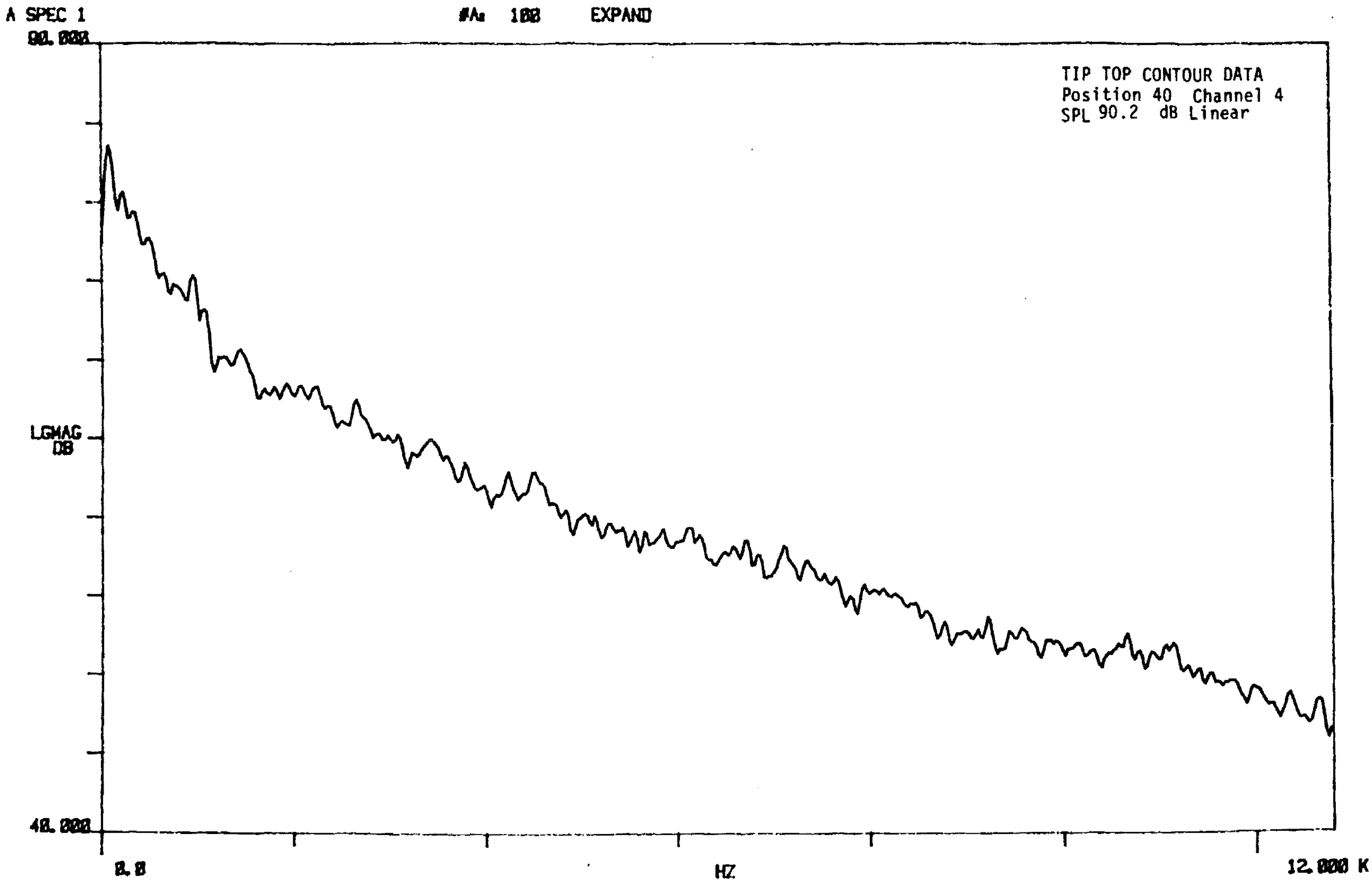

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 23 Channel 2
SPL 90.37 dBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 56B



A SPEC 1
#A: 100
EXPAND
90.000
LGMAG DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 40 Channel 4
SPL 90.2 dB Linear

Figure 57B

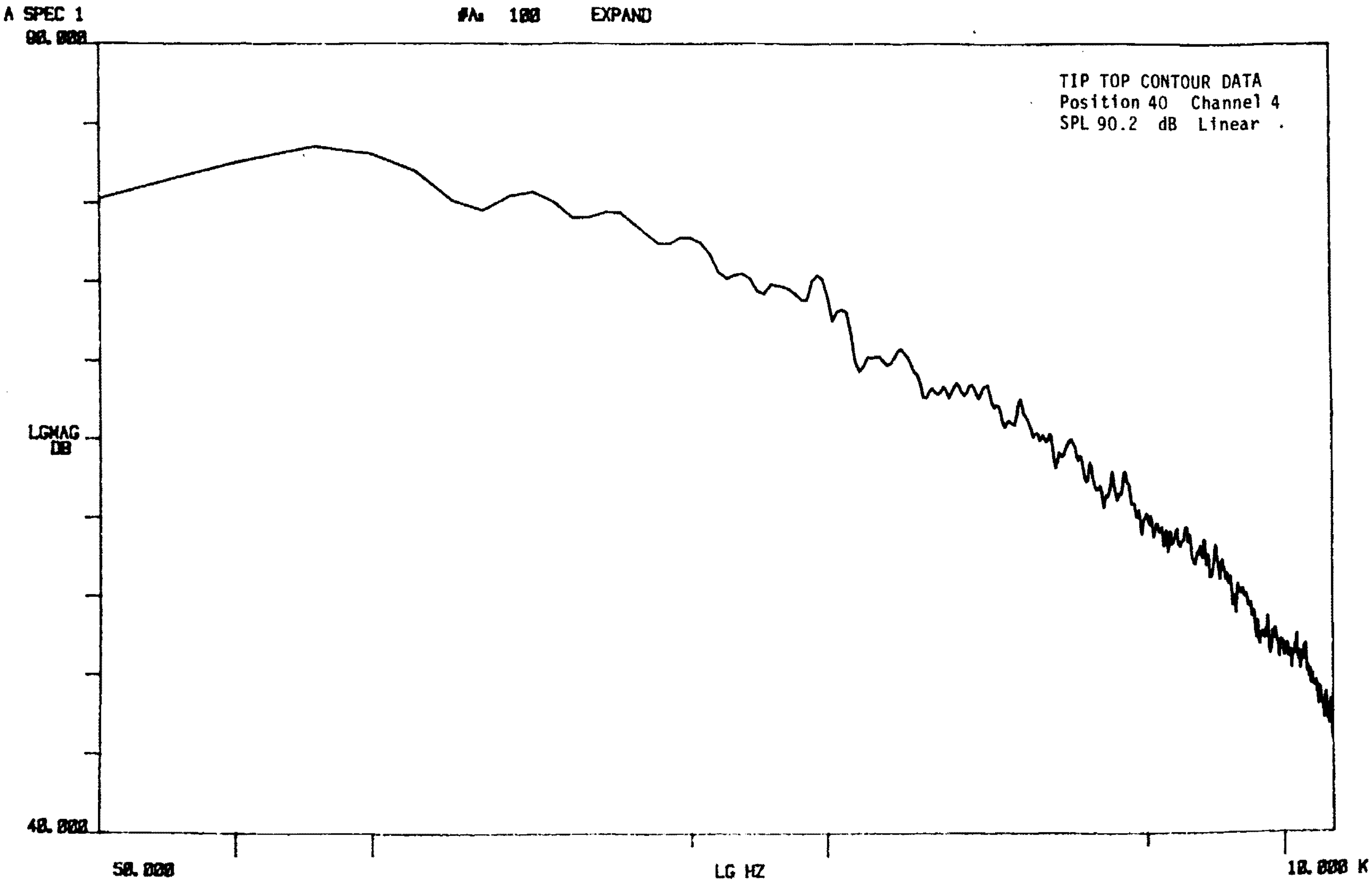

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 40 Channel 4
SPL 90.2 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 58B

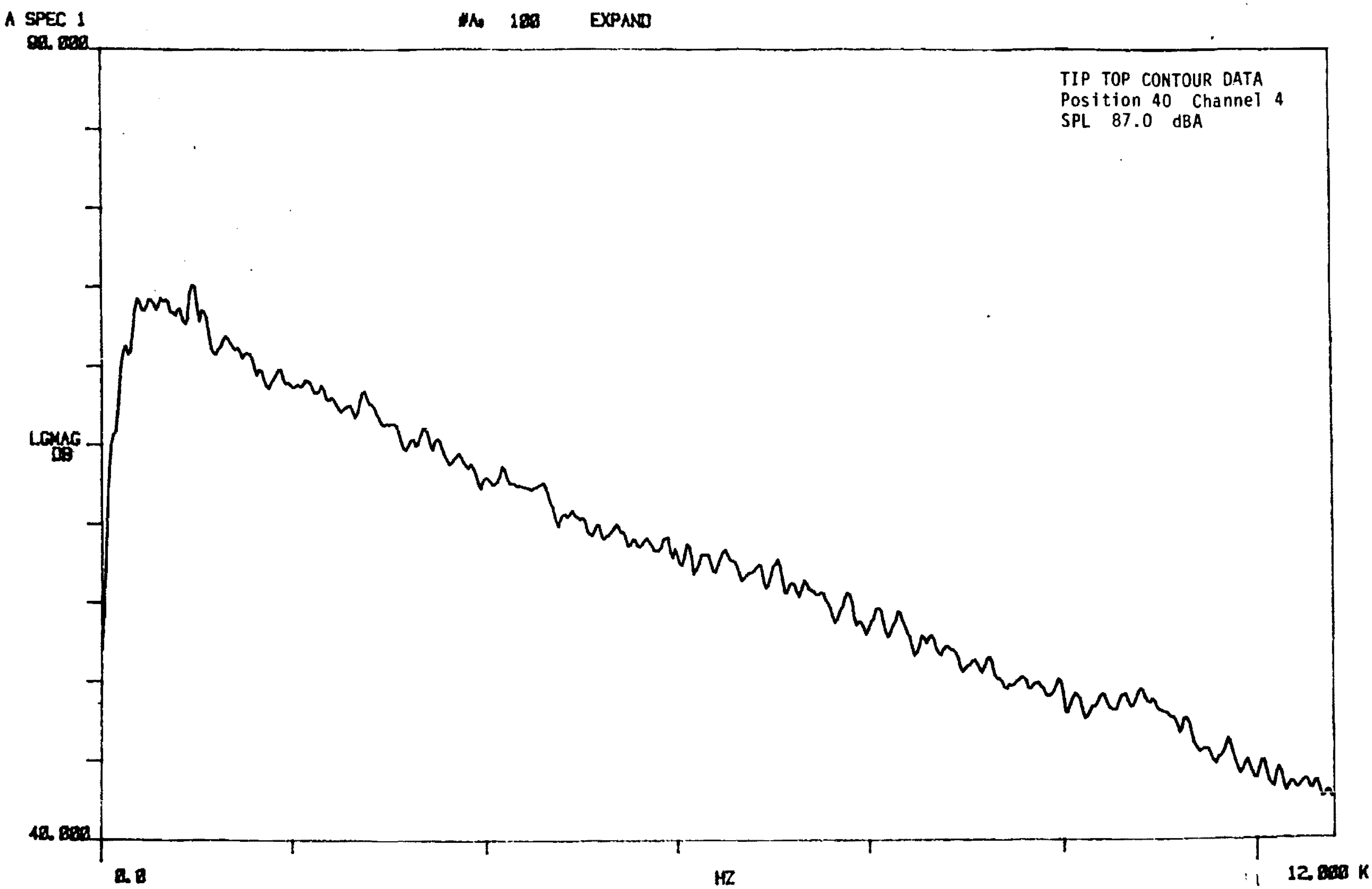
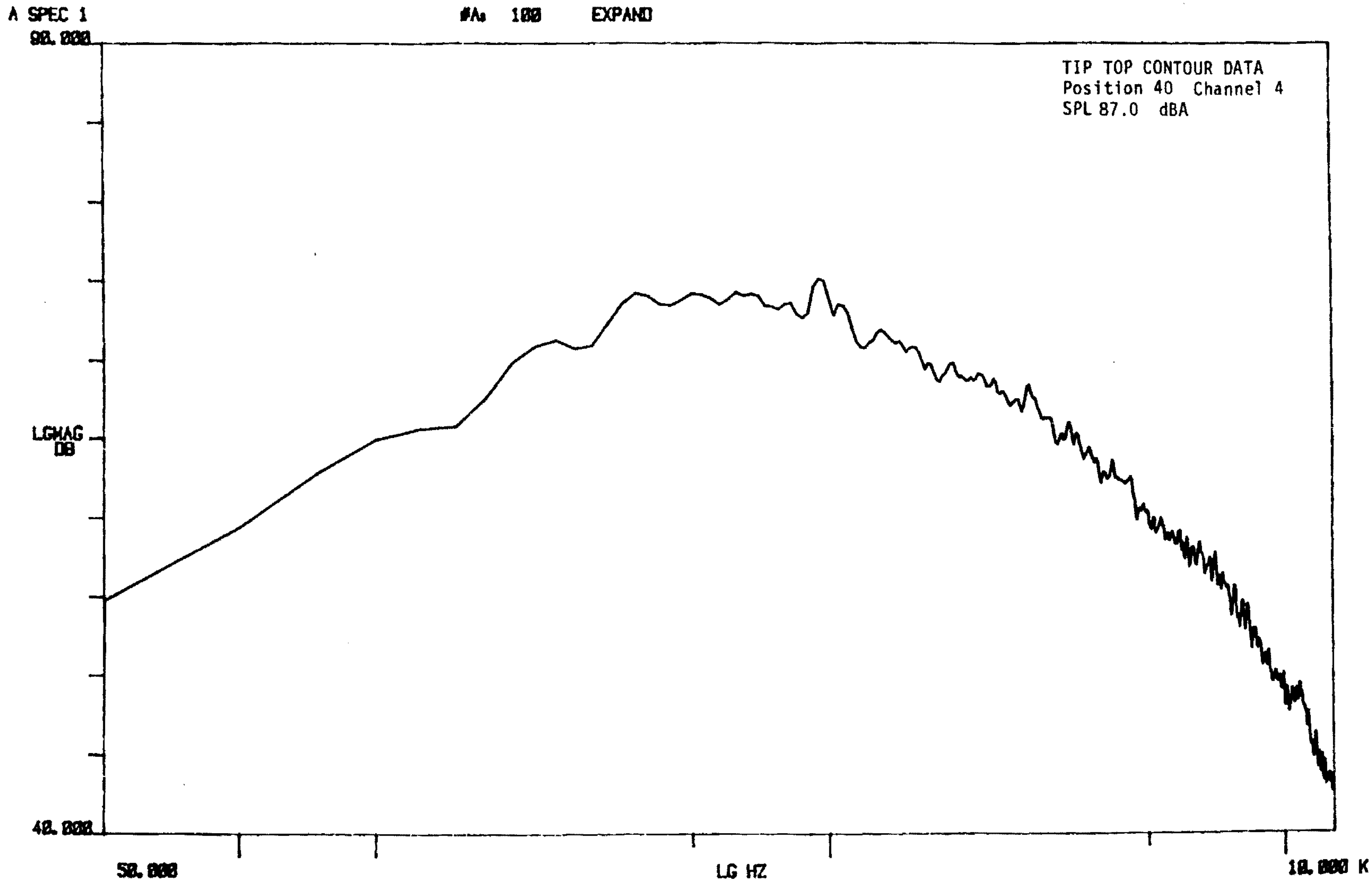

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 40 Channel 4
SPL 87.0 dBA

Figure 59B



A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 40 Channel 4
SPL 87.0 dBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 60B

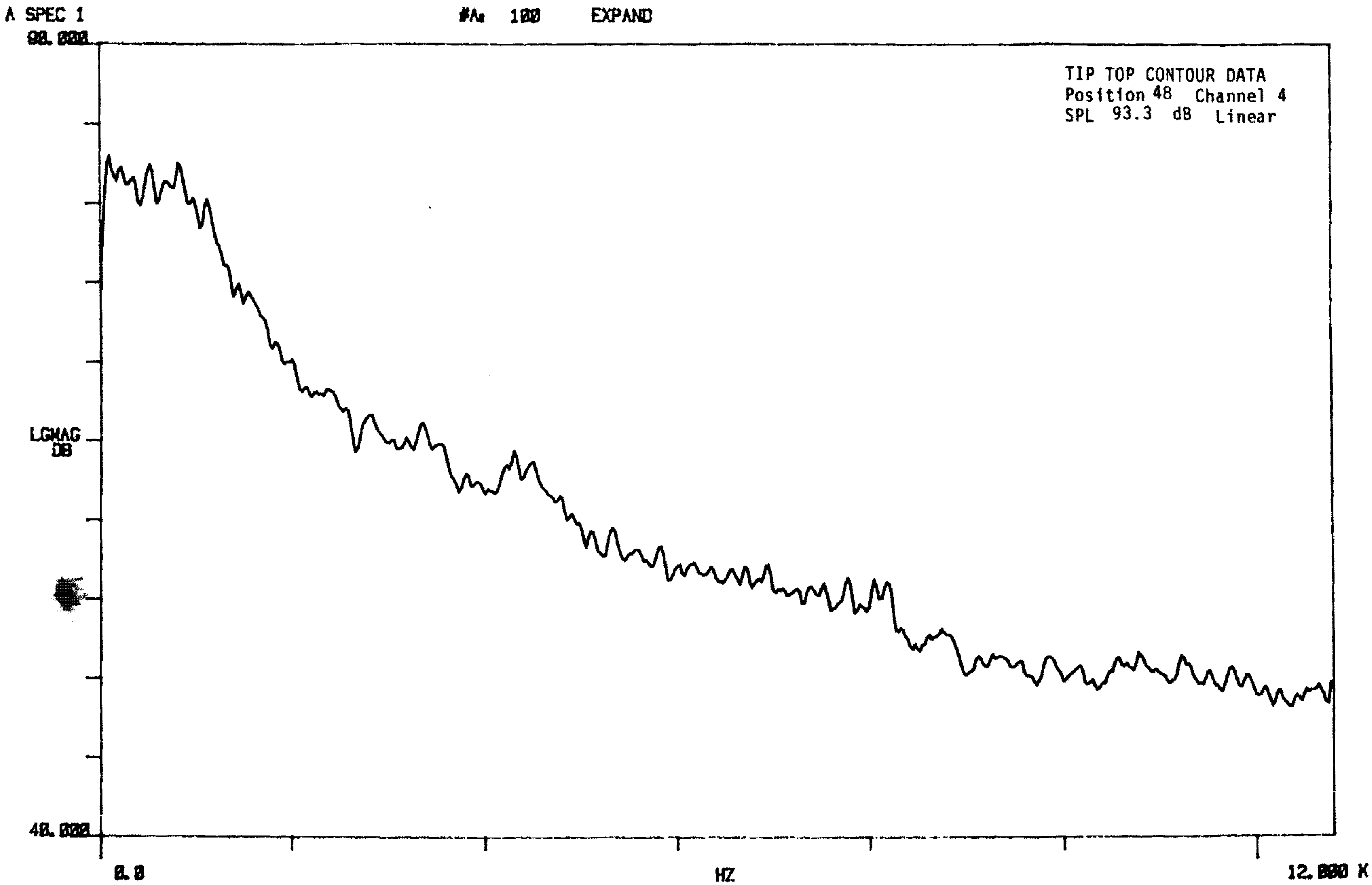

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 48 Channel 4
SPL 93.3 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 61B

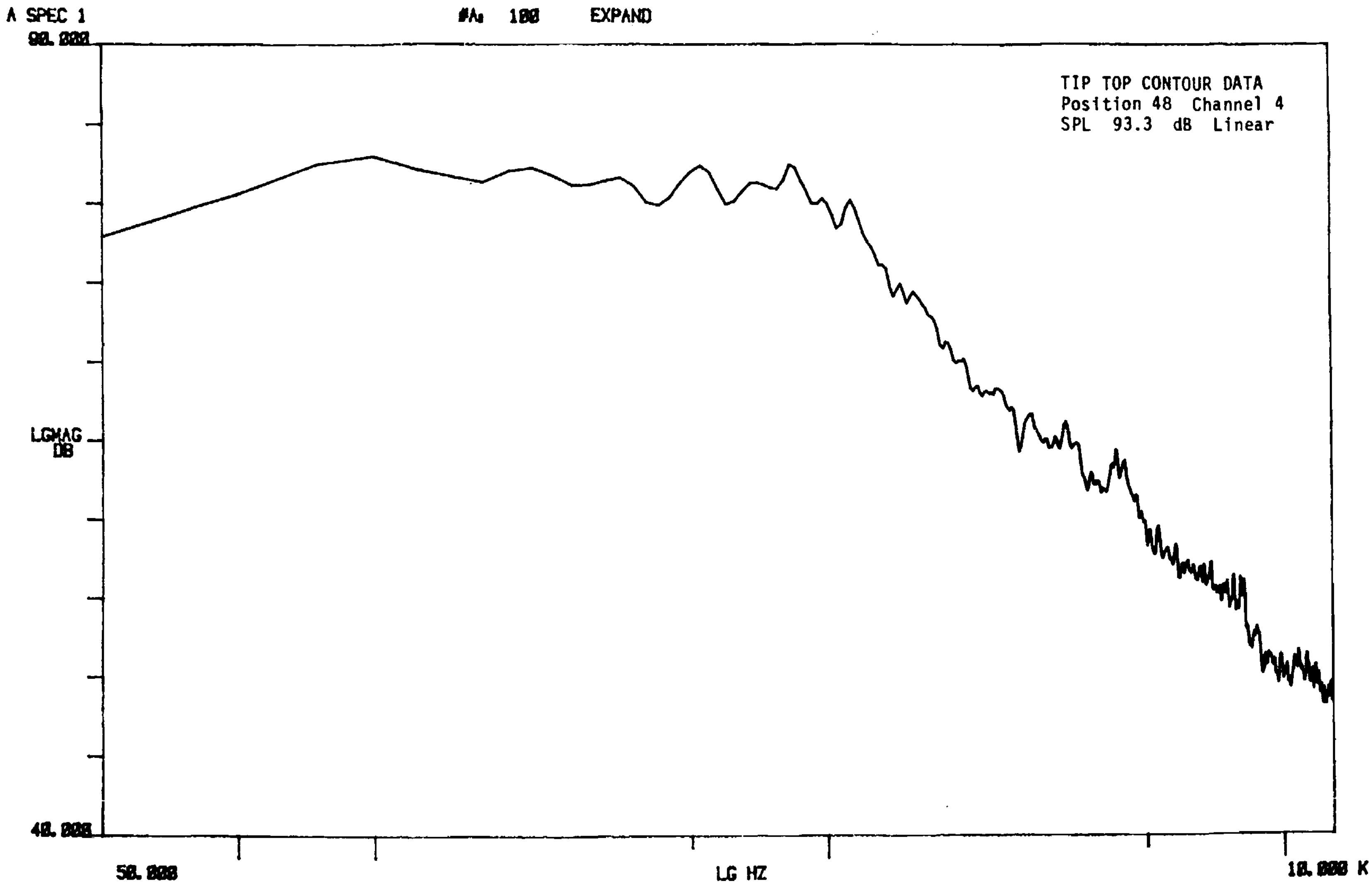

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 48 Channel 4
SPL 93.3 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 62B

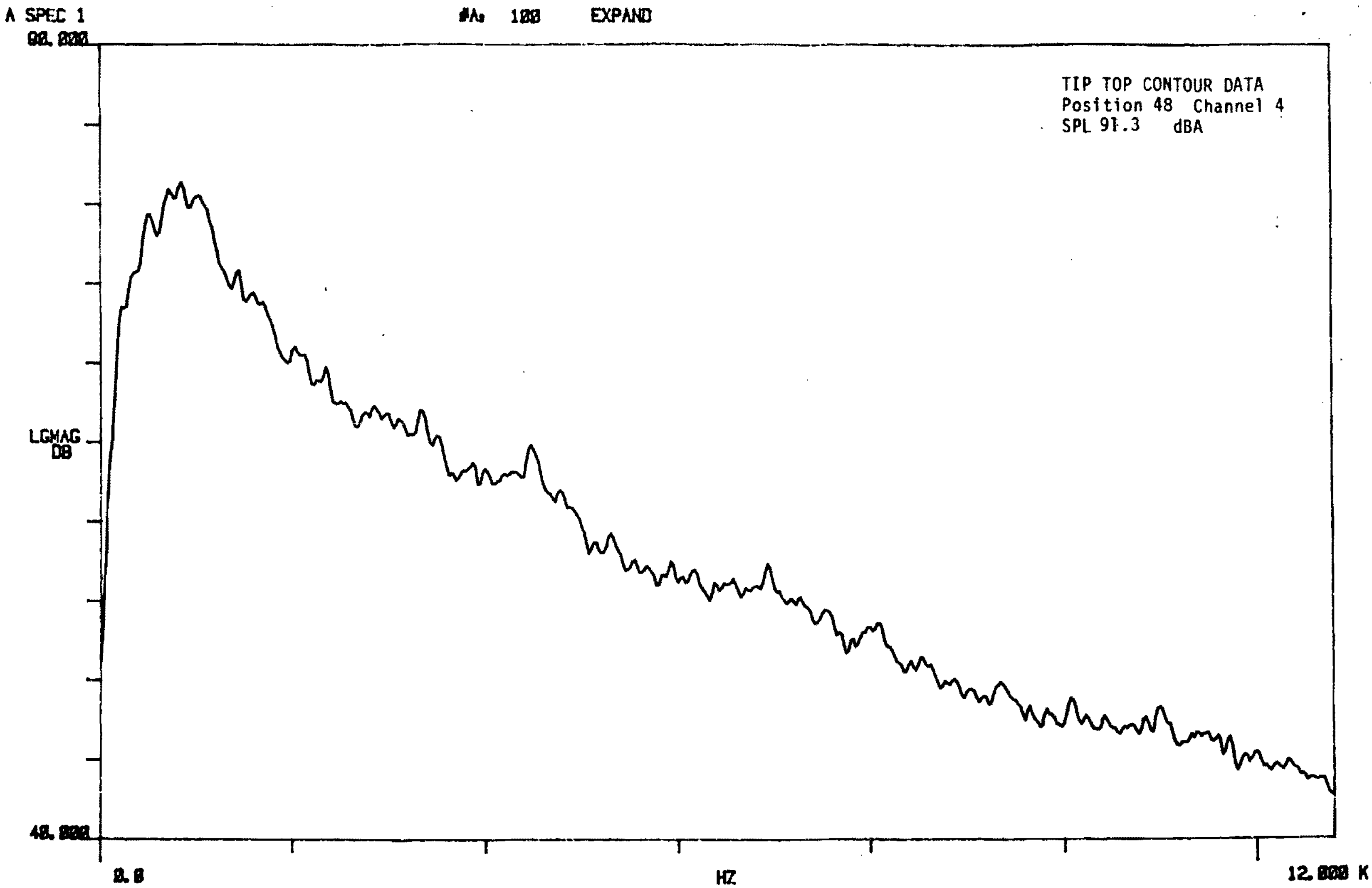

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 48 Channel 4
SPL 91.3 dBA
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 63B

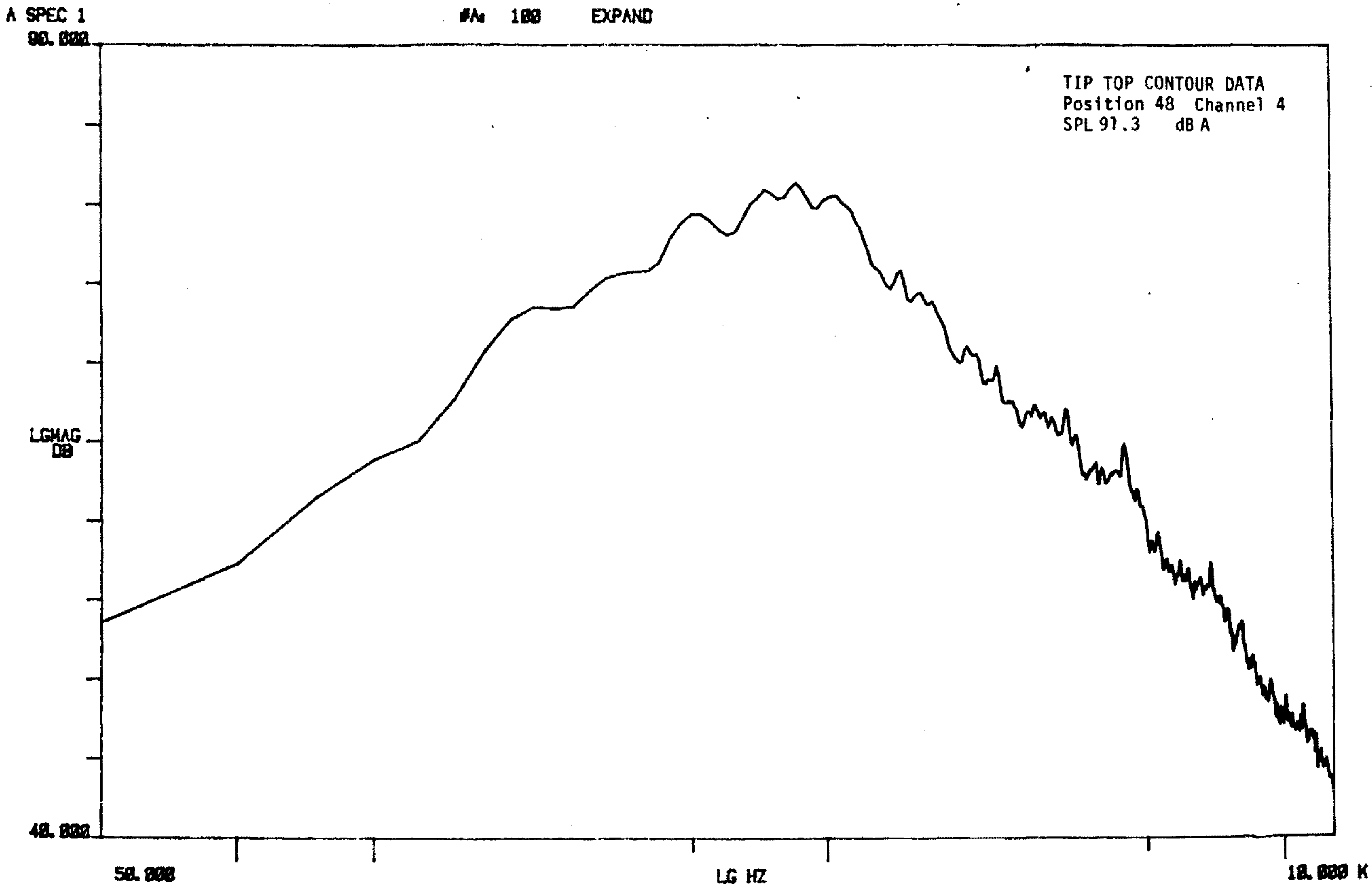
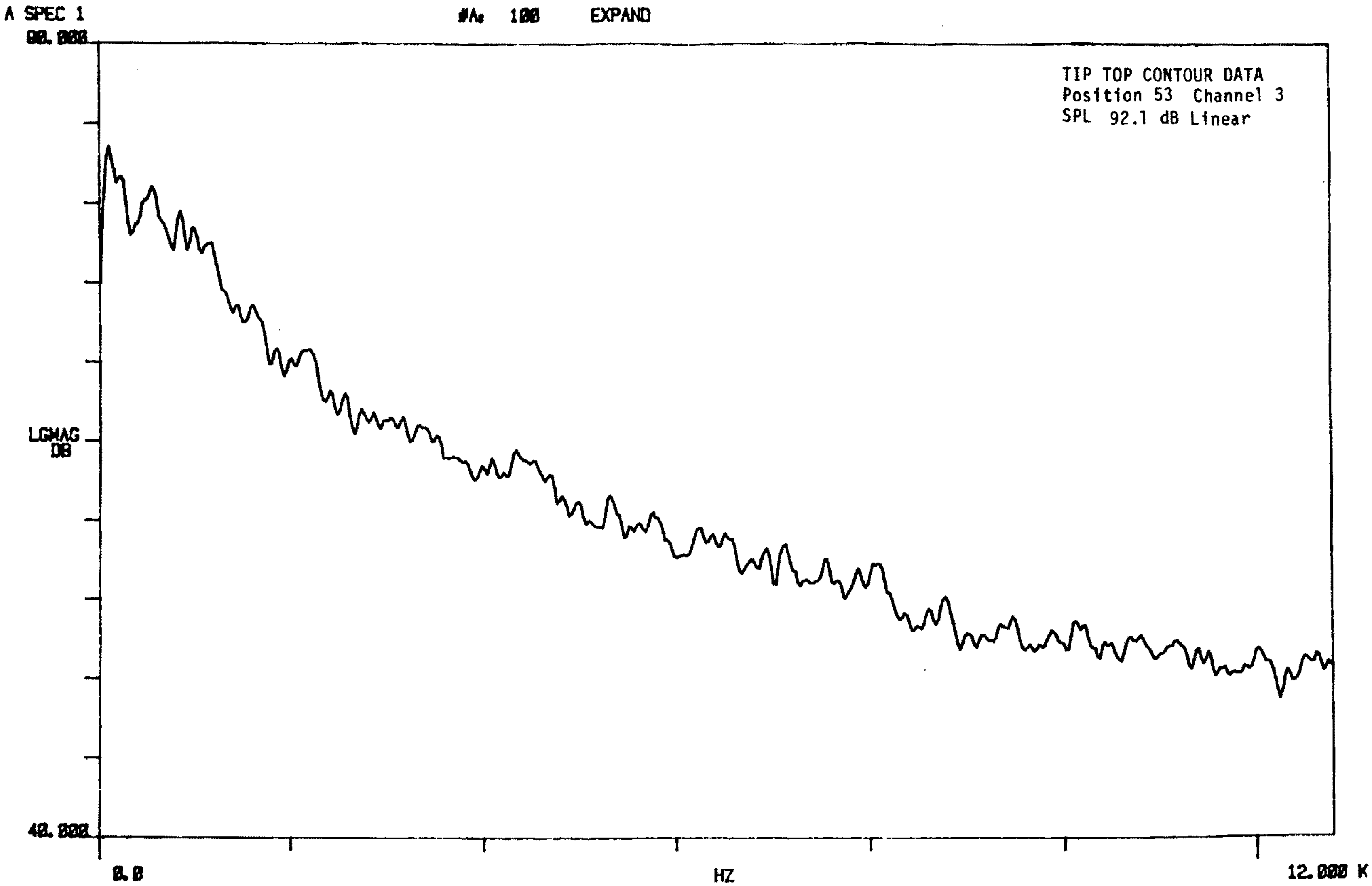

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 48 Channel 4
SPL 91.3 dB A
LGMAG DB
40.000
50.000
LG HZ
10.000 K

Figure 64B



A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 53 Channel 3
SPL 92.1 dB Linear
LGMAG DB
40.000
0.0
HZ
12.000 K

Figure 65B

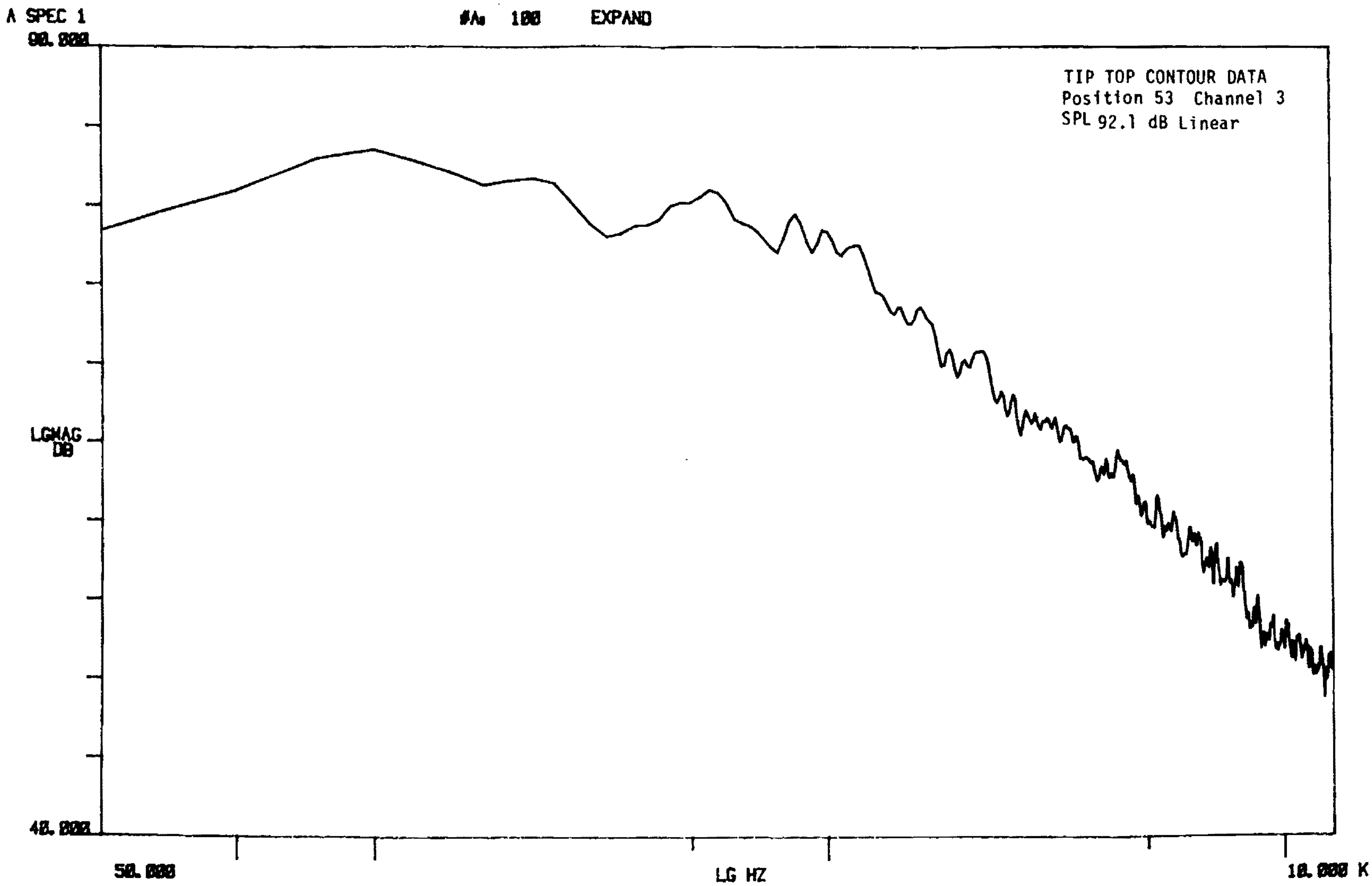

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 53 Channel 3
SPL 92.1 dB Linear
LGMAG DB
40.000
50.000
LG HZ
10.000 K

Figure 66B

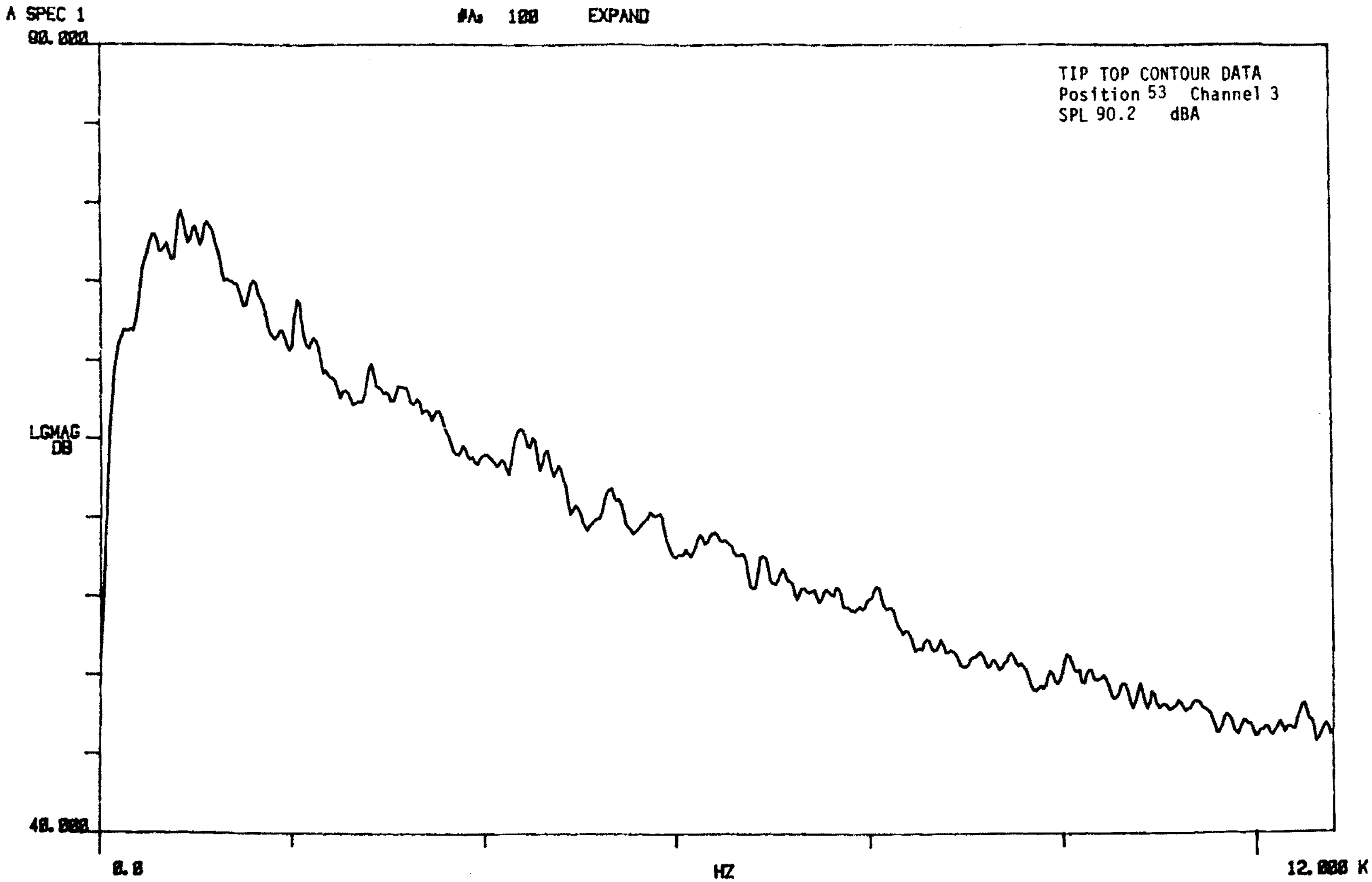

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 53 Channel 3
SPL 90.2 dBA
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 67B

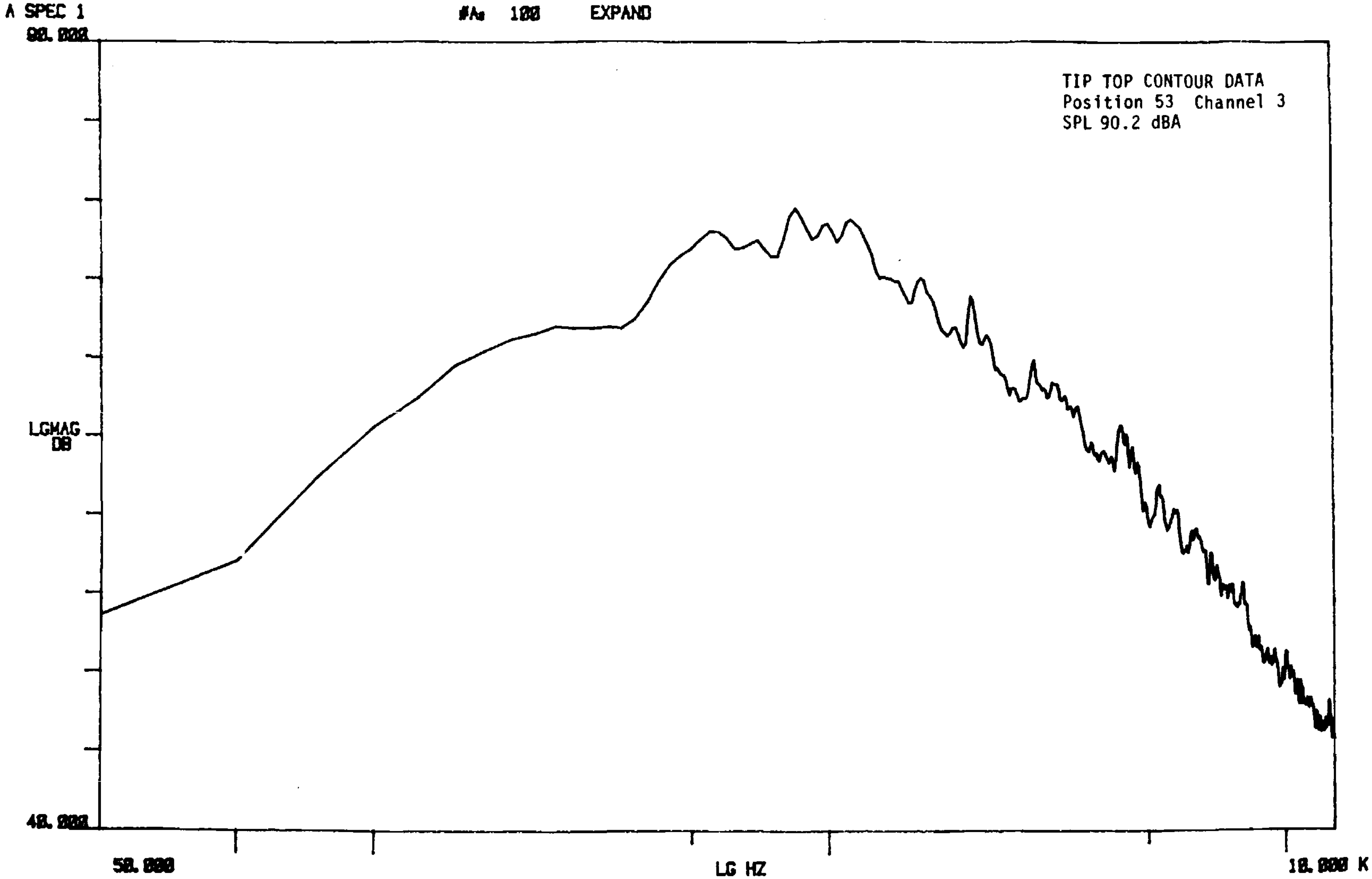
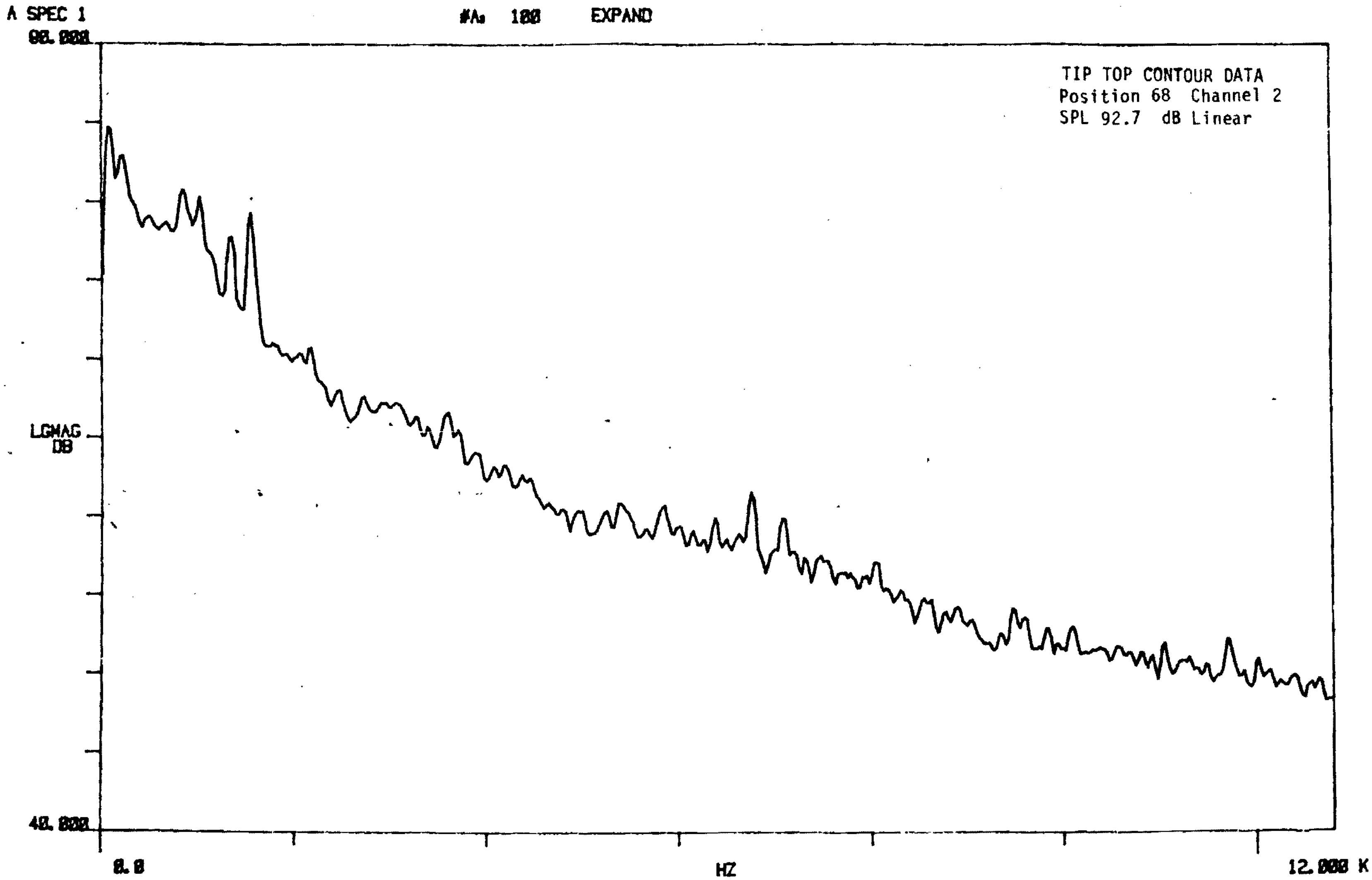
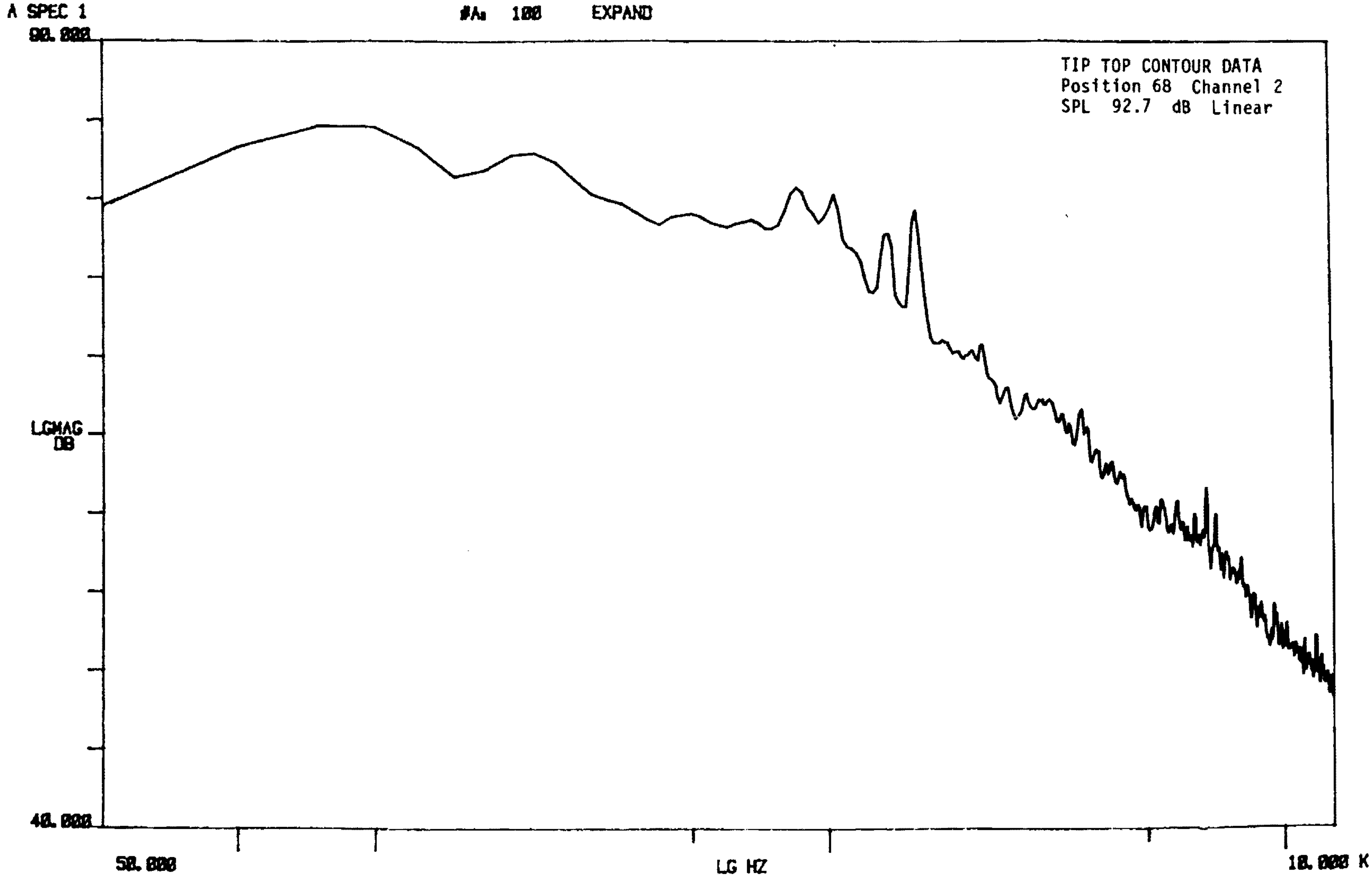

A SPEC 1
#A: 100 EXPAND
80.000
TIP TOP CONTOUR DATA
Position 53 Channel 3
SPL 90.2 dBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 68B



A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 68 Channel 2
SPL 92.7 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 69B



A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 68 Channel 2
SPL 92.7 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 70B

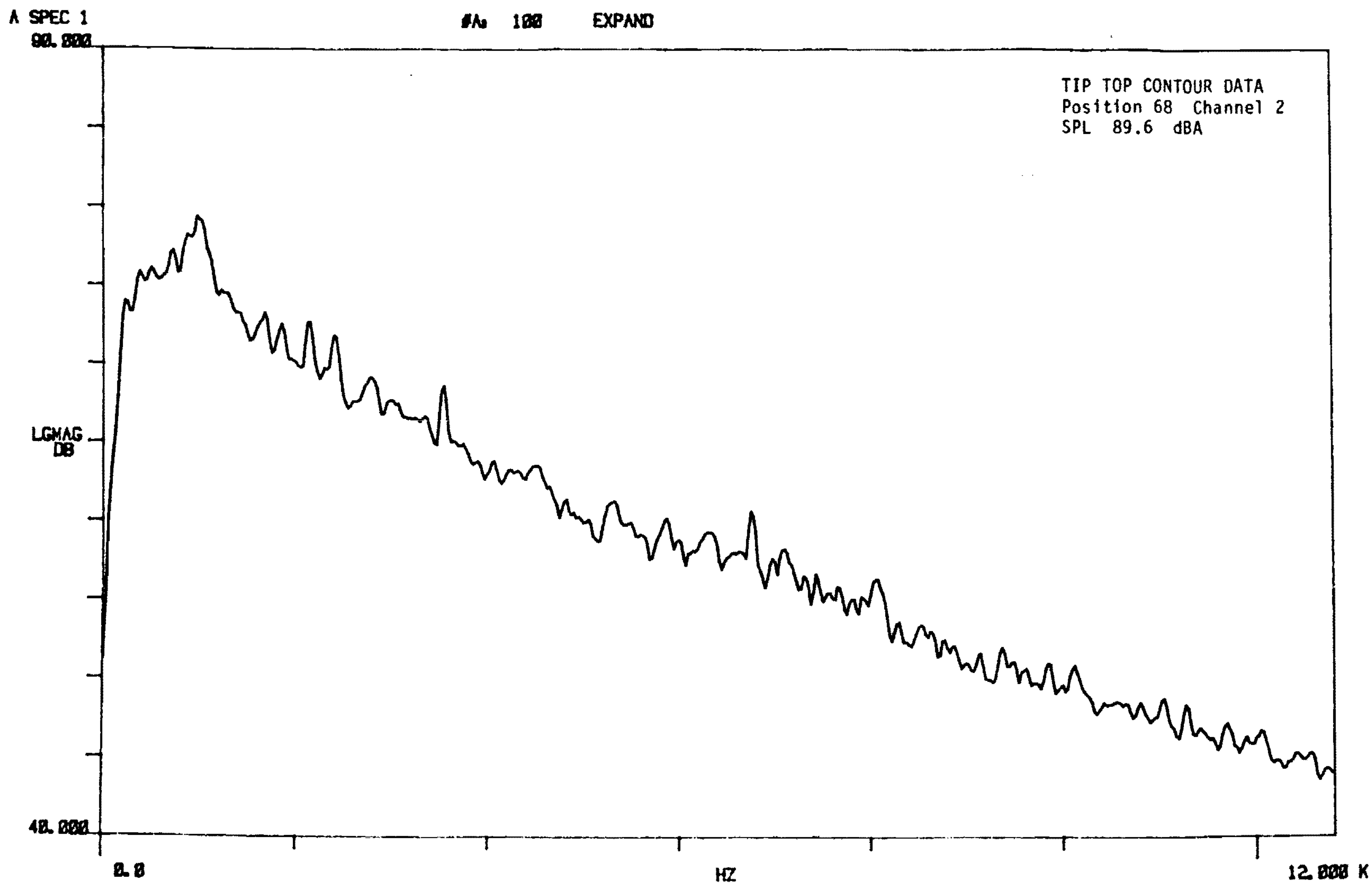

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 68 Channel 2
SPL 89.6 dBA
LGMAG DB
40.000
0.0
HZ
12.000 K

Figure 71B

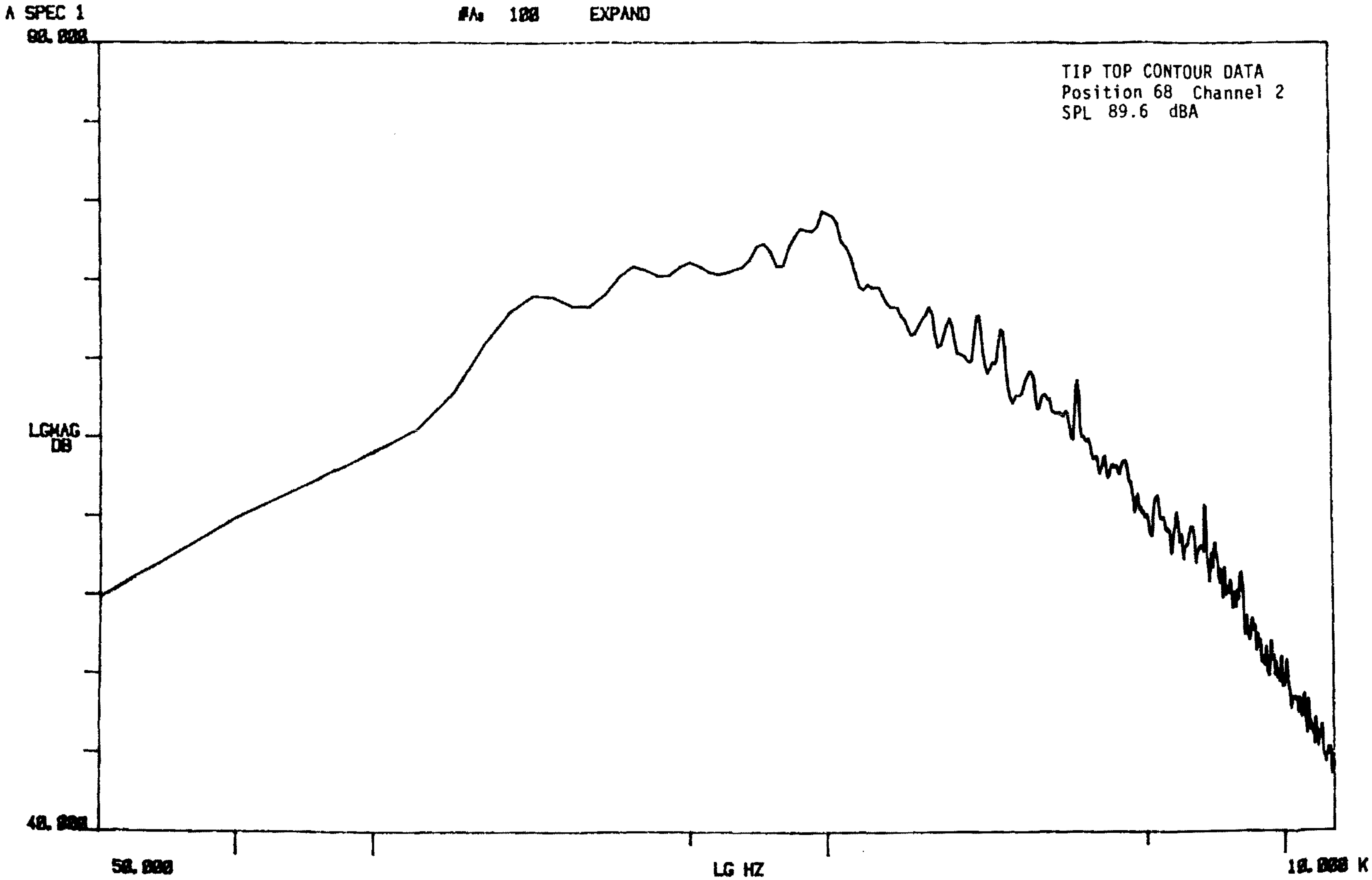

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG DB
40.000
50.000
LG HZ
10.000 K
TIP TOP CONTOUR DATA
Position 68 Channel 2
SPL 89.6 dBA

Figure 72B

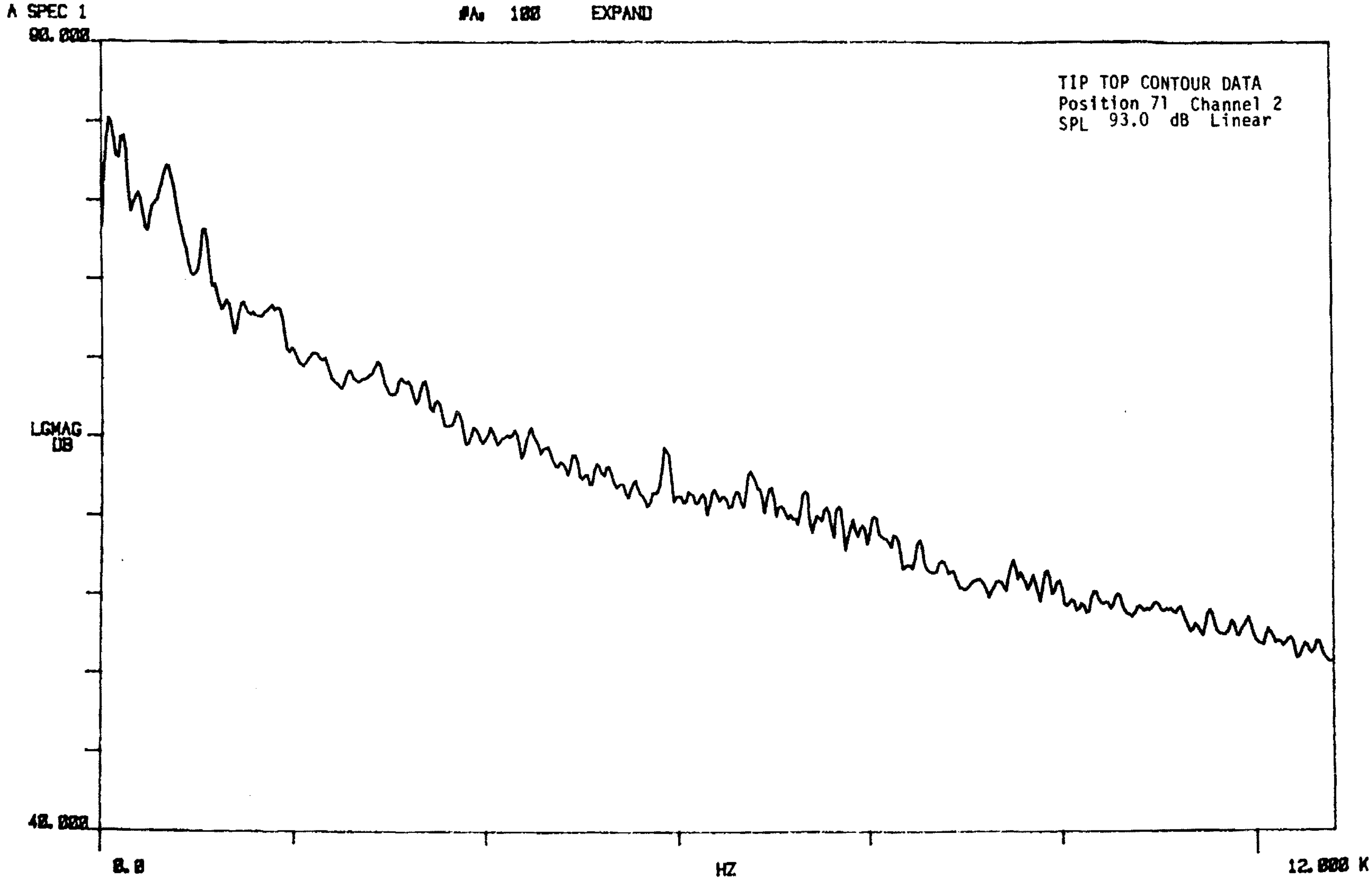

A SPEC 1
#A: 100
EXPAND
90.000
LGMAG
DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 71 Channel 2
SPL 93.0 dB Linear

Figure 73B

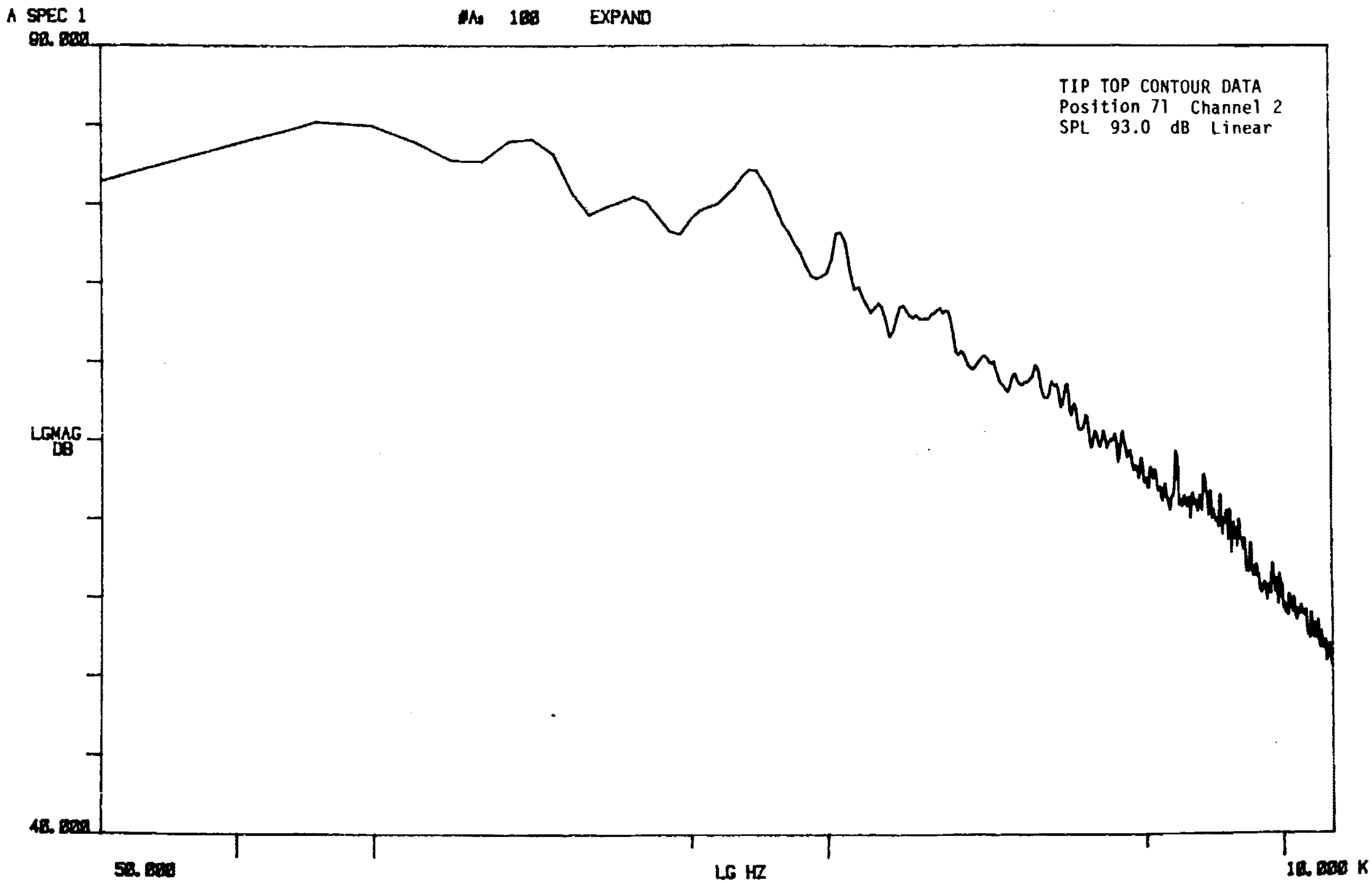

A SPEC 1
#A: 100
EXPAND
80.000
TIP TOP CONTOUR DATA
Position 71 Channel 2
SPL 93.0 dB Linear
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 74B

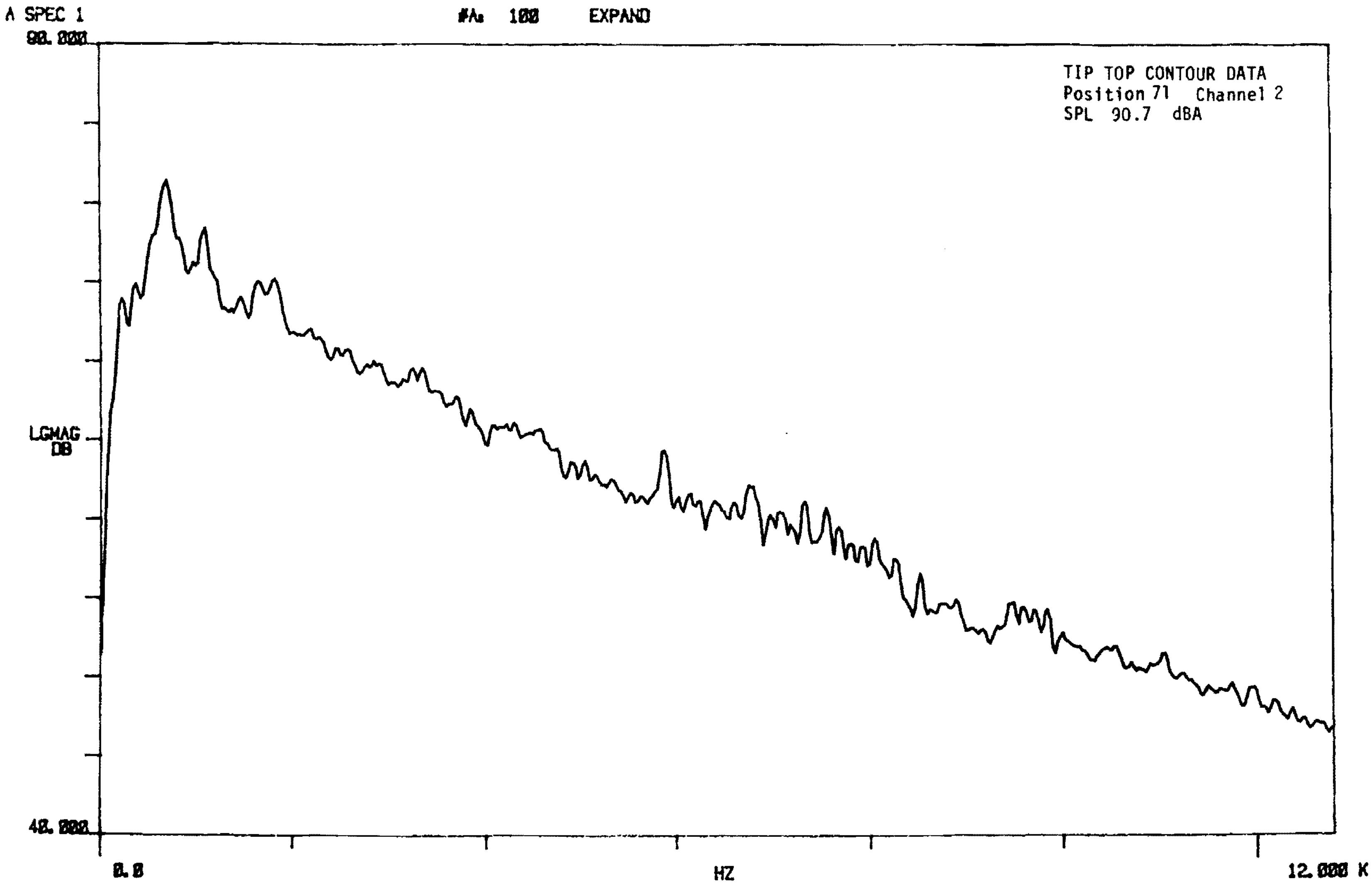

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 71 Channel 2
SPL 90.7 dBA
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 75B

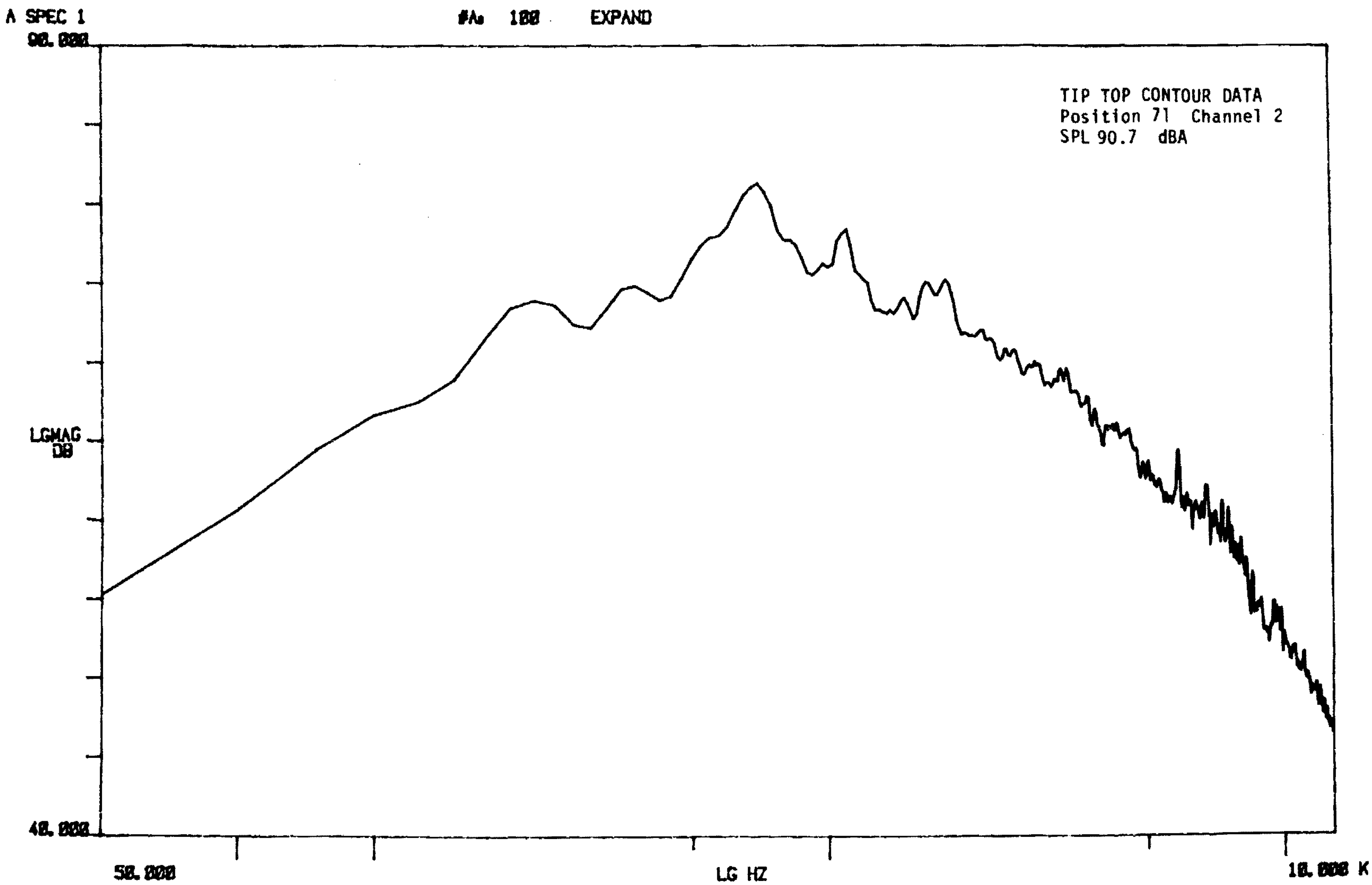

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 71 Channel 2
SPL 90.7 dBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 76B

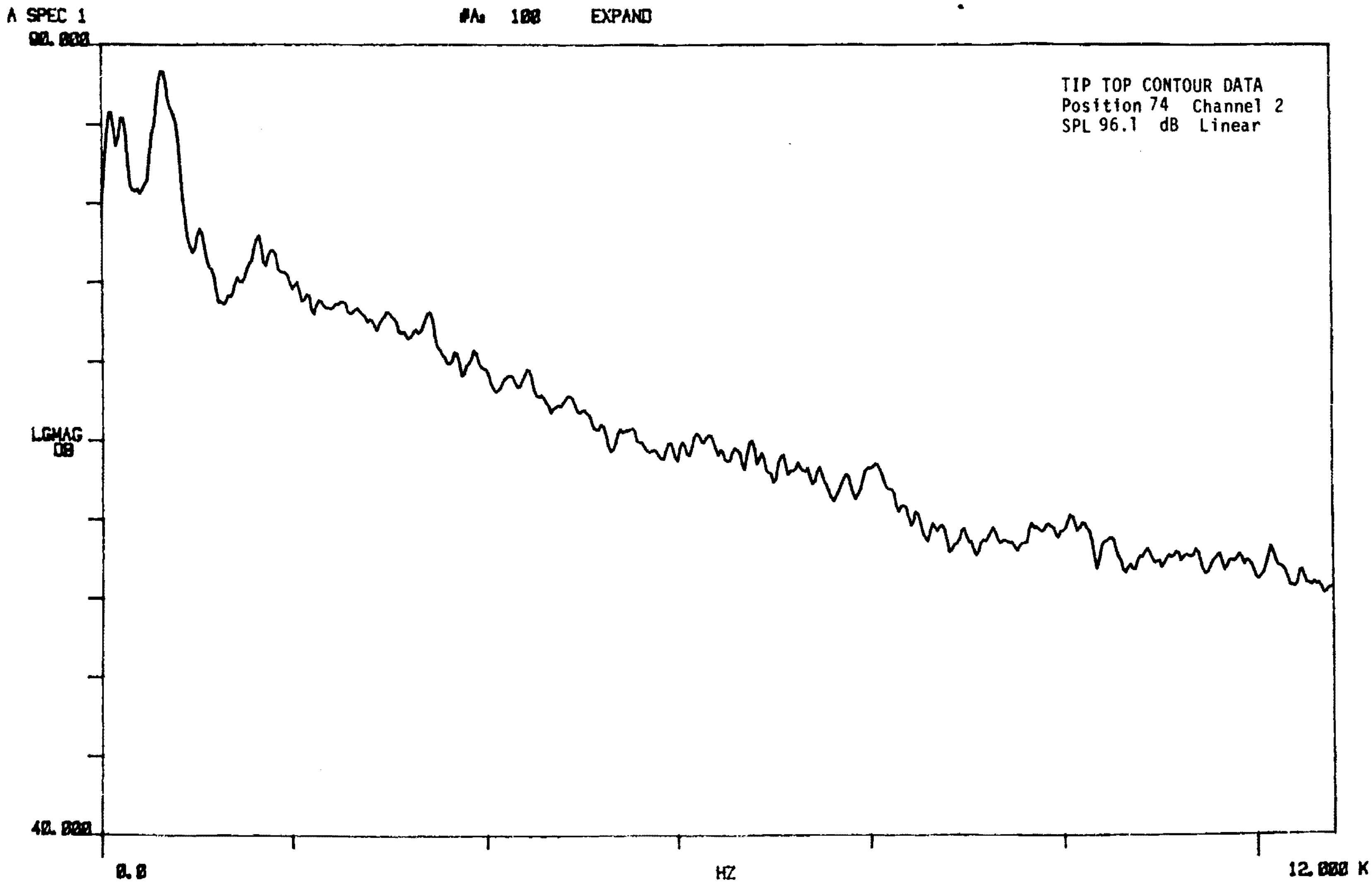
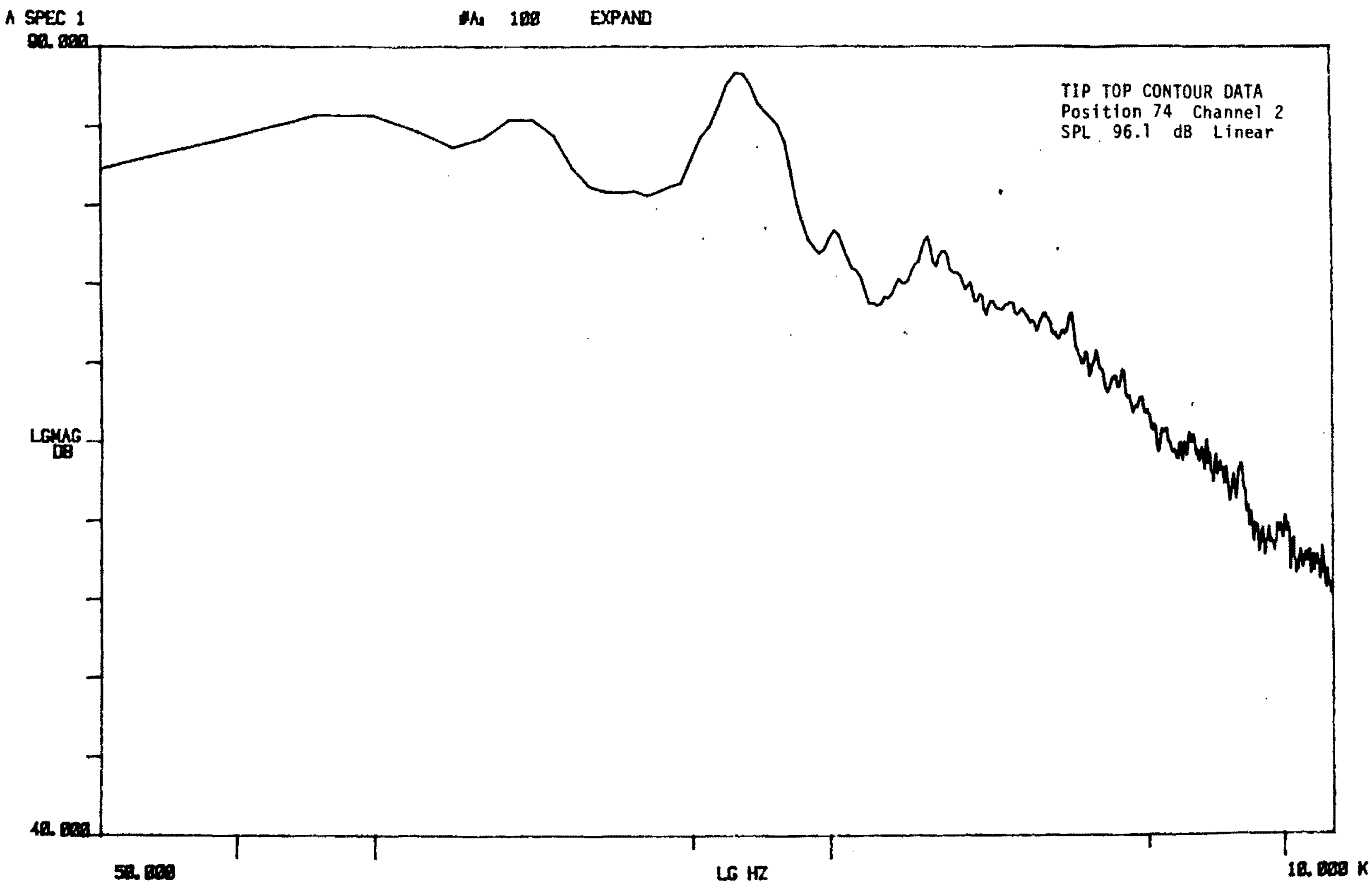
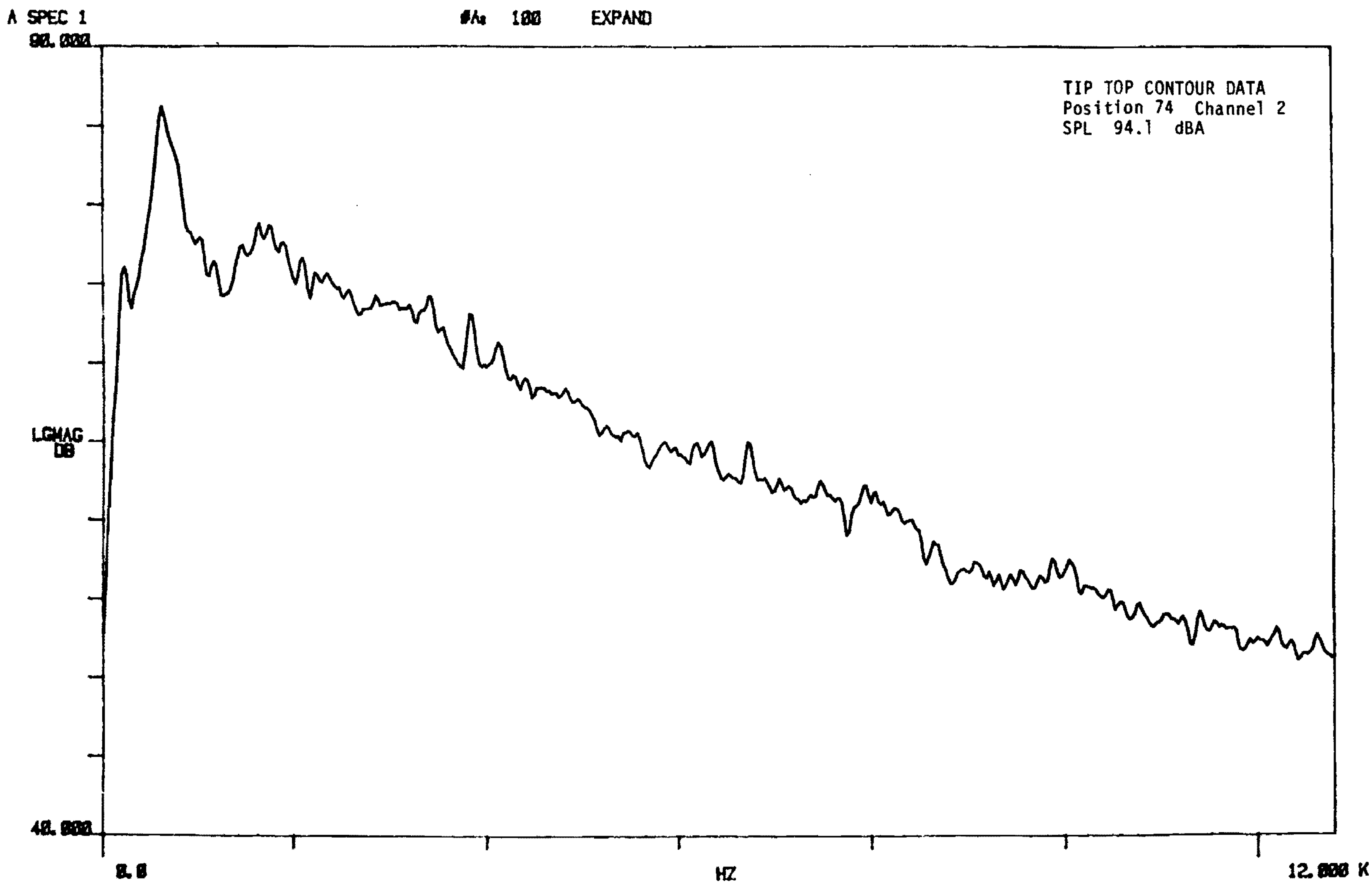

A SPEC 1
#A: 100
EXPAND
90.000
TIP TOP CONTOUR DATA
Position 74 Channel 2
SPL 96.1 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 77B



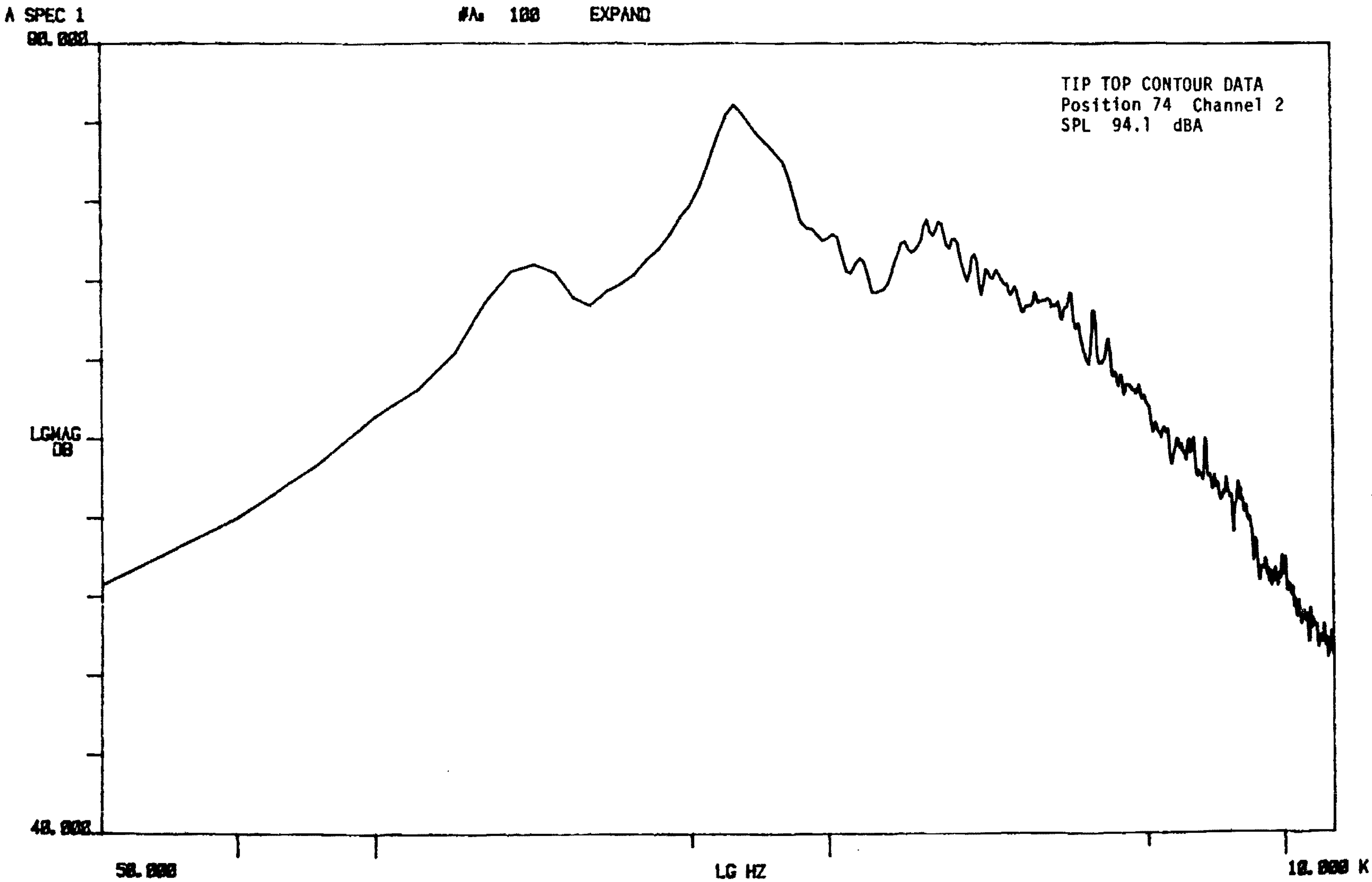
A SPEC 1
#A: 100
EXPAND
90.000
LGMAG DB
40.000
50.000
LG HZ
10.000 K
TIP TOP CONTOUR DATA
Position 74 Channel 2
SPL 96.1 dB Linear

Figure 78B



A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 74 Channel 2
SPL 94.1 dBA
LGMAG DB
40.000
0.0
HZ
12.000 K

Figure 79B



A SPEC 1
#A: 100
EXPAND
80.000
TIP TOP CONTOUR DATA
Position 74 Channel 2
SPL 94.1 dBA
LGMAG
DB
40.000
50.000
LG HZ
10.000 K

Figure 80B

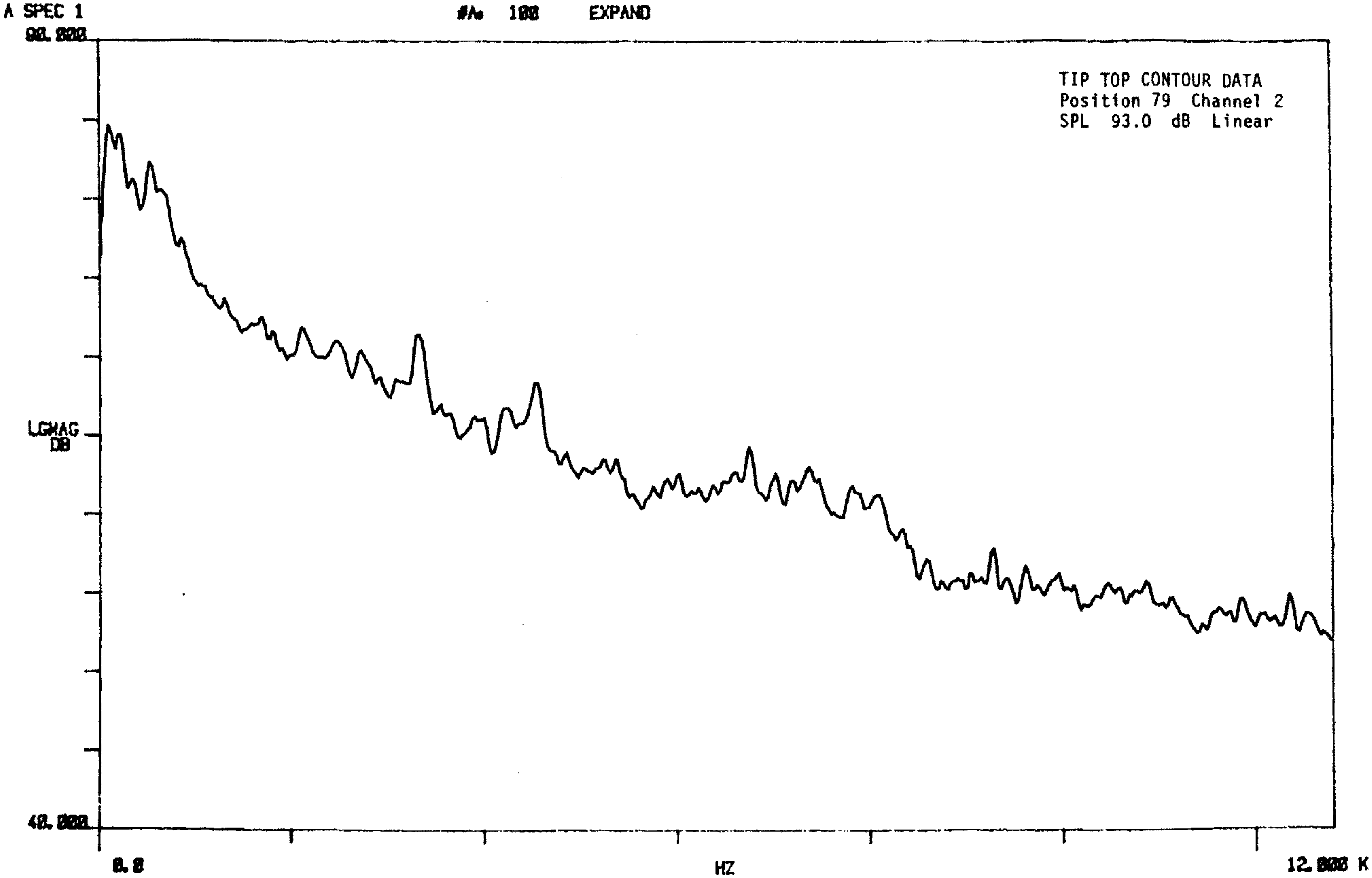
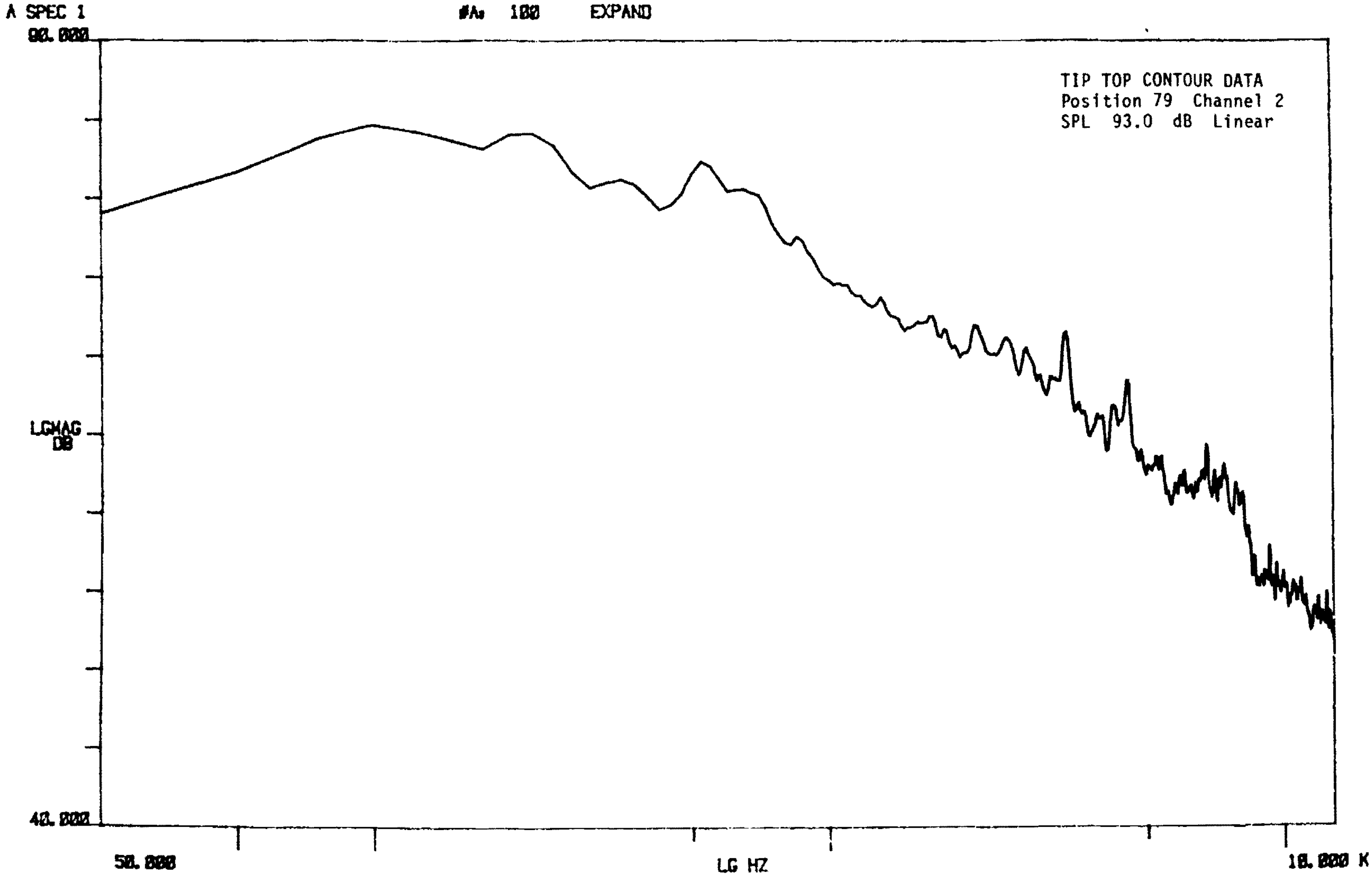
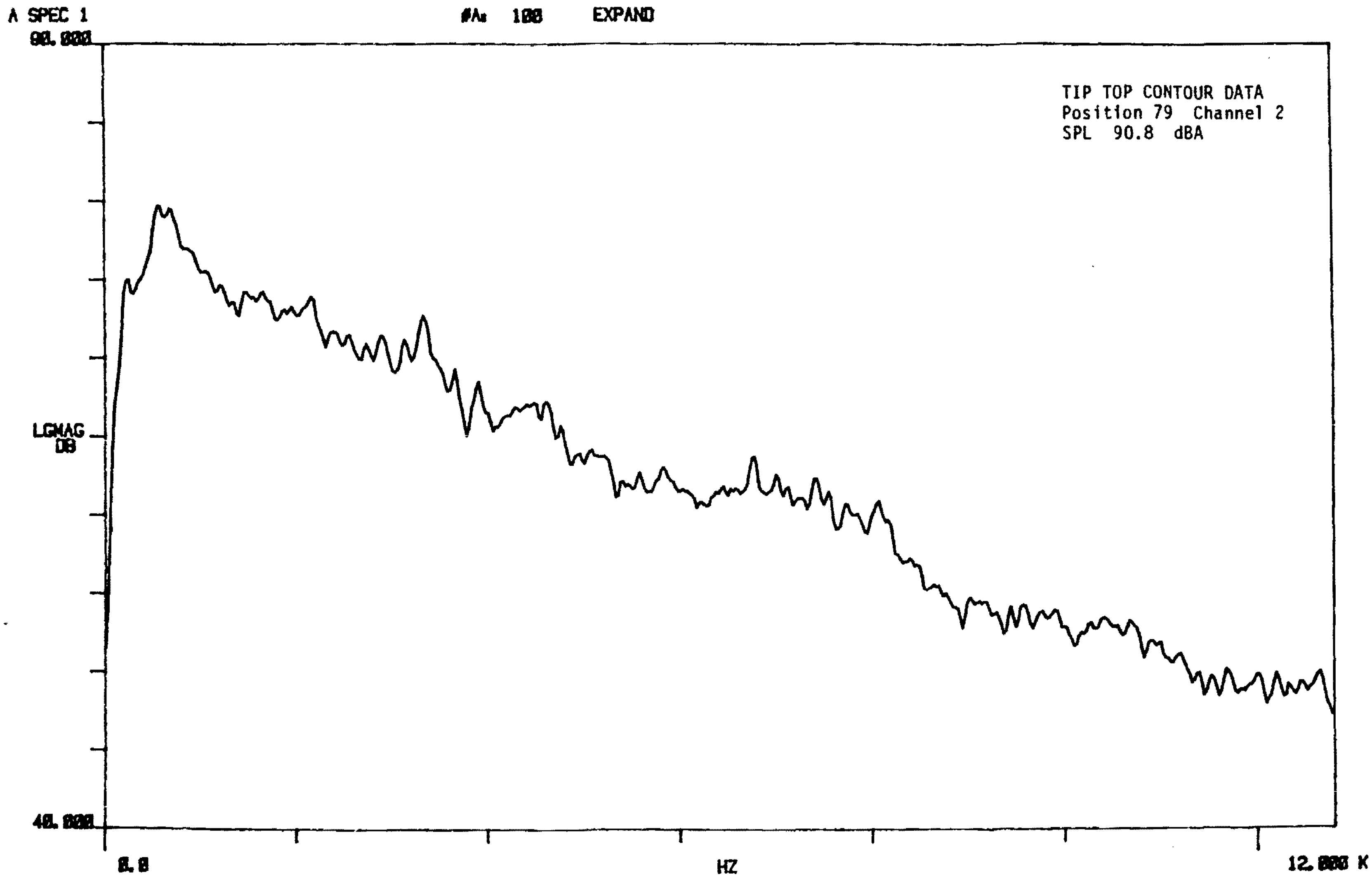

A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 79 Channel 2
SPL 93.0 dB Linear
LGMAG
DB
40.000
0.0
HZ
12.000 K

Figure 81B



A SPEC 1
#A: 100 EXPAND
90.000
TIP TOP CONTOUR DATA
Position 79 Channel 2
SPL 93.0 dB Linear
LGMAG DB
40.000
50.000
LG HZ
10.000 K

Figure 82B



A SPEC 1
#A: 100
EXPAND
90.000
LGMAG DB
40.000
0.0
HZ
12.000 K
TIP TOP CONTOUR DATA
Position 79 Channel 2
SPL 90.8 dBA

Figure 83B

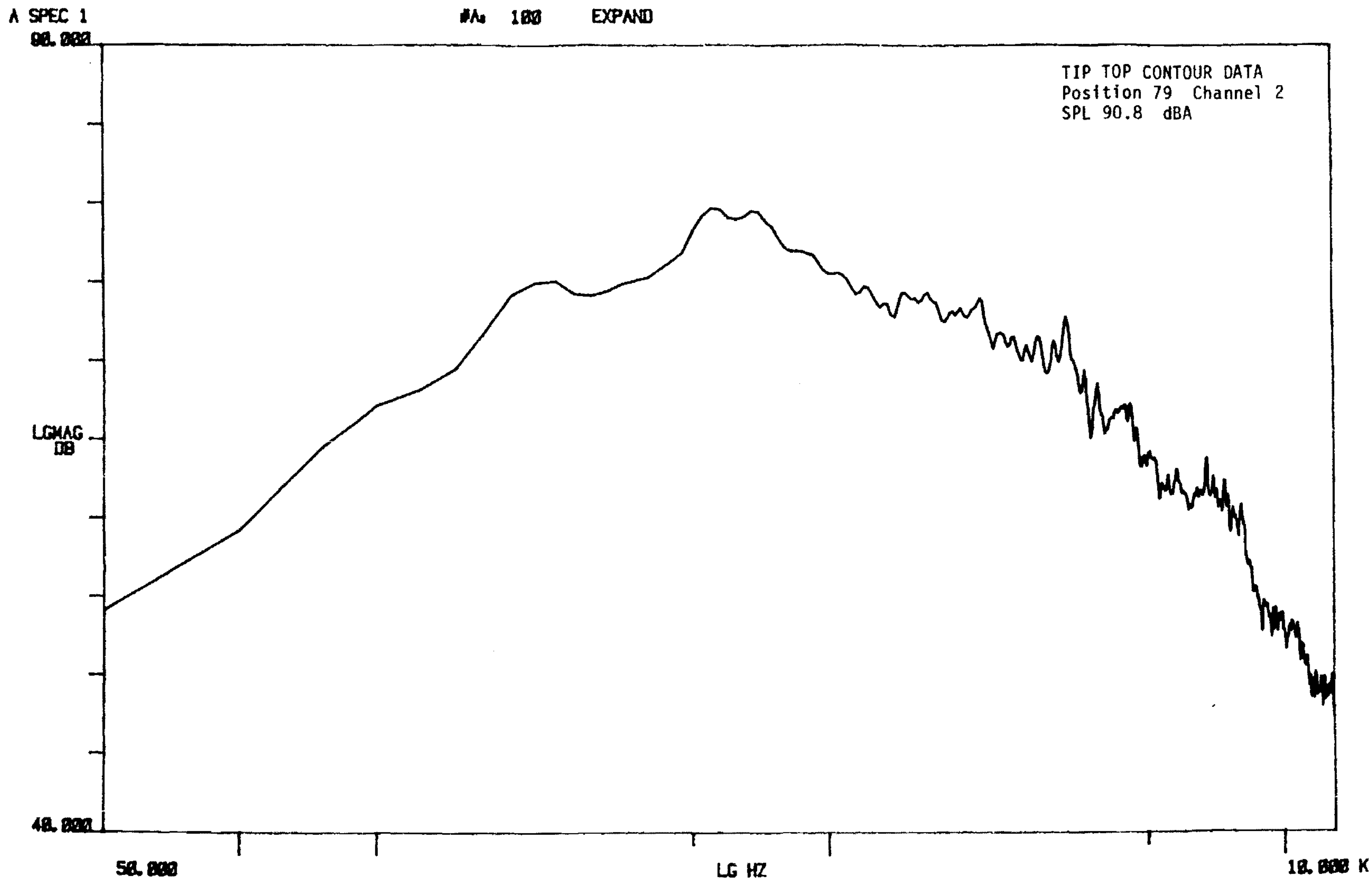

A SPEC 1
#A: 100
EXPAND
80.000
TIP TOP CONTOUR DATA
Position 79 Channel 2
SPL 90.8 dBA
LGMAG DB
40.000
50.000
LG HZ
10.000 K

Figure 84B

Appendix C

OCTAVE BAND ANALYSIS OF DIRECT/REVERBERANT FIELD TEST

OCTAVE BAND ANALYSIS OF DIRECT/REVERBERANT FIELD TEST

The broadband test data gathered in each plant during the direct/reverberant noise field test were octave band analyzed to provide an assessment of the frequency characteristics of the direct and reverberant sound fields associated with the output of the test speaker. The findings are presented in this appendix. They indicate that the reverberant sound field becomes dominant at a distance of only a few feet from the source at all frequency intervals studied.

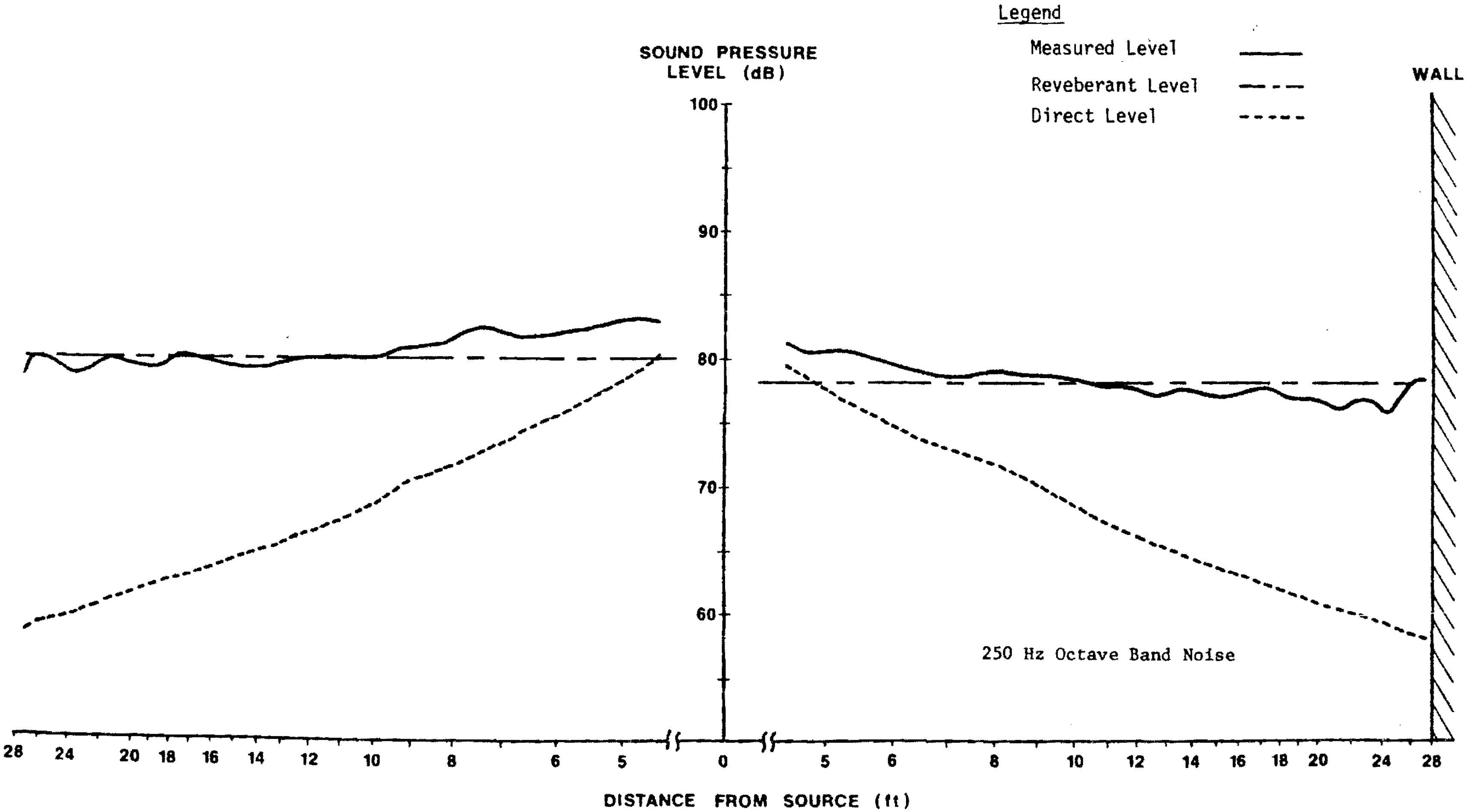


Fig. 1C Direct/Reverberant Noise Fields for Test Speaker
TIP TOP PLANT

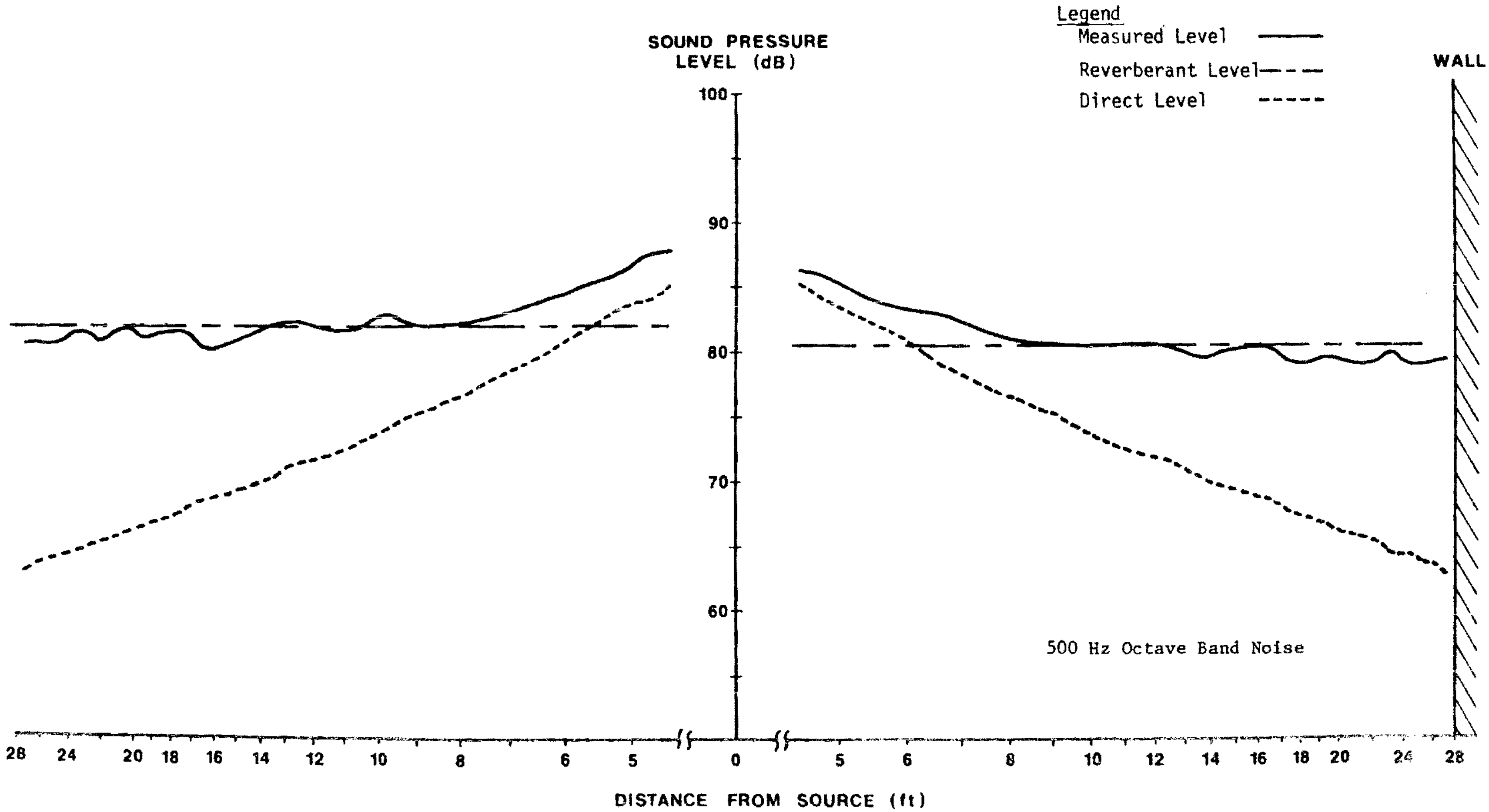


Fig. 2C Direct/Reverberant Noise Fields for Test Speaker

TIP TOP PLANT

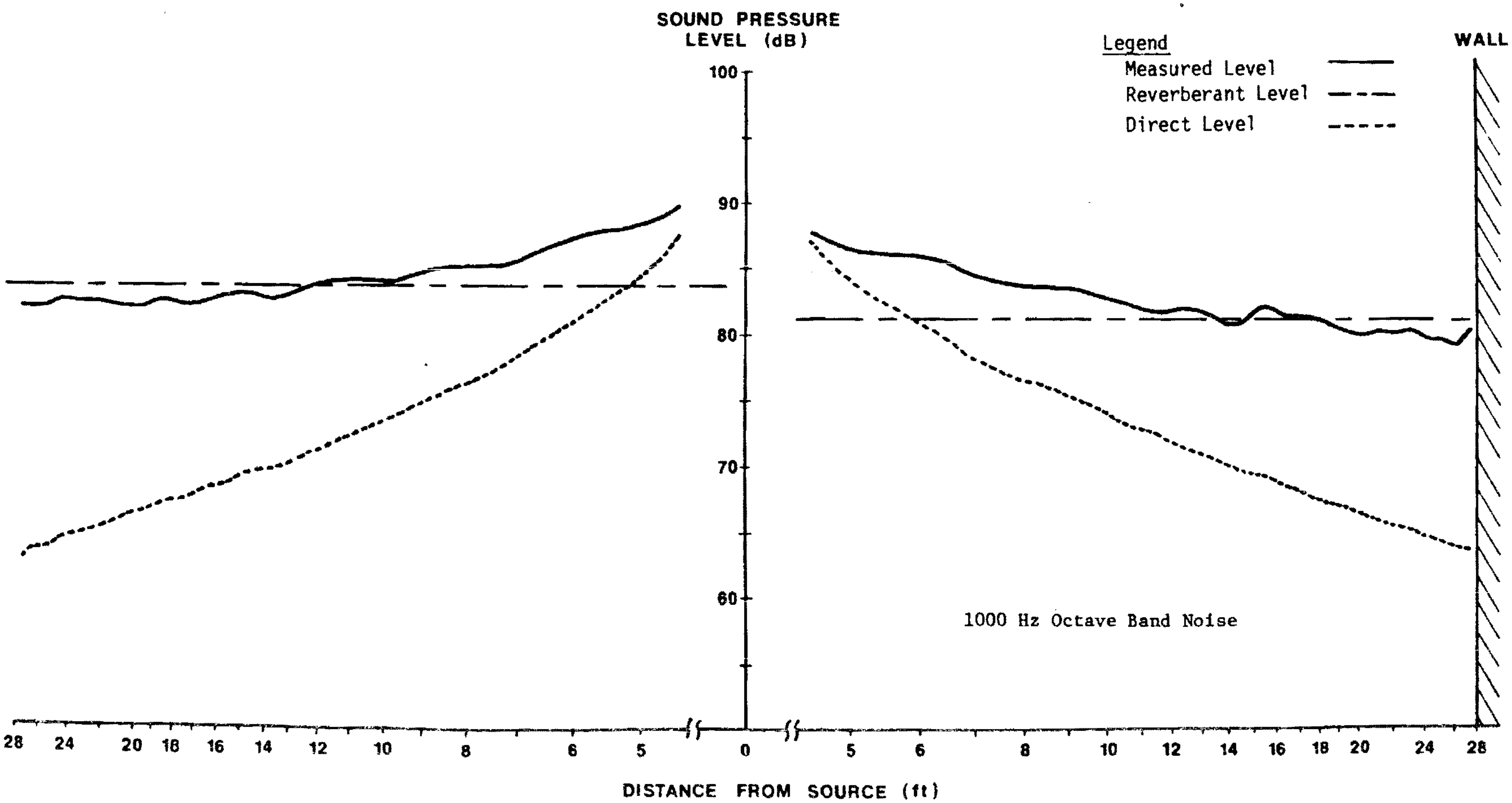


Fig. 3C Direct/Reverberant Noise Fields for Test Speaker

TIP TOP PLANT

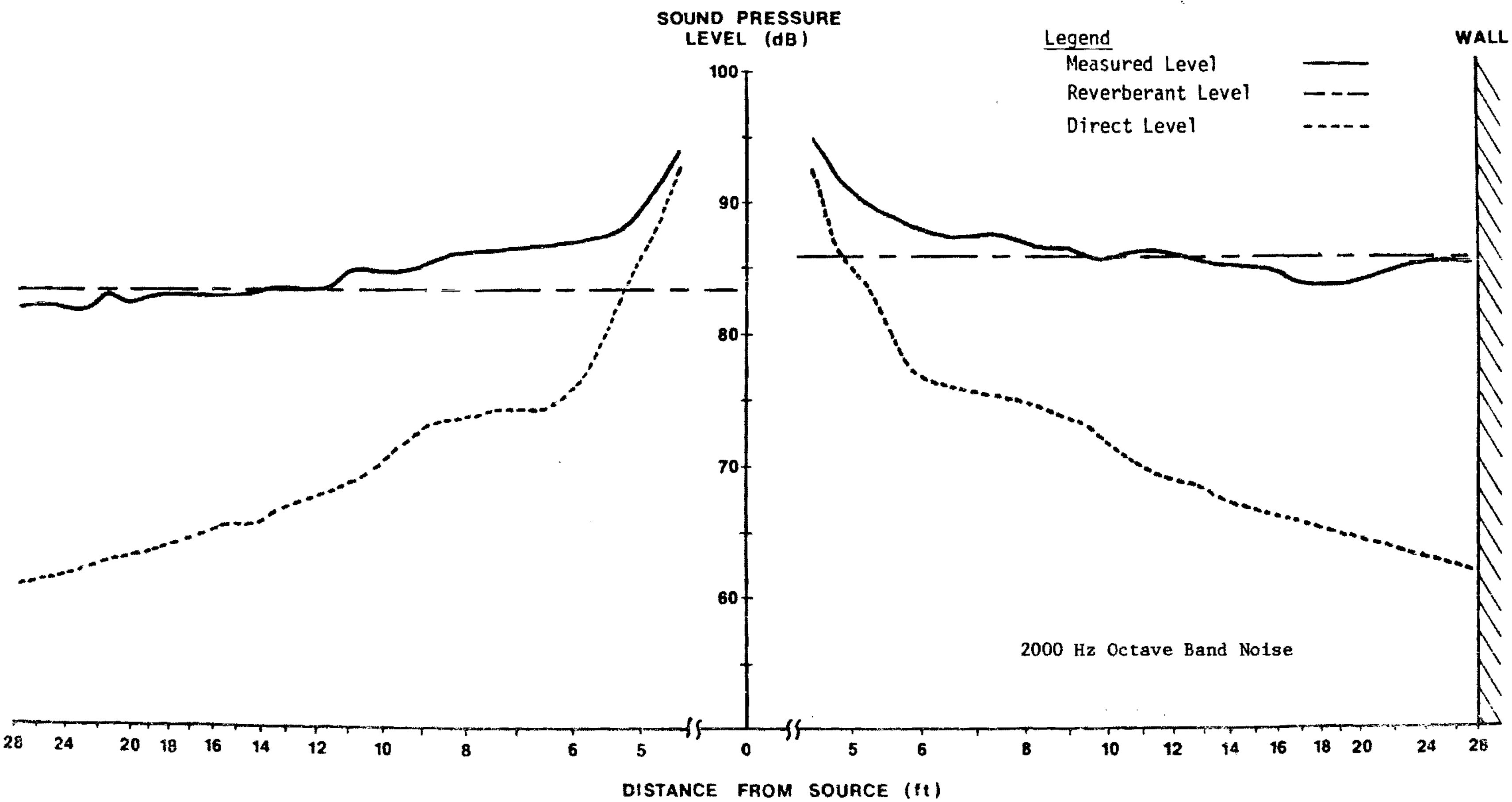


Fig. 4C Direct/Reverberant Noise Fields for Test Speaker
TIP TOP PLANT

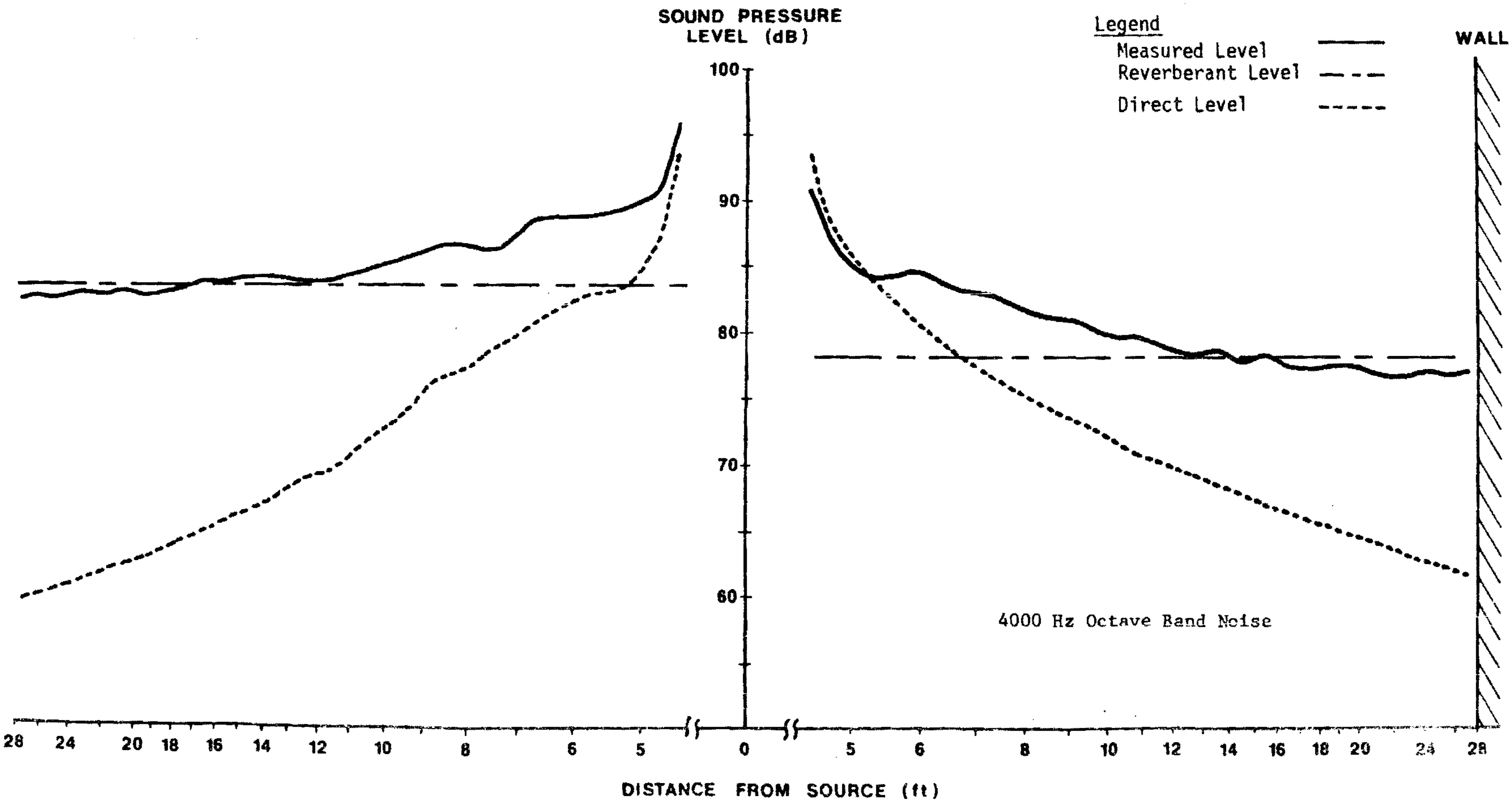


Fig. 5C Direct/Reverberant Noise Fields for Test Speaker

TIP TOP PLANT

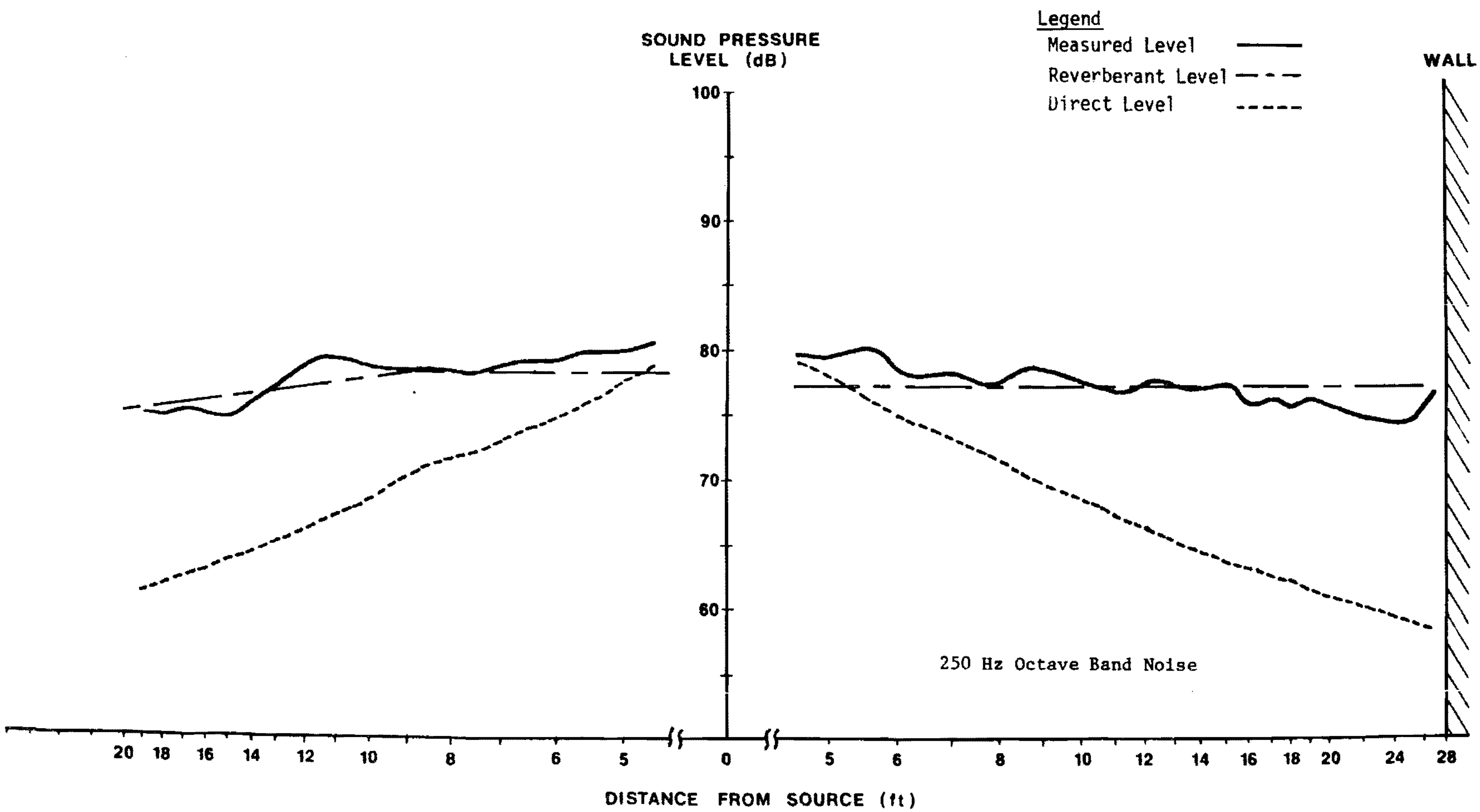


Fig. 6C Direct/Reverberant Noise Fields for Test Speaker
CENTRAL SOYA PLANT

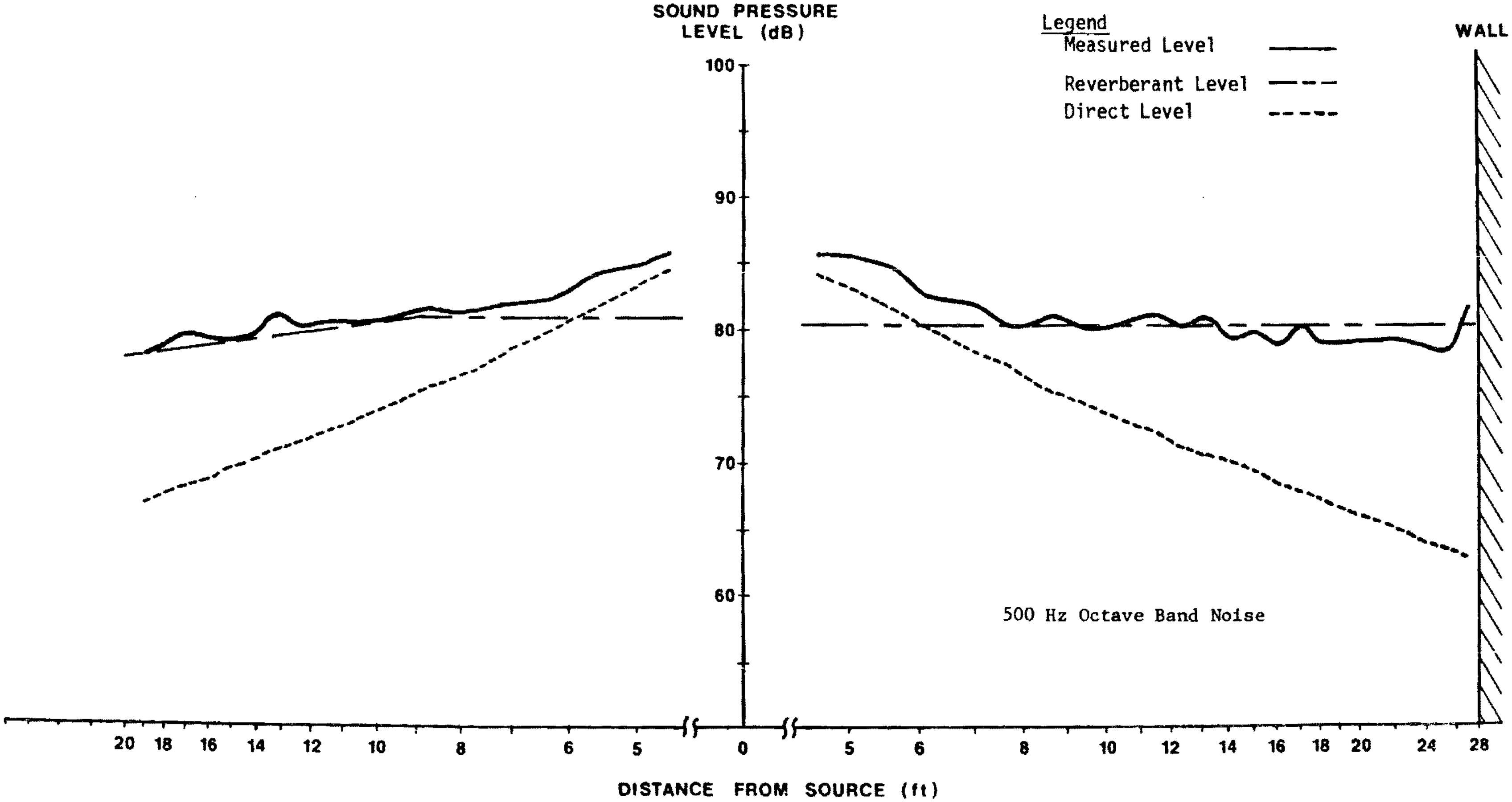


Fig. 7C Direct/Reverberant Noise Fields for Test Speaker
CENTRAL SOYA PLANT

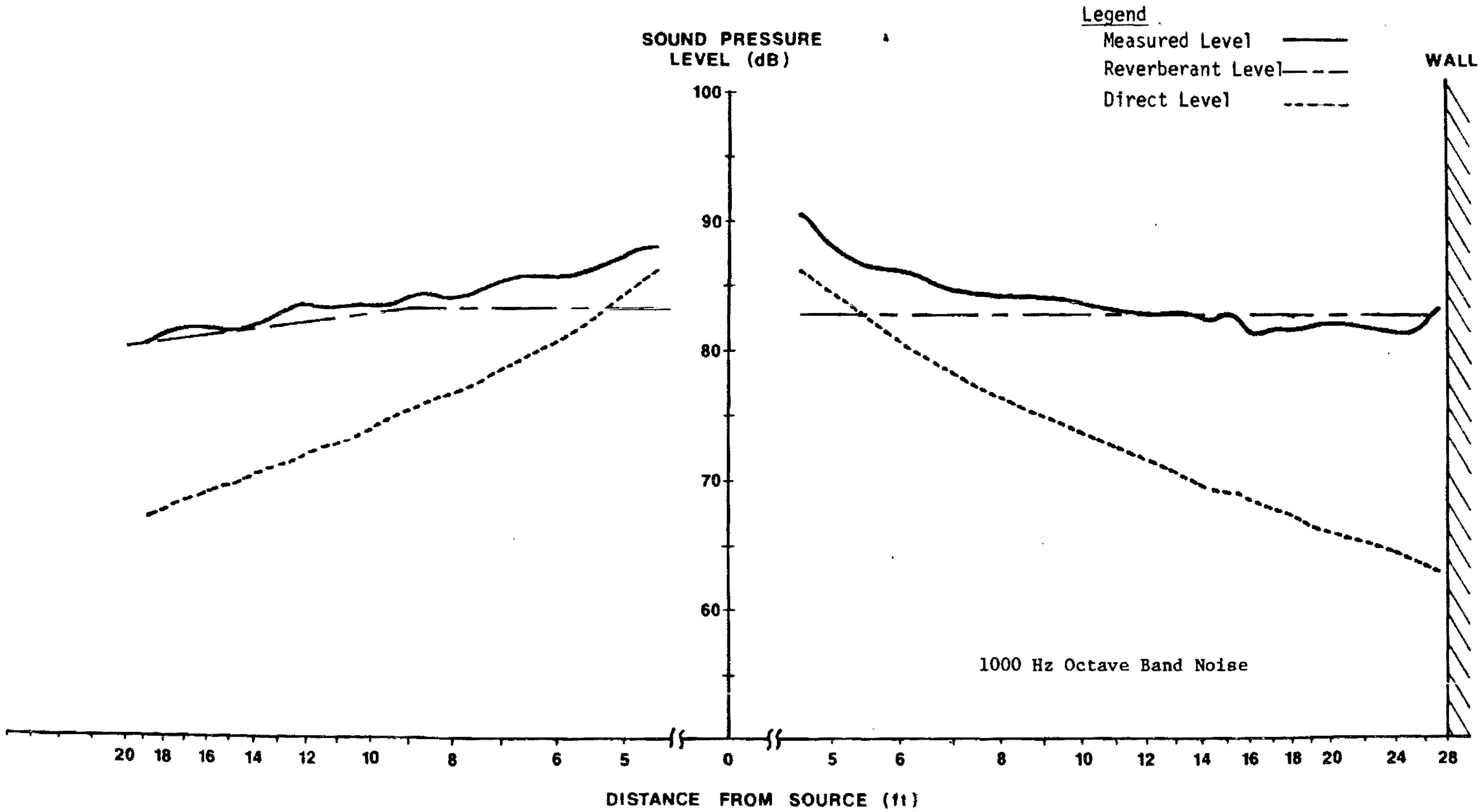


Fig. 8C Direct/Reverberant Noise Fields for Test Speaker

CENTRAL SOYA PLANT

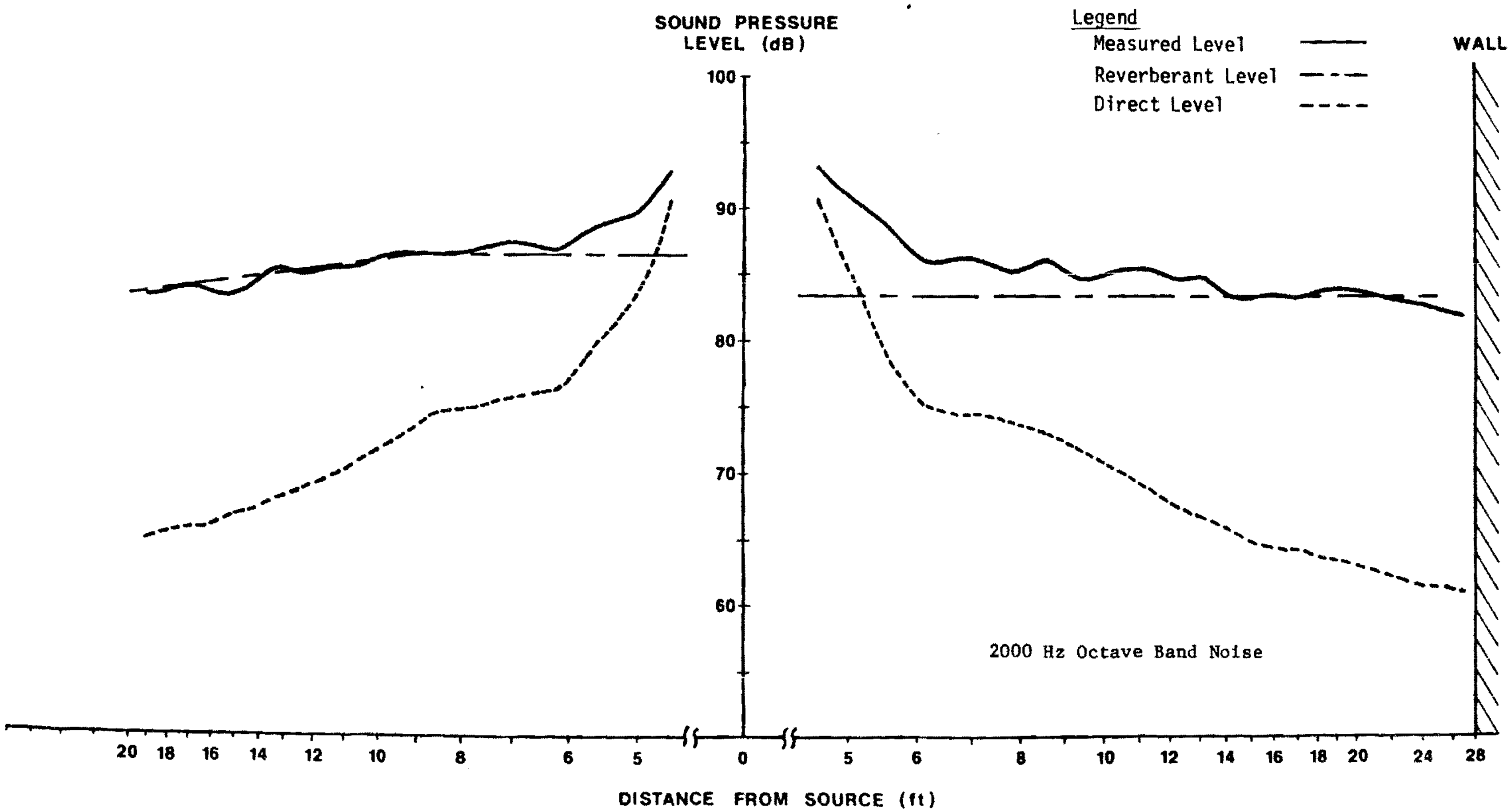


Fig. 9C Direct/Reverberant Noise Fields for Test Speaker
CENTRAL SOYA PLANT

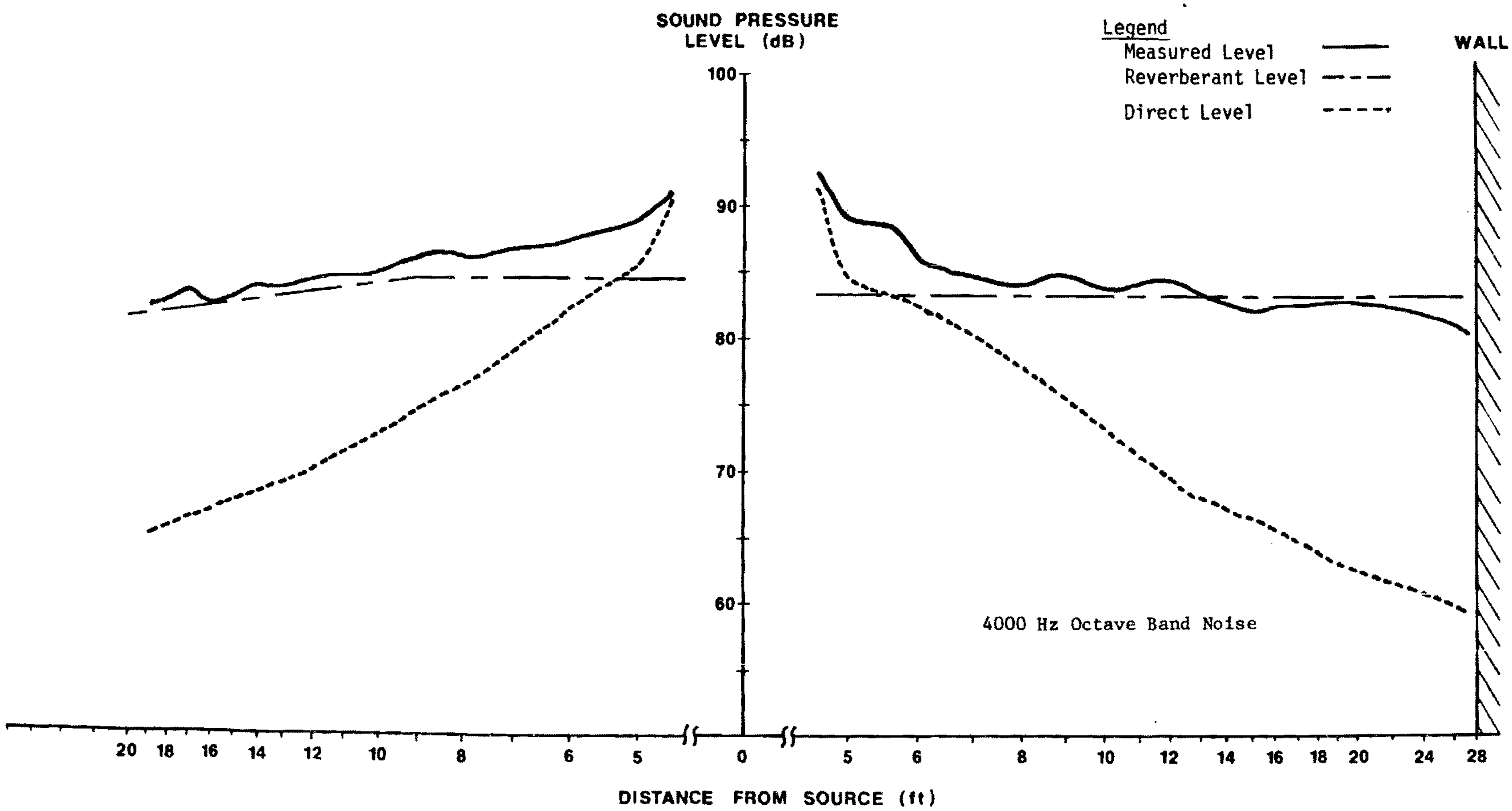


Fig. 10C Direct/Reverberant Noise Fields for Test Speaker
CENTRAL SOYA PLANT

Appendix D

AN ANALYSIS OF INDIVIDUAL SOURCE CONTRIBUTION CHARACTERISTICS

AN ANALYSIS OF SOURCE FREQUENCY CHARACTERISTICS

The data gathered to evaluate the contribution of various sources to the observed sound pressure level at a point in each plant were also analyzed for frequency content. This was done to distinguish qualities about the sources which might be useful in any subsequent source abatement efforts. Unfortunately, as mentioned in the text, the data must be reviewed very carefully since the measurements were taken with some of the sources operating under conditions which were other than typical.

Regarding the Central Soya plant sources, the circulating fans are very close to being a major source in this area of the plant. While they are not always operated, when they are they could still go essentially undetected under normal operations because of their nearness to the lung guns. The spray wash station, on the other hand, shows level peaks which reach significant proportions and appear to contribute significantly to a 350Hz peak in the operating data taken at this point. The detected source of these peaks is a series of restrictor valves in the water system, valves which are commonly used throughout the industry. The neck cutter plot is not believed to be characteristic of this device because the blade rubbed on a bare plastic shield without the typical presence of animal fat from the birds to lubricate this contact. And as mentioned in the text, the chillers lacked water and ice, of which the water is probably an attenuator and the ice (through the dump cycle) a source. Figure 11D shows a comparison of the observed levels of the combined sources versus the observed level during normal operations. With the exception of the peaks in the upper frequency range caused by the neck cutter, the two spectra are reasonably similar in shape. The frequency shift of the 350Hz peak on the red plot is believed to be attributable to a higher than normal water line pressure during the individual source testing.

Regarding the Tip Top plant sources, the fans, at least in this area of the plant, are very quiet. But both the hock cutter and the chillers are intense sources which unfortunately during this test are suspected of producing noise levels not typical of those observed under normal operating conditions. Figure 22D seems to bear this out. When a comparison is made between the observed level of the combined sources versus the observed level during normal operations, the former is higher. This is probably again because the chillers were operated without water or ice and because the hock cutter was operated without birds. In addition to level differences, the two spectra also exhibit substantial differences in shape at several points, which further raise questions regarding the representativeness of the source signatures observed from these two machines.

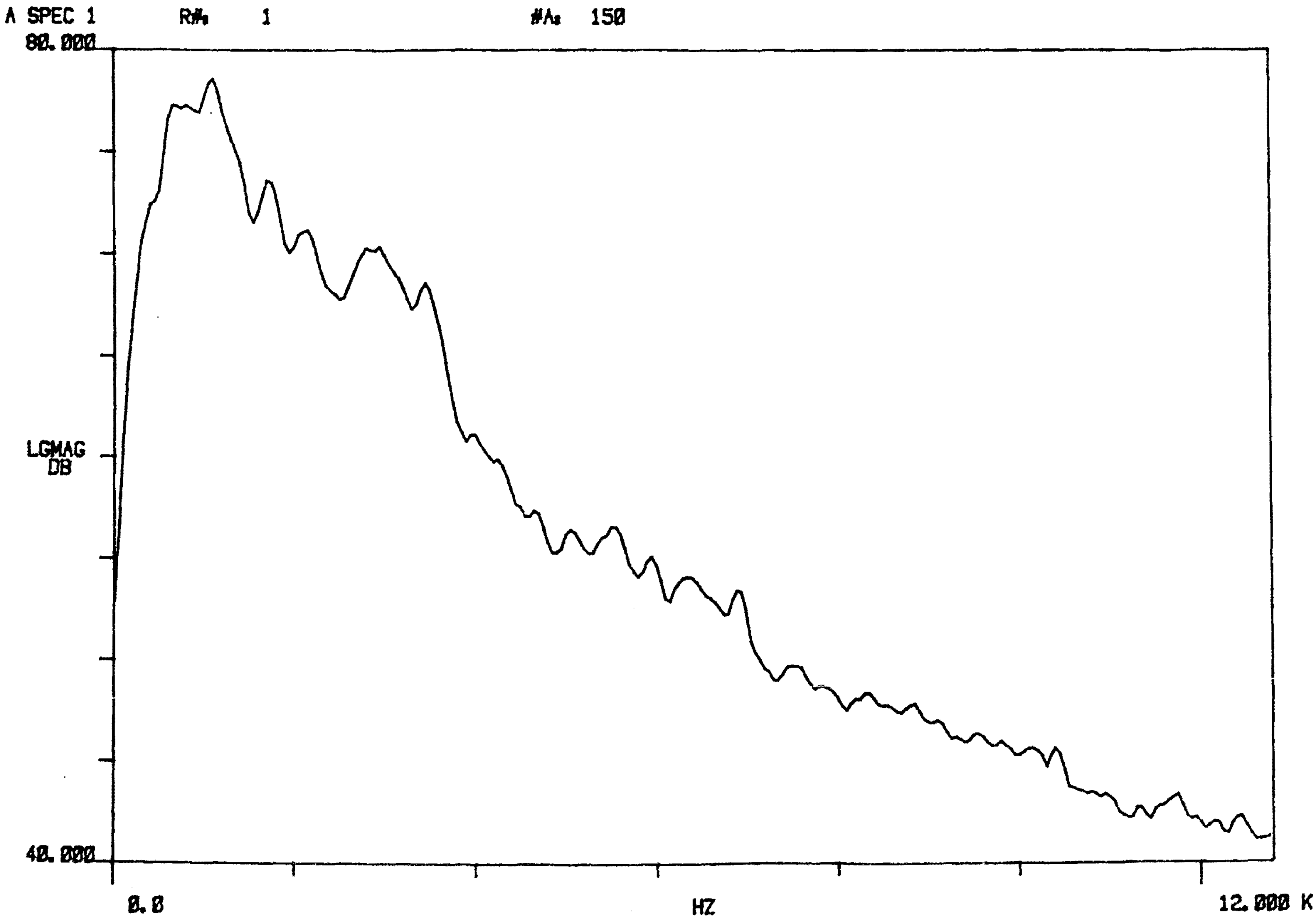


Fig. 1D - A-Weighted Source Contribution Analysis : Circulating Fans
CENTRAL SOYA PLANT (pt. 6B, ch.2)

D-3

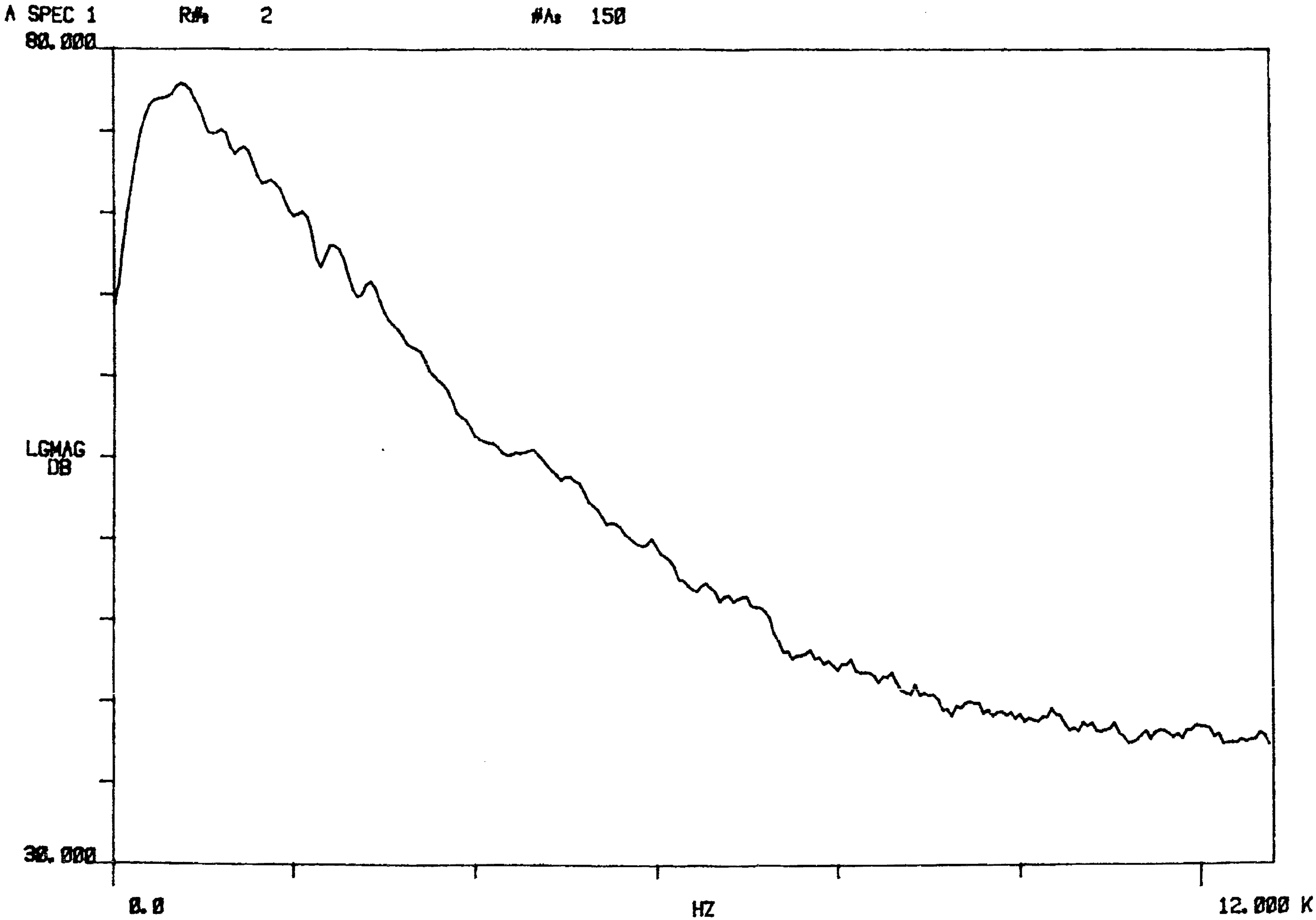


Fig. 2D - A-Weighted Source Contribution Analysis : Chillers
CENTRAL SOYA PLANT (pt. 6B, ch.2)

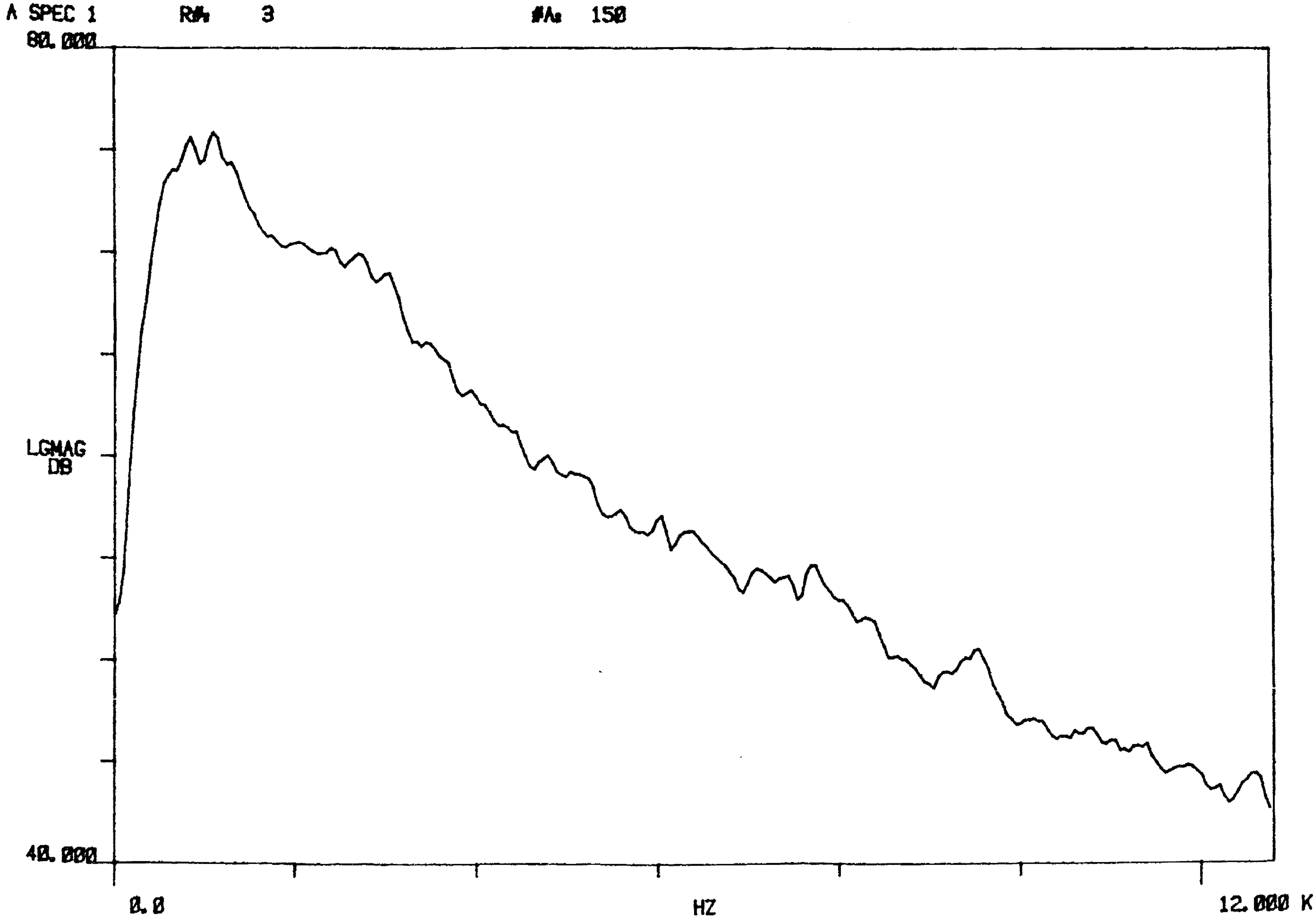


Fig. 3D - A-Weighted Source Contribution Analysis : Gizzard Machine
CENTRAL SOYA PLANT (pt. 6B, ch.2)

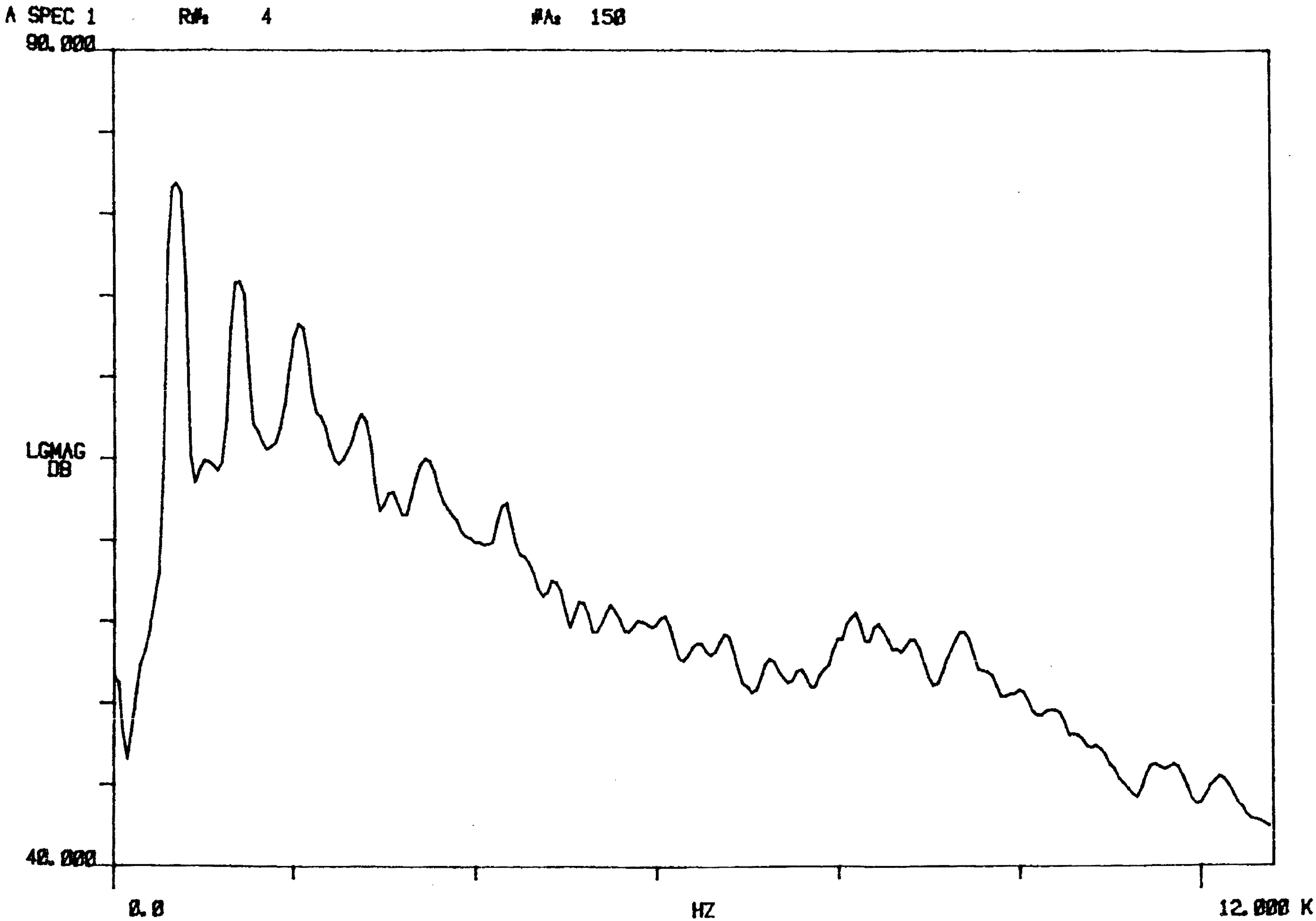


Fig. 4D - A-Weighted Source Contribution Analysis : Spray Washer
CENTRAL SOYA PLANT (pt. 6B, ch.2)

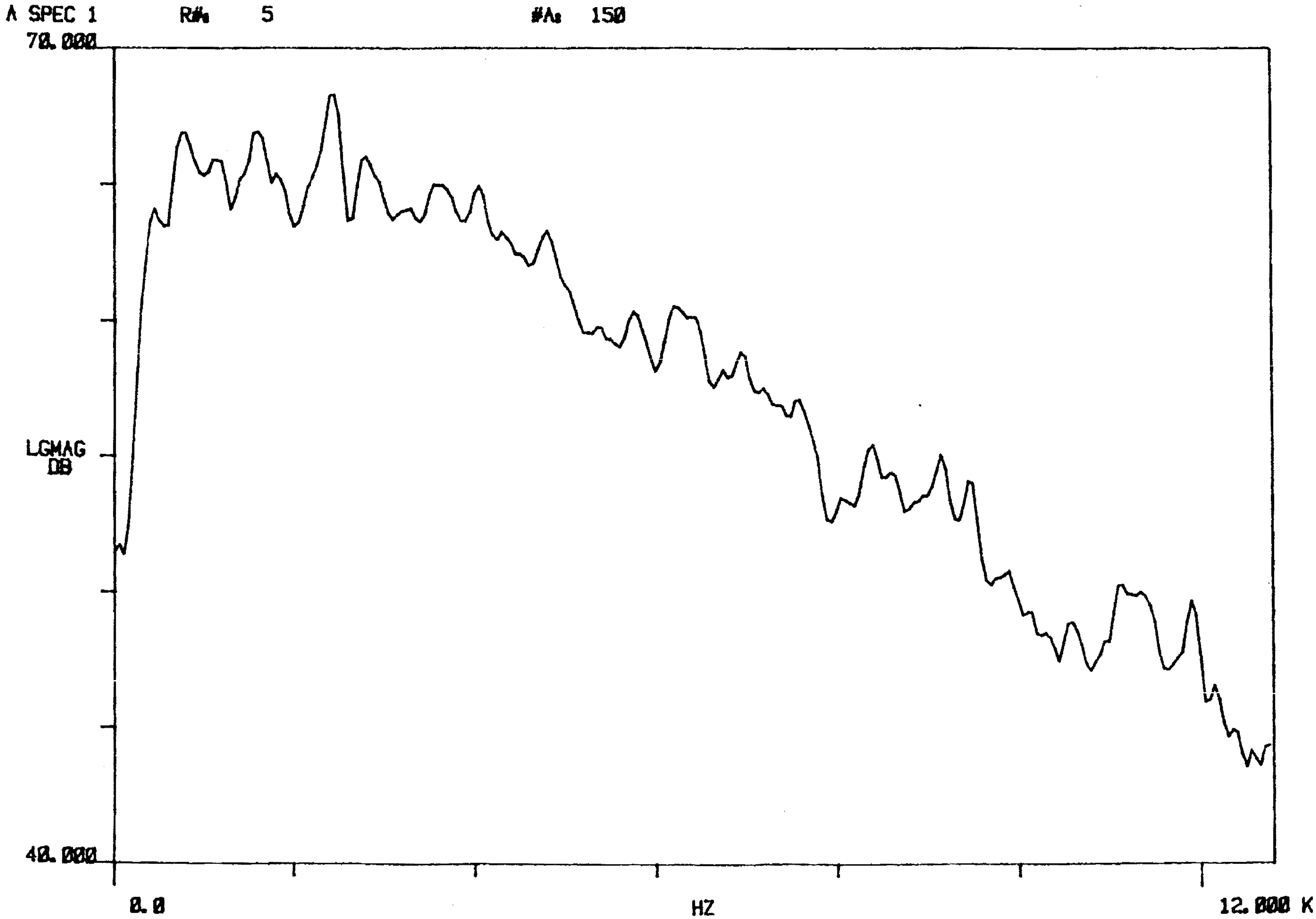


Fig. 5D - A-Weighted Source Contribution Analysis : Shackle Lines
CENTRAL SOYA PLANT (pt. 6B, ch.2)

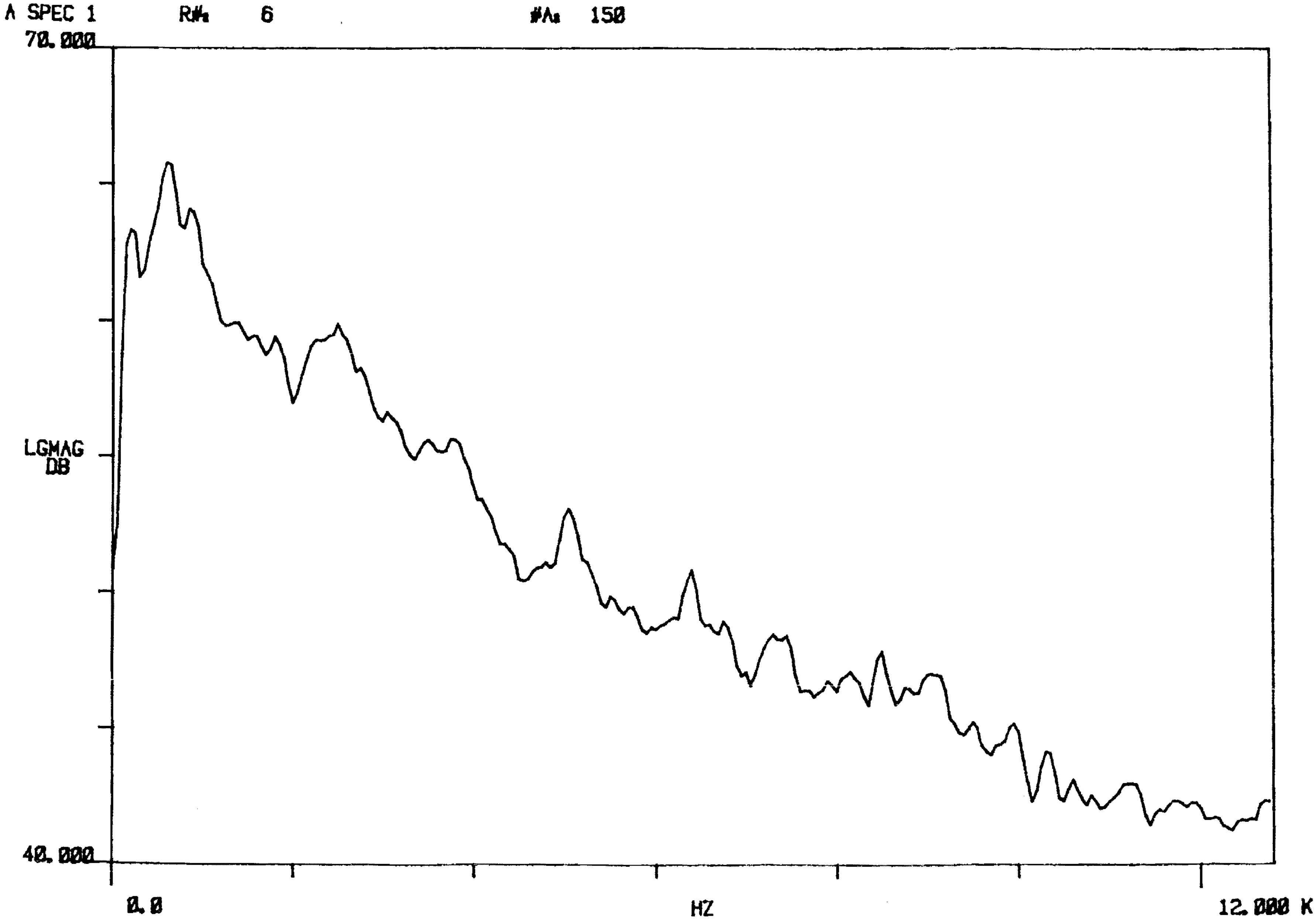


Fig 6D - A-Weighted Source Contribution Analysis : Vacuum Pump
CENTRAL SOYA PLANT (pt. 6B, ch.2)

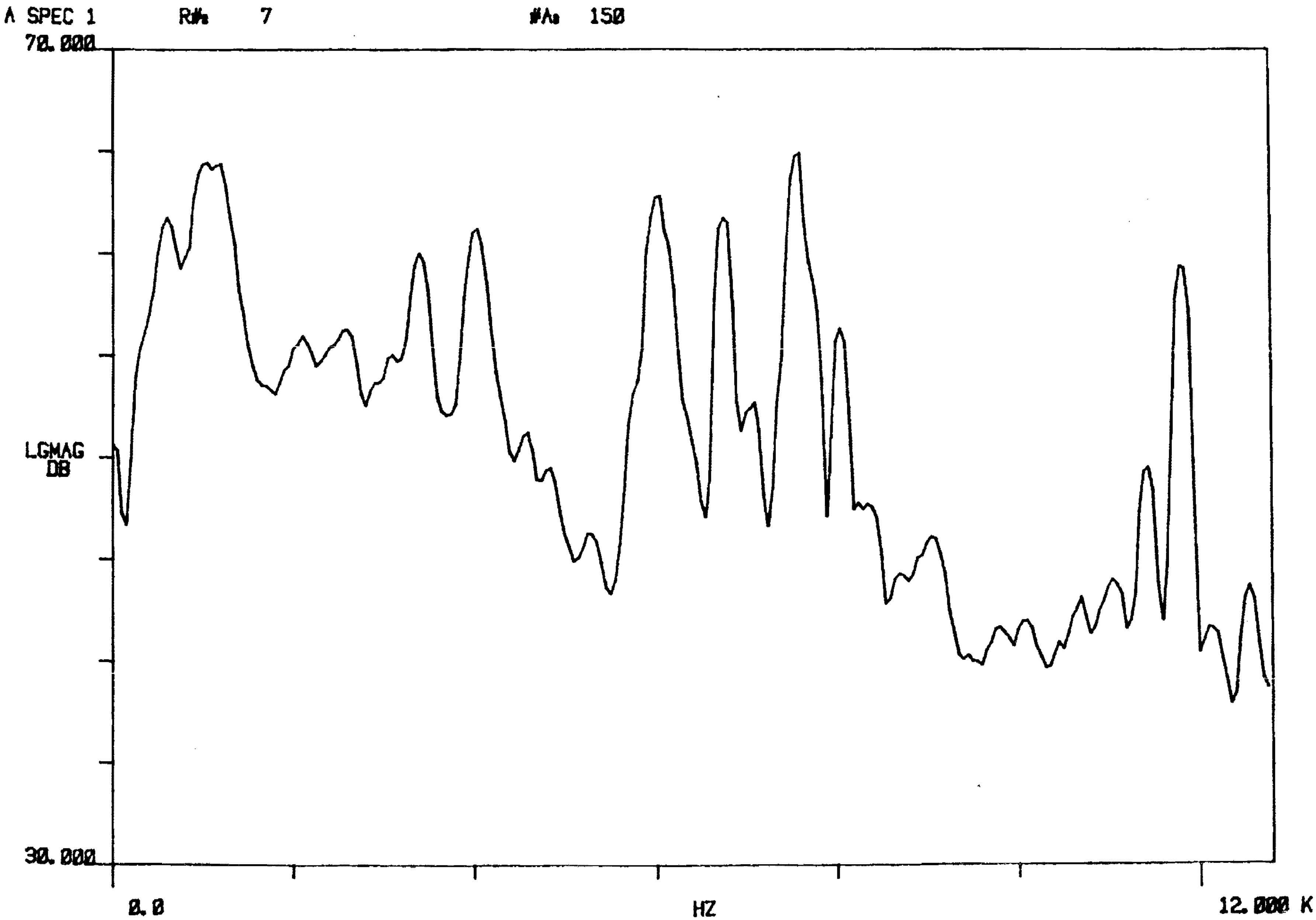


Fig. 7D - A-Weighted Source Contribution Analysis : Neck Cutters
CENTRAL SOYA PLANT (pt. 6B, ch.2)

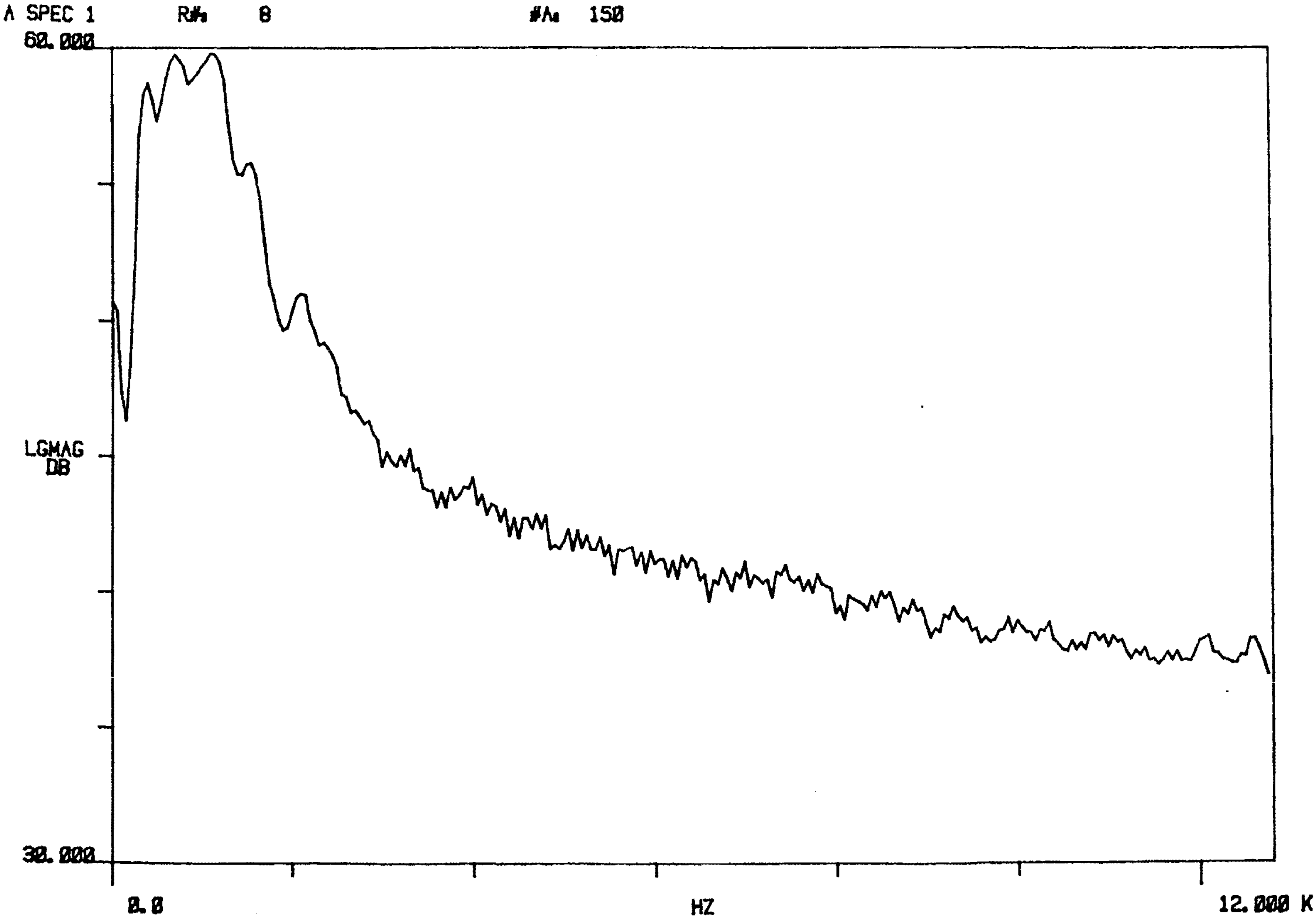


Fig. 8D - A-Weighted Source Contribution Analysis : Pickers
CENTRAL SOYA PLANT (pt. 6B, ch.2)

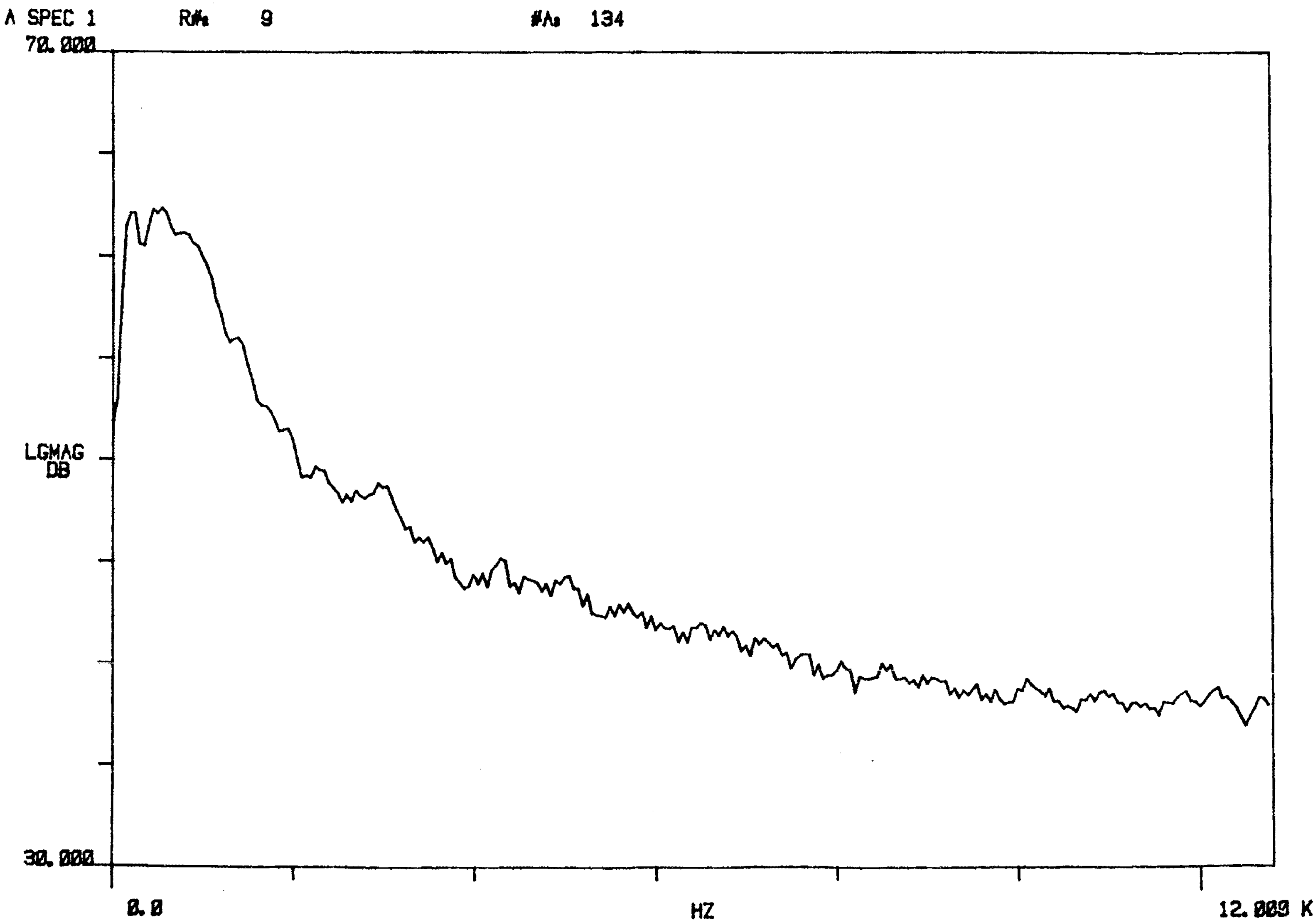


Fig. 9D - A-Weighted Source Contribution Analysis : Waste Vacuum
CENTRAL SOYA PLANT (pt. 6B, ch.2)

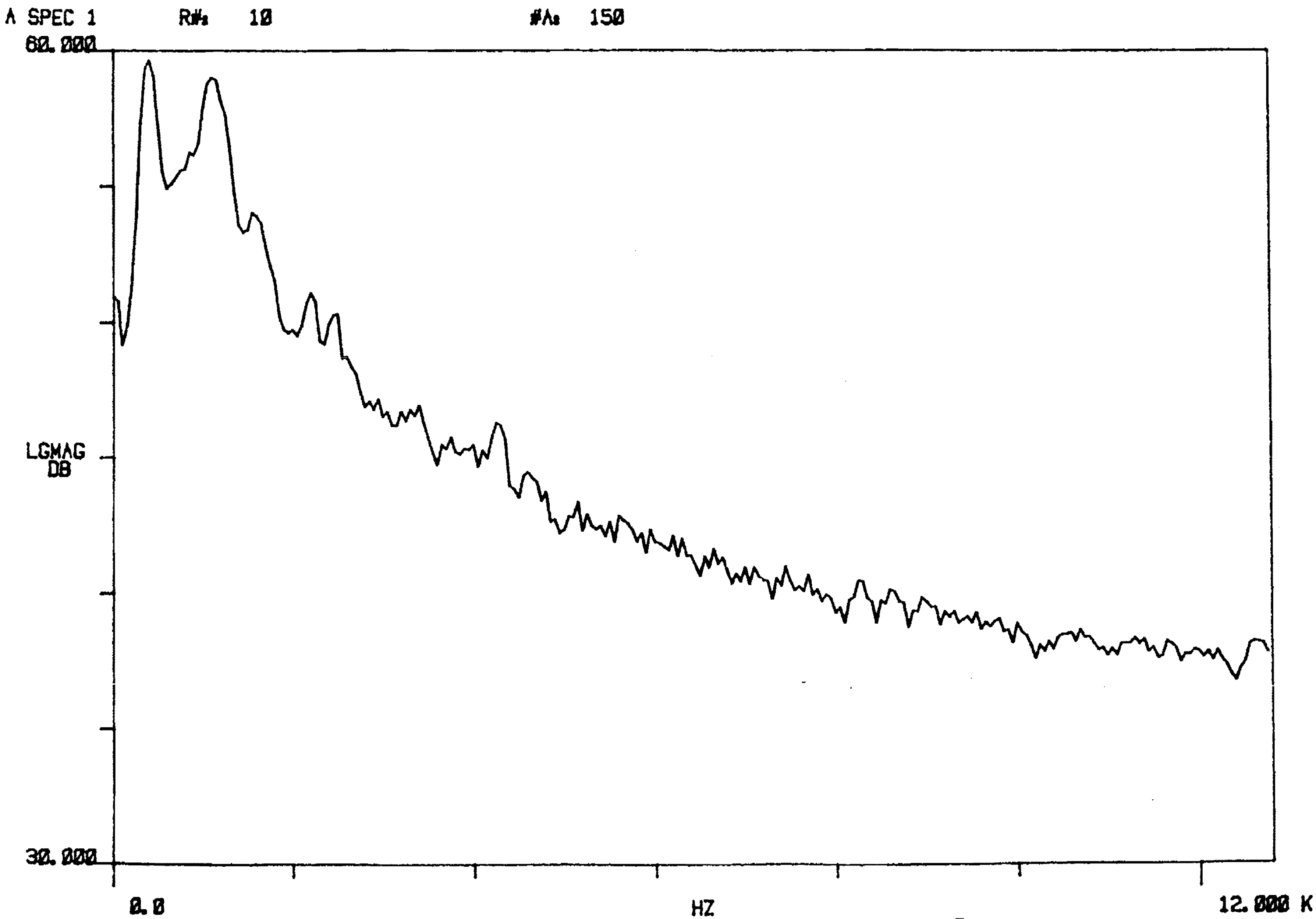


Fig. 10D - A-Weighted Source Contribution Analysis : Exhaust Fans
CENTRAL SOYA PLANT (pt. 6B, ch.2)

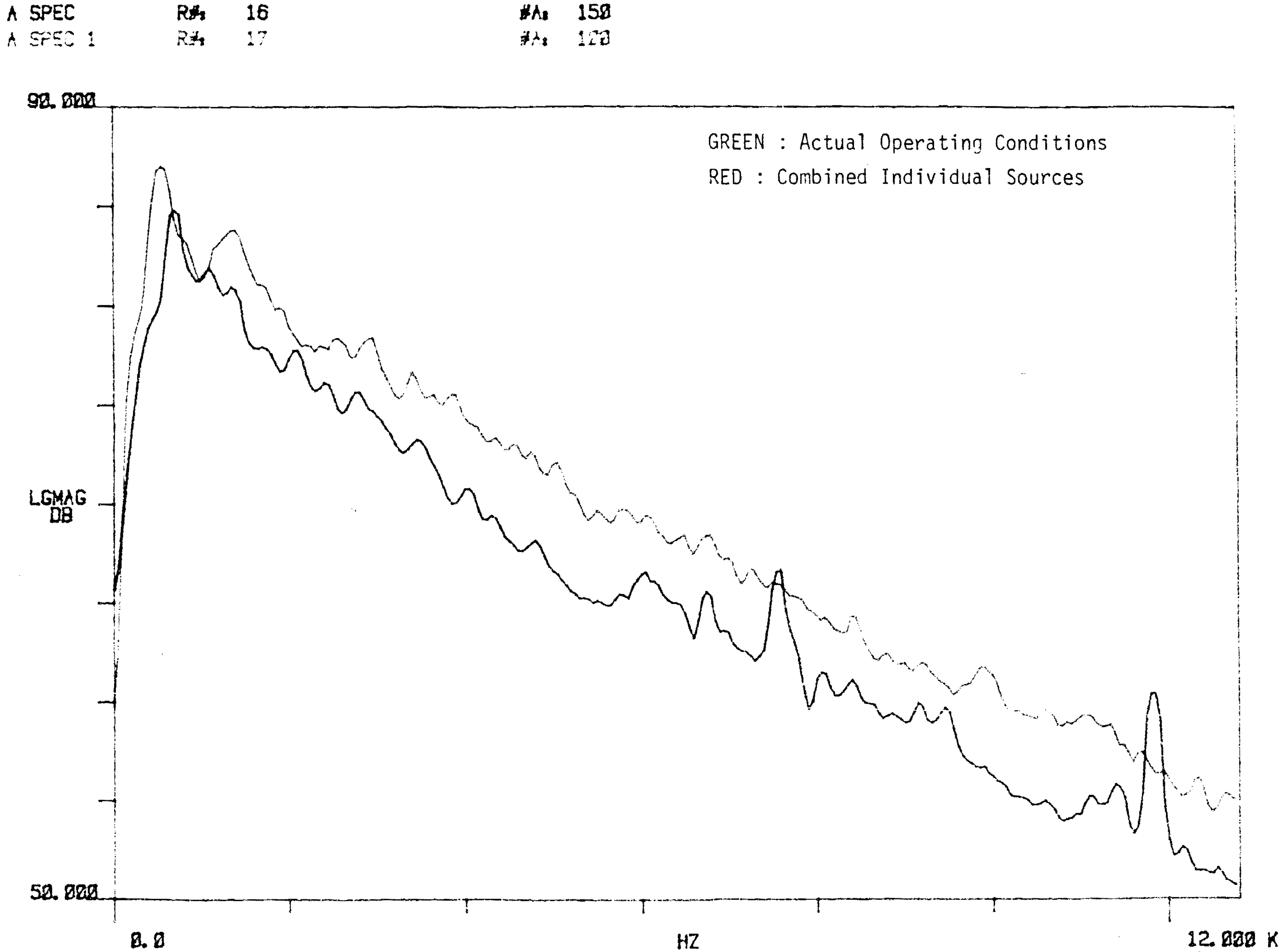


Fig. 11D - A-Weighted Comparison of Combined Individual Sources vs Actual Operating Conditions
CENTRAL SOYA PLANT (pt.6B, ch.2)

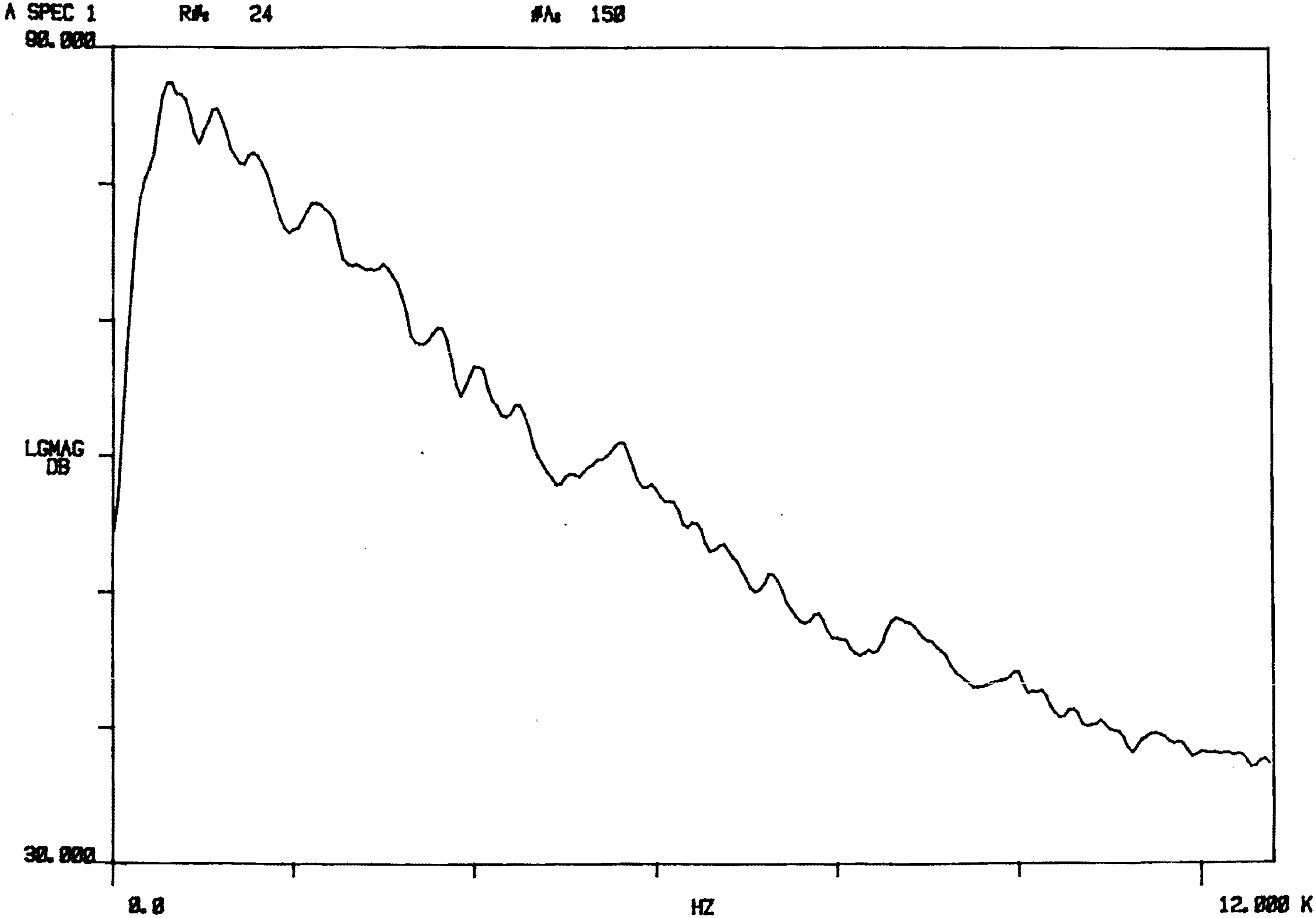


Fig. 12D - A-Weighted Source Contribution Analysis : Hock Cutter
TIP TOP PLANT (pt. 53, ch.2)

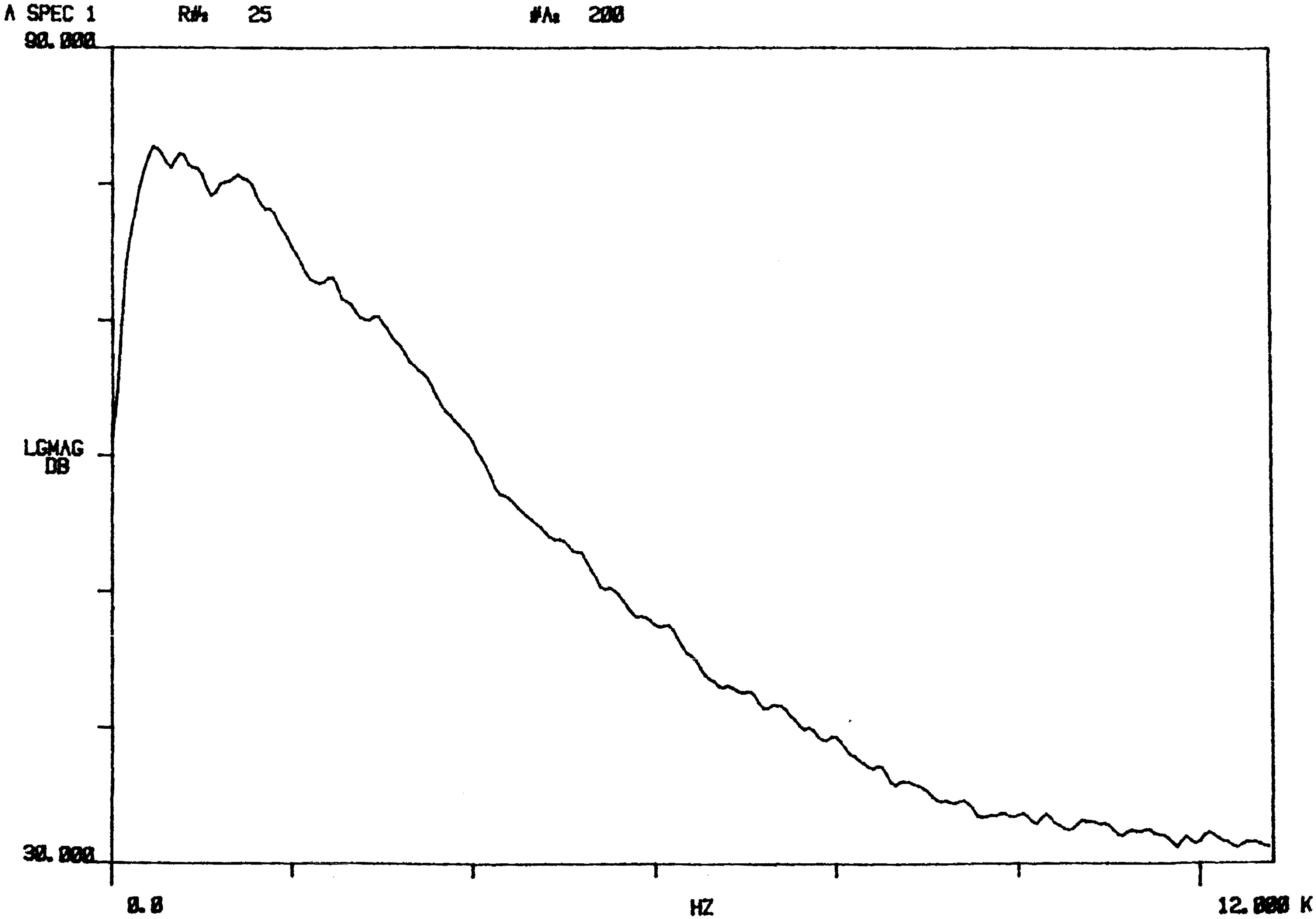


Fig. 13D - A-Weighted Source Contribution Analysis : Chillers
TIP TOP PLANT (pt. 53, ch.2)

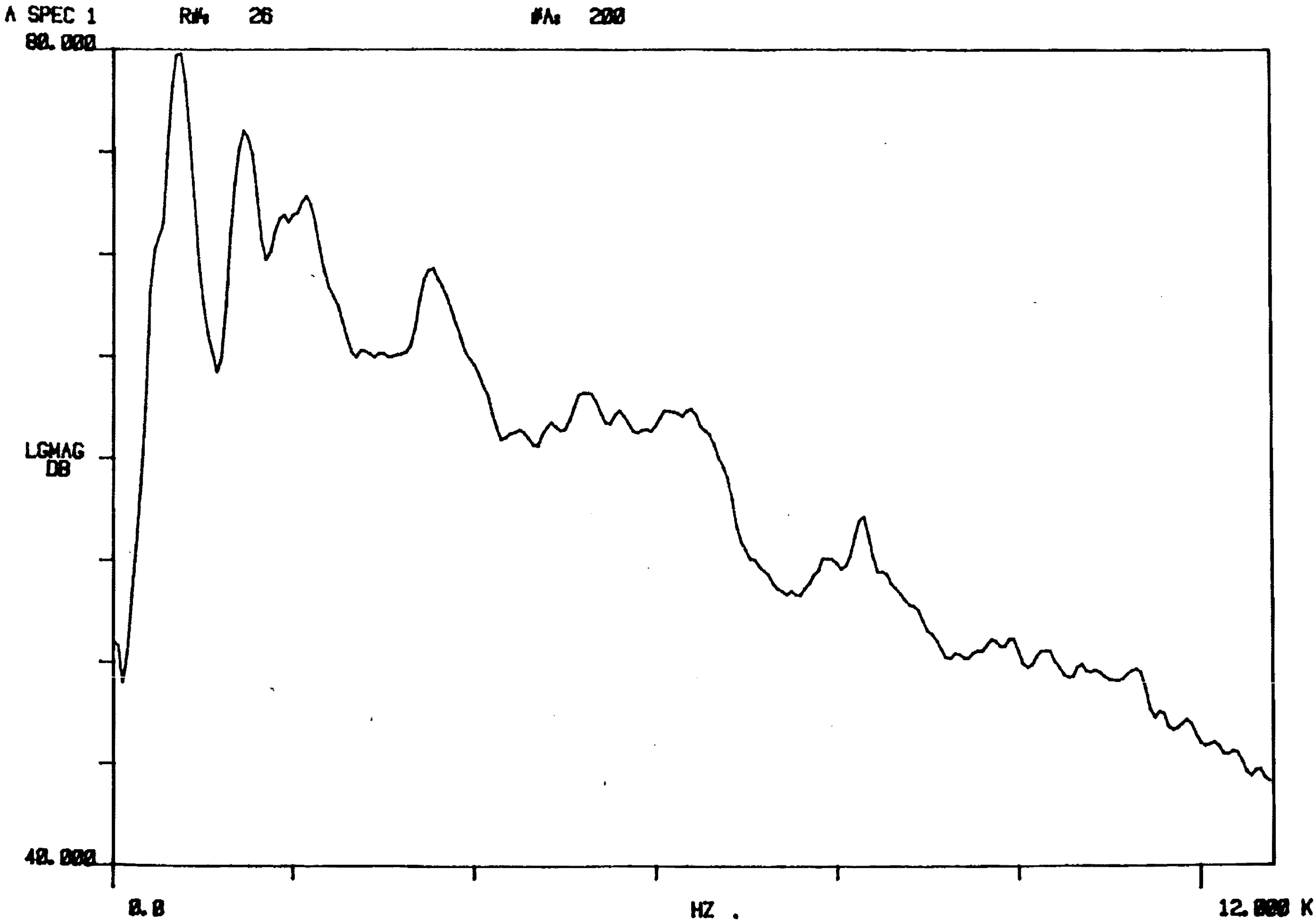


Fig. 14D - A-Weighted Source Contribution Analysis : Vent Cutters
TIP TOP PLANT (pt. 53, ch.2)

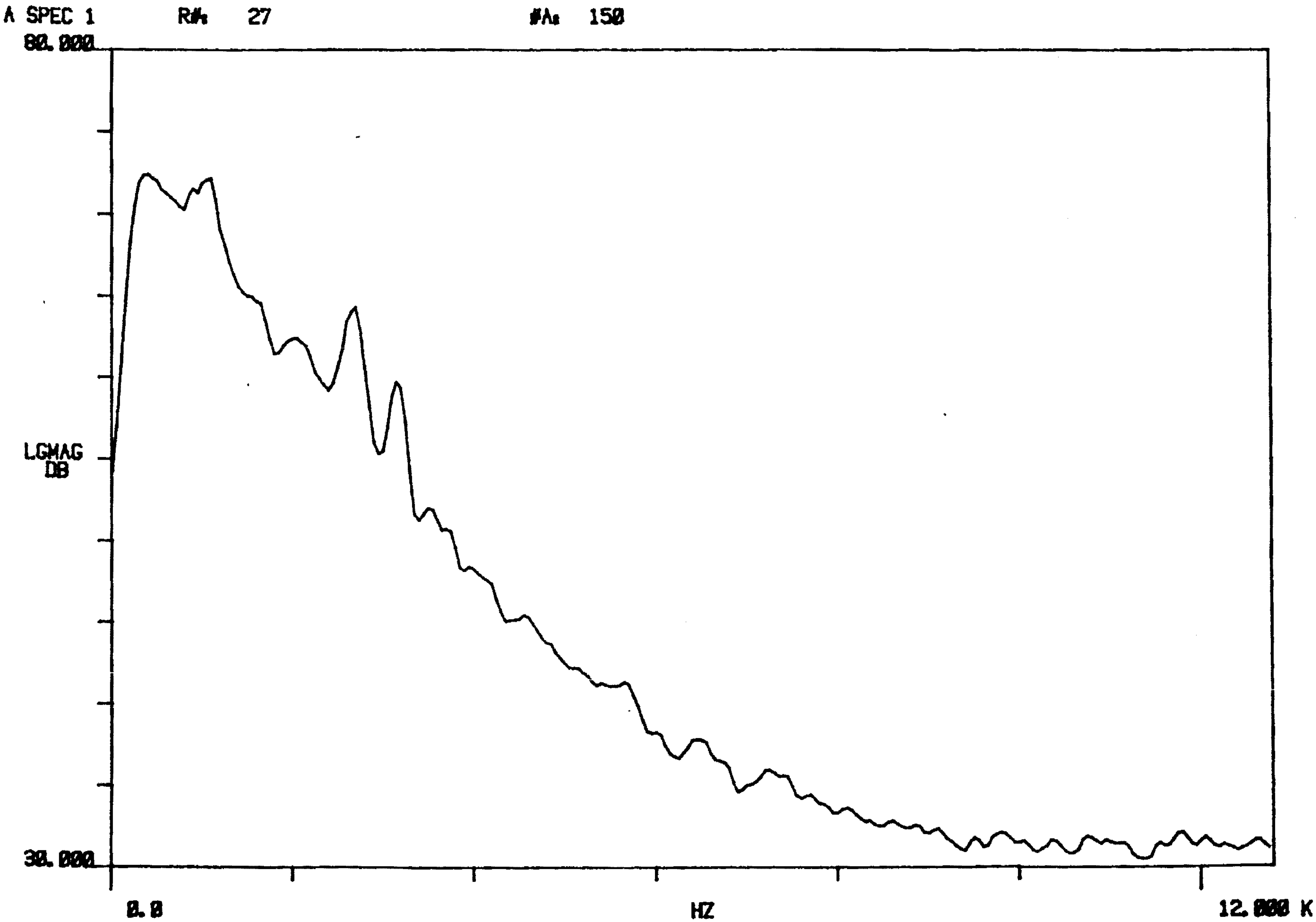


Fig. 15D - A-Weighted Source Contribution Analysis : Pickers
TIP TOP PLANT (pt. 53, ch.2)

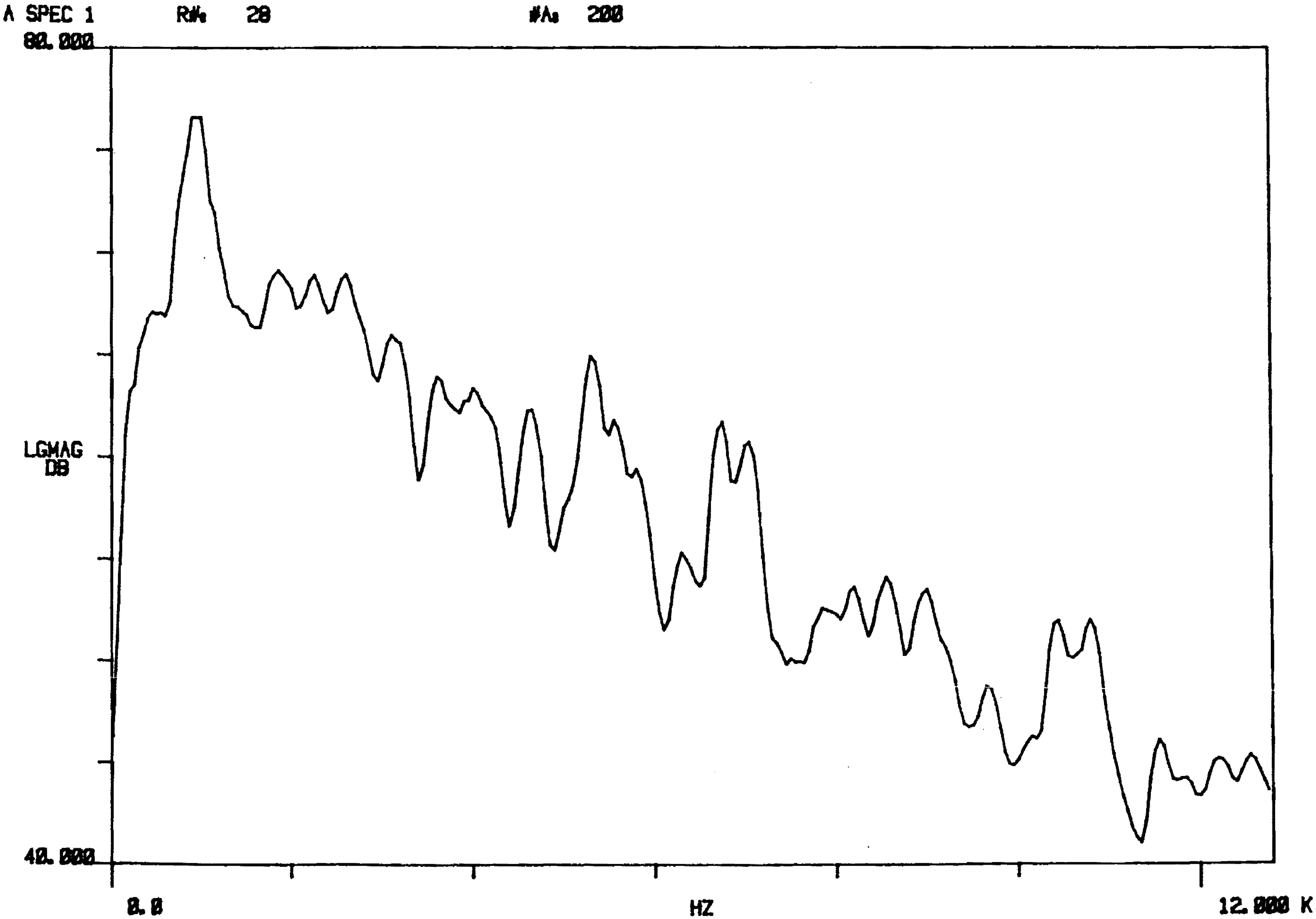


Fig. 16D - A-Weighted Source Contribution Analysis : Gizzard Machine
TIP TOP PLANT (pt. 53, ch.2)

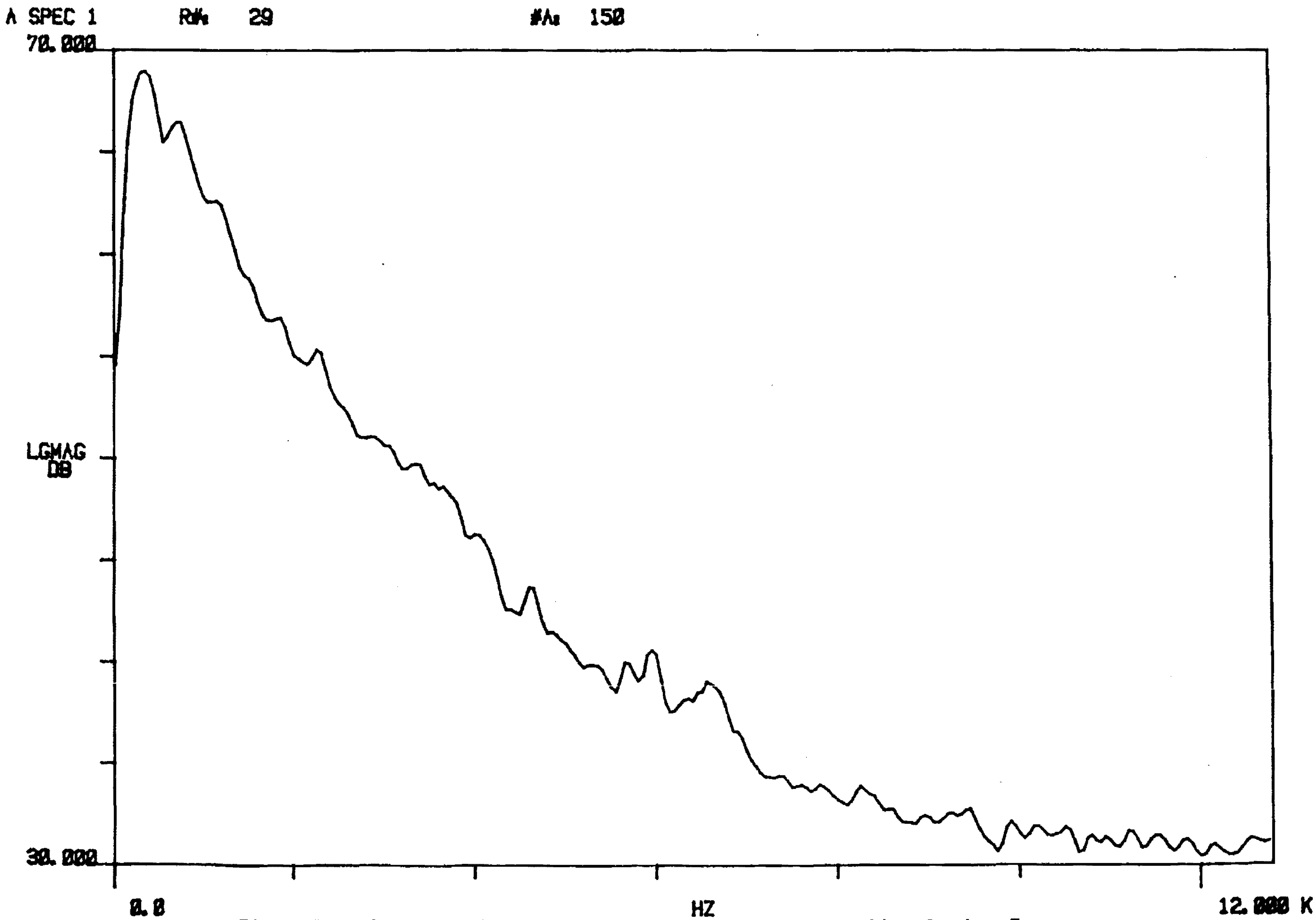


Fig. 17D - A-Weighted Source Contribution Analysis : Circulating Fans
TIP TOP PLANT (pt. 53, ch.2)

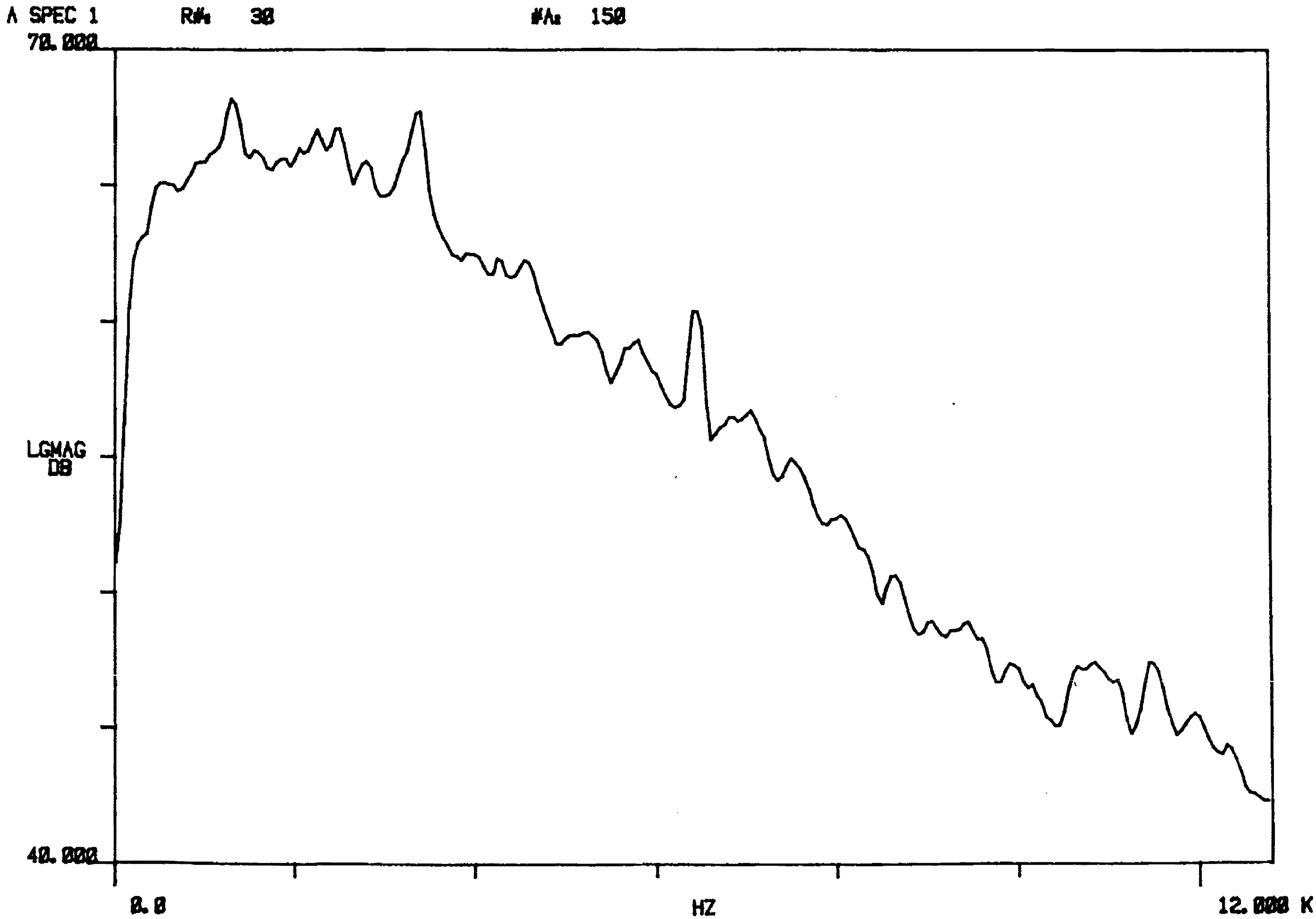


Fig. 18D - A-Weighted Source Contribution Analysis : Shackle Lines
TIP TOP PLANT (pt. 53, ch.2)

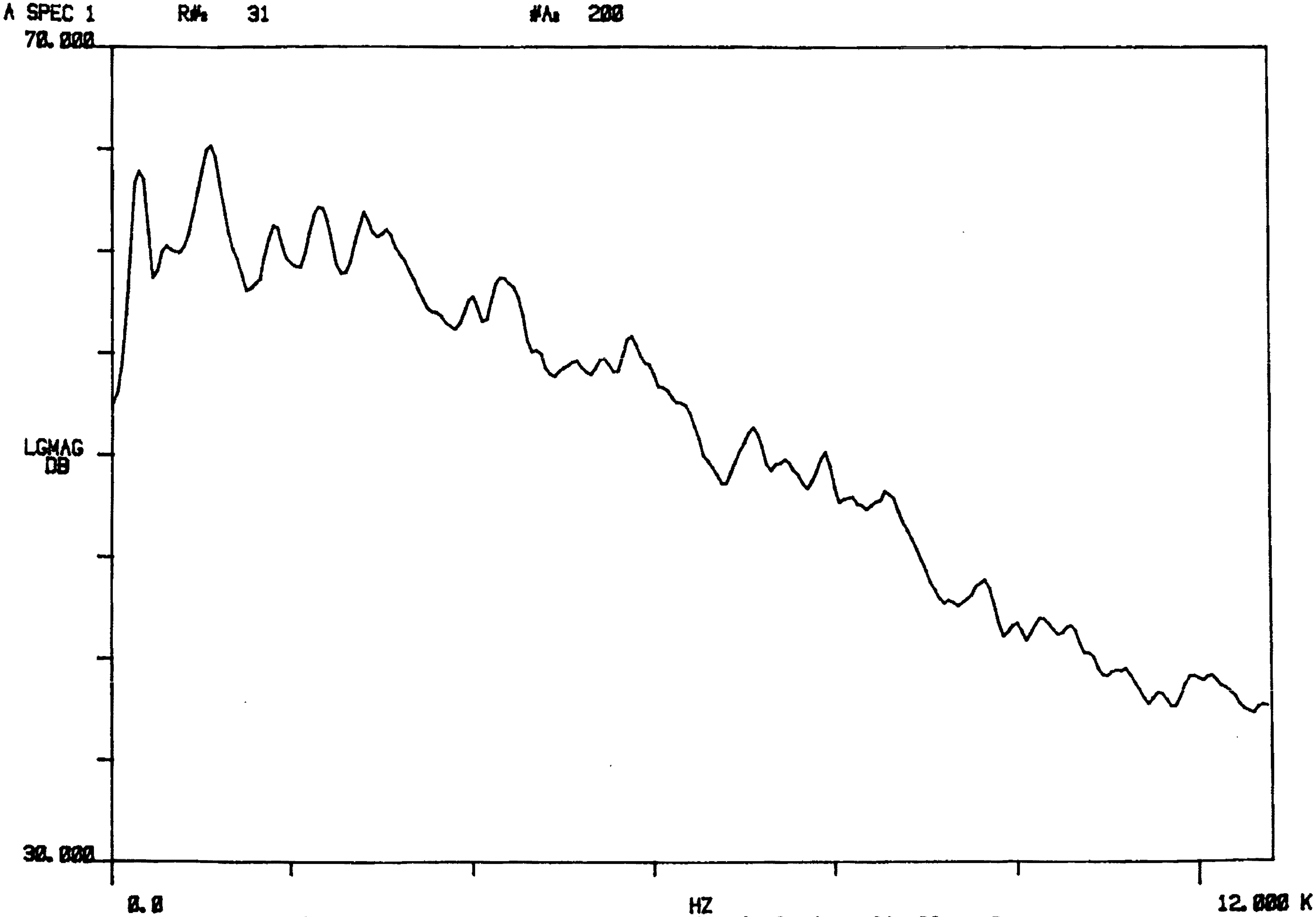


Fig. 19D - A-Weighted Source Contribution Analysis : Air Blast Dryer
TIP TOP PLANT (pt. 53, ch.2)

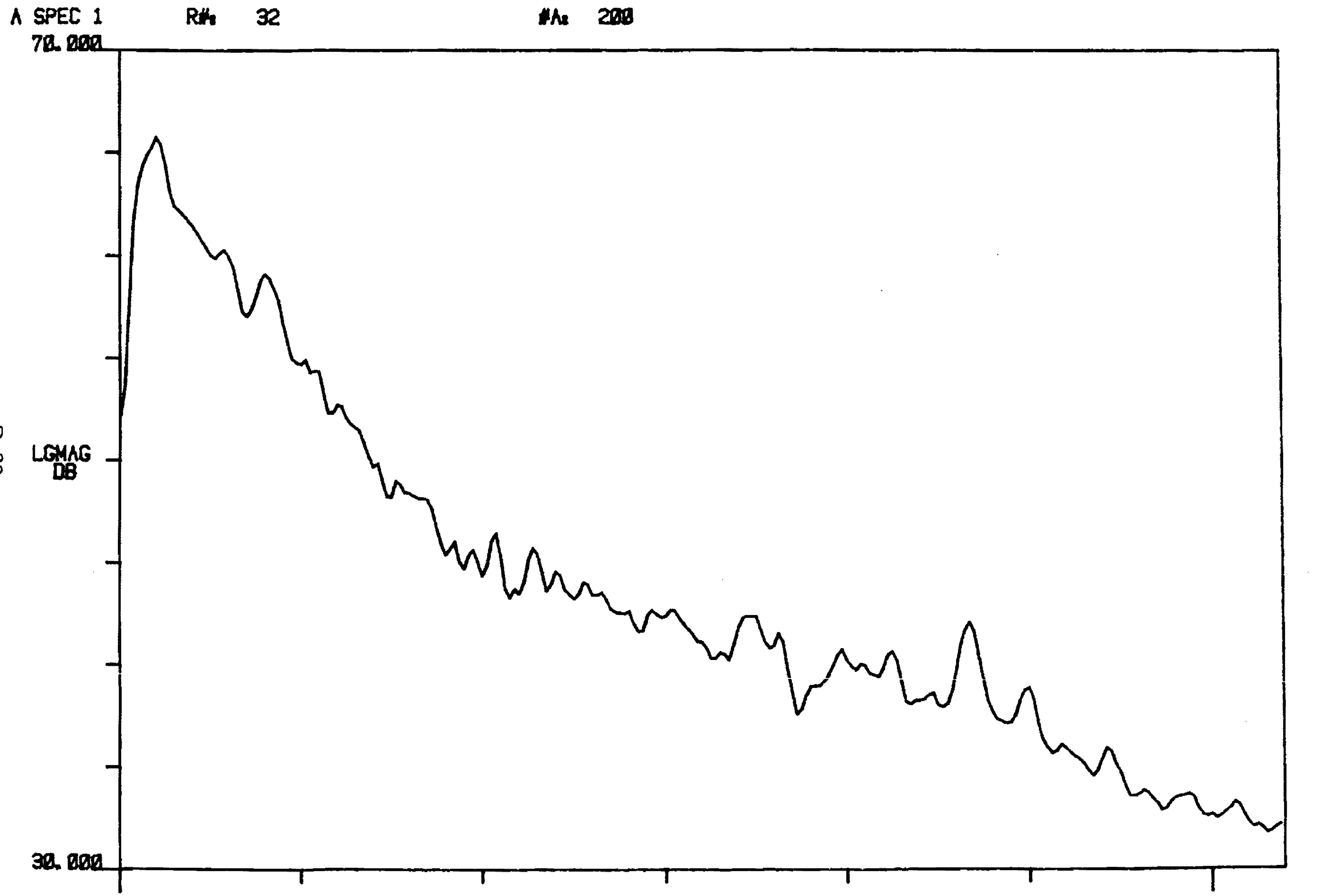


Fig. 20D - A-Weighted Source Contribution Analysis : Shackle Line Foot Remover
TIP TOP PLANT (pt. 53, ch.2)

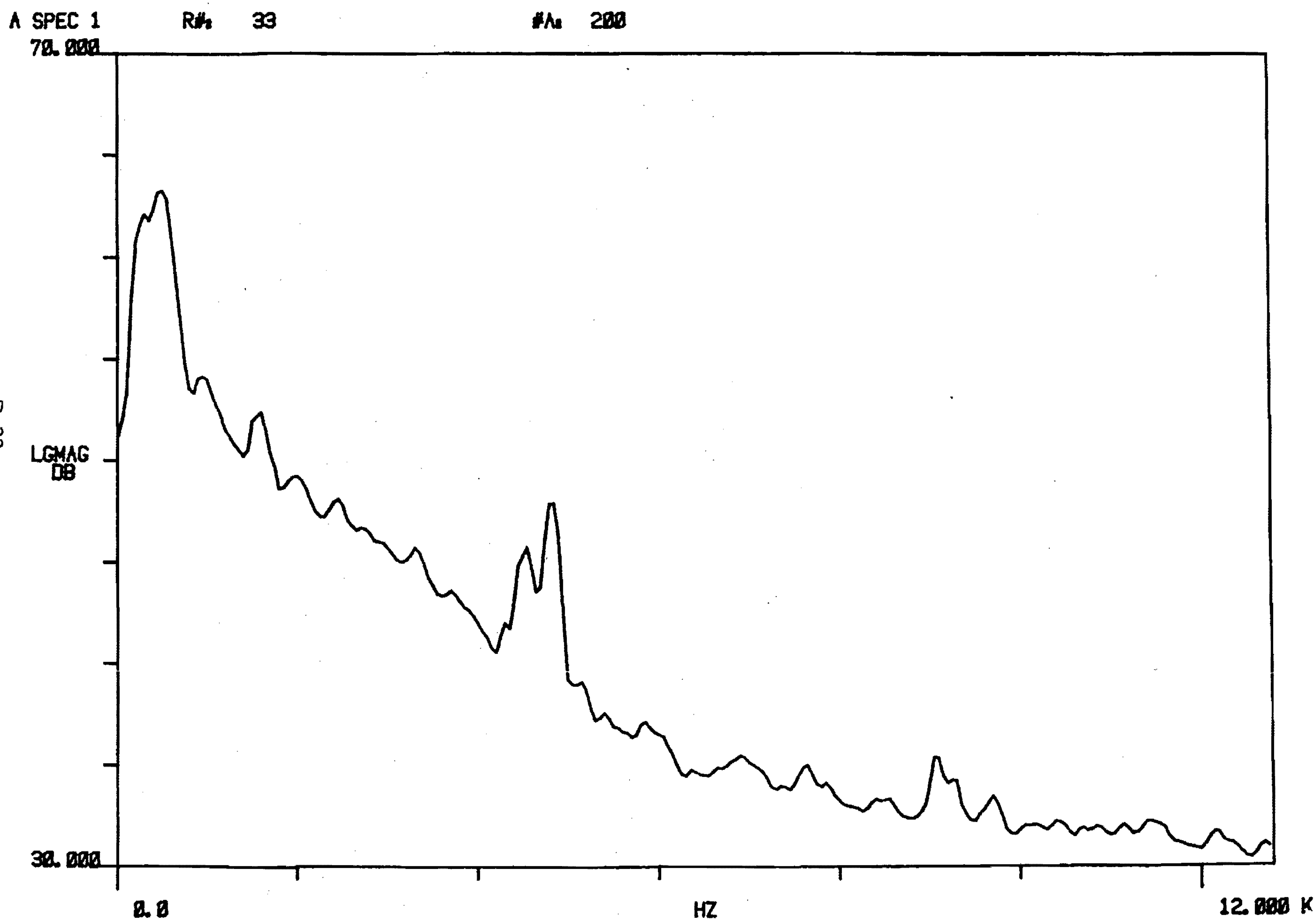


Fig. 21D - A-Weighted Source Contribution Analysis : Hanging Conveyor
TIP TOP PLANT (pt. 53, ch.2)

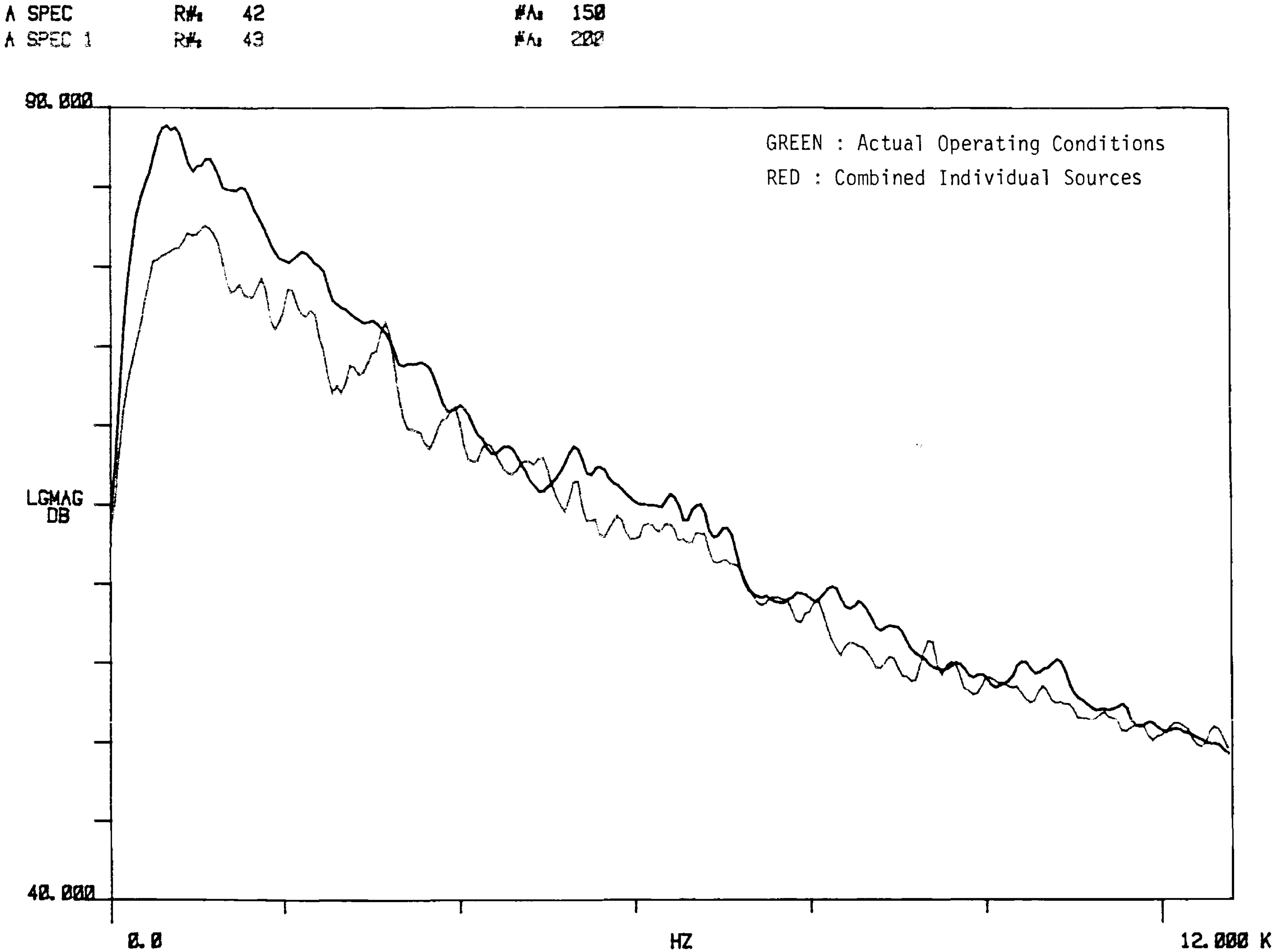


Fig. 22D - A-Weighted Comparison of Combined Individual Sources vs Actual Operating Conditions
TIP TOP PLANT (pt. 53, ch.2)