SIMULATED AND EXPERIMENTAL ATTENUATION OF TRACTOR EXHAUST NOISE

By

GERALD ALBERT STANGL

Bachelor of Science
South Dakota State University
Brookings, South Dakota
1966

Master of Science
South Dakota State University
Brookings, South Dakota
1968

Submitted to the Faculty of the Graduate College
of the Oklahoma State University
in partial fulfillment of the requirements
for the Degree of
DOCTOR OF PHILOSOPHY
July, 1971

SIMULATED AND EXPERIMENTAL ATTENUATION OF

TRACTOR EXHAUST NOISE

Thesis Approved:

Jay G Porterfield
Thesis Adviser

Lawrence O. Roth

R L Lowery

D Durham
Dean of the Graduate College

ACKNOWLEDGMENTS

Appreciation is extended to the following members of my committee: Professor Jay Porterfield, major adviser, for his guidance and allocation of funds for this research project; Dr. Richard Lowery, for his loan of instrumentation and assistance in planning the research; Dr. Lawrence Roth, for his interest and constructive criticism.

A very special thanks to the staff of the Agricultural Engineering Laboratory: Mr. Clyde Skoch, Mr. Norvil Cole, and Mr. Jesse Hoisington, and to Mr. James Gayer, student employee, for assistance in design and construction of test apparatus and in conducting tests.

I would like to thank Professor Paul McCollum and other members of the Electrical Engineering Department for providing analog-to-digital conversion equipment and assistance in data analysis.

Finally, I thank Professor E. W. Schroeder, Head of the Agricultural Engineering Department, for providing facilities and personal financial support; and Mr. Jack Fryrear, for his skill in drafting and photographic arts.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

CHAPTER I

INTRODUCTION

Technology has aided in developing a wide variety of useful and labor-saving devices, but it has also created undesirable side effects such as pollution. A type of pollution prevalent on the agricultural and industrial scenes is noise,[1] particularly noise produced by power sources such as internal combustion engines.

Loud sounds may reduce speech intelligibility and impair hearing, especially after extended exposure. In most cases a sound pressure level of less than 85 decibels (dB) over the octave bands from 300 to 2400 Hertz (Hz) is recommended when the exposure is five or more hours per day (18, 40, 62). Agricultural machines may violate these limits. For example, several researchers have found tractor noise at the operator station to be in excess of these limits (42, 46, 57, 62, 70).

Even with a muffled exhaust, exhaust noise usually proves to be the major source of tractor noise. Perhaps this is because tractor manufacturers are unwilling to accept the losses associated with effective muffling by conventional methods or because operators incorrectly associate noise with power. Usually significant noise reduction at the operator station results from more effective exhaust muffling (41).

Conventional exhaust silencers are typically composed of a combination of expansion chambers and resonators. These elements reflect

[1]Noise is defined as unwanted sound (54).

certain portions of the sound energy back toward the source to cancel some of the oncoming energy (58). Flow resistance creates pressure losses and requires energy to push the exhaust gas through the system. When flow losses in a muffler are low, power losses can decrease as silencing increases (27). Hence, simplicity of design is desirable; and this is practical for engines operating at relatively constant speed and load.

An active silencer,[2] using the principle of acoustic interference, was applied to a tractor engine by Chen (16). It electronically generated noise nearly equal in amplitude and opposite in phase to the noise components that were cancelled. However, active silencers using electronic equipment, such as a microphone, amplifier, and loudspeaker, may be prohibitively expensive. Thus, a mechanical method to cancel noise "actively" appears desirable.

Acoustic theory is often applied in muffler design. This theory linearly relates exhaust system pressure and noise at the exhaust outlet (33). Unfortunately, the assumption of small amplitude plane waves is violated since the actual amplitudes are about 7 psi (22). Therefore, an alternative means of relating pressure and noise may be required to analyze effects of pressure wave modifications on exhaust noise. This relation is of interest because the muffler acts upon the pressure waves in the system to affect the character of noise external to the system.

Simultaneous recording of pressure-time history along the exhaust pipe and of noise at the outlet provided the information to relate pressure and noise. A search of the literature indicated that the

---

[2]Active elements draw upon an auxiliary source of energy.

relationship between these two quantities could be obtained through a transfer function analysis of the exhaust system. This relationship provided the means of evaluating the effect of pressure-time history modifications on the exhaust noise by computer simulation prior to actually constructing the modified exhaust systems. Hence, the most effective silencer modification could be chosen, from those proposed, without experimental evaluation of each one.

Fluctuations in the pressure amplitude were reduced by mechanically superimposing a time-delayed version of the pressure wave upon the original wave. A side-branch resonator produced the cancellation wave in this study. The general type of the system was selected from the results of the simulation analysis.

A total loudness analysis (67) of the exhaust noise was used to evaluate the effectiveness of silencer improvements. This analysis also indicated which noise frequency components were the major contributors to the total exhaust noise. Initial silencing at these frequencies would cause the greatest reduction in total loudness.

The method of analysis developed in this study provides for computer evaluation of potential silencer modifications once the existing system transfer function has been experimentally determined. This simplifies the usual trial-and-error approach to muffler design and may result in development of simple, more effective silencers.

Major exhaust noise frequency components are identified. Reduction of these components by induced changes in the pressure wave has the greatest effect on total loudness. Since tractor noise exceeds hearing conservation criteria, reduction of a major tractor noise source will benefit tractor operators.

Mechanically-induced phase shift provides a low-cost, relatively effective method of silencing by acoustic interference. However, mechanical phase shifters may need further adaptation for application to engines operating at varying speed and load.

CHAPTER II

REVIEW OF LITERATURE

The following areas of literature review are presented to support achievement of the objectives.

Preliminary Concepts

Sound is the alteration in pressure or particle position that is propagated through an elastic medium such as air (41). Sound waves are propagated as longitudinal waves, i.e., particle oscillations (compressions and rarefactions) are along a line parallel to the direction of propagation (6). The strength or amplitude of a sound wave is determined by net unidirectional displacement of the (air) molecules (7). The primary properties of sound and vibration are: (a) intensity or pressure level, and (b) pitch or frequency. Loudness depends upon both of these quantities (15). The chief sources of sounds in air are vibrations in solid objects (54). These vibrations are transmitted by air (usually) to the ear where they cause sensation which may be interpreted as sound. Thus, a receiver must be present to detect sound.

The human ear responds to frequencies ranging from 20 Hz (Hertz, cycles/second) to 15000 Hz, or 20000 Hz for a normal young man. The ear also responds to a wide range of sound intensity. Audible intensities range from 10^{-12} to 10 watts/square meter. The weakest sound pressure detected by an "average" person at a frequency of 1000 Hz is

0.0002 μbar.[1] The largest sound pressure heard without pain is about 1000 μbar, thus the scale of sound pressures covers a range of over $1{:}10^6$. The use of the decibel scale reduces this to a range of 0 to 120 dB (12).

Table I compares absolute sound energy increases with changes in sound pressure level (SPL) and subjective response. This table indicates that reduction of sound transmission by 10 dB requires isolation or removal of 90% of the original sound energy (66).

TABLE I

COMPARISON OF SUBJECTIVE LOUDNESS AND ENERGY CHANGES

Energy Increase	SPL Change, dB	Subjective Loudness Change
Energy doubles	3	Barely perceptible difference
Energy triples	5	Noticeable difference
Energy increases tenfold	10	Loudness seems doubled

Source: (66), page 4.

Sound pressure level is a measure of amplitude. It may be an overall value (linear frequency response) or it may be frequency analyzed. Other measures of noise include subjective loudness ratings such as: total loudness (sones), loudness level (Phons), perceived noise level (noys), the "A" scale weighting of overall SPL (dBA), and the "C" scale weighting (dBC). The total loudness scale, for example, is linear. A sound with twice the sone value of another sound is judged twice as loud by an "average" observer (44, 54, 67).

---

[1] μbar = 10^{-6} bar. 1 bar = 14.5 psia ≃ 1 atmosphere.

Noise must be defined in terms of its amplitude, frequency, and duration to analyze its effects on loss of hearing. Different amplitudes and frequencies cause different effects on hearing loss (49).

Hearing conservation is most important in the speech frequency range of 500 to 2000 Hz (18). Hearing impairment usually occurs without discomfort or pain in the ears. Early loss is at 4000 Hz, then it progresses in both directions (31). Its first indication may be difficulty in hearing speech clearly, or a loss of tone in music (70). This loss is usually not noticed until speech frequencies have losses of at least 15 dB. Thus, substantial losses at 3000 to 6000 Hz may produce no subjective awareness of change in hearing. Years of exposure may elapse before noticeable loss occurs (31). This change in operational efficiency must occur before hearing loss is classed as damage (30). In view of the above facts, several damage-risk criteria proposals have been made, as mentioned in Chapter I.

Many farm machines generate excessive noise and vibration which reduce operator performance and can cause permanent physical damage. The tractor, probably the most frequently used farm machine, subjects its operator to noise and vibration levels that are known to be injurious to health and deleterious to performance (64). Thus, various forms of environmental protection or isolation are required to prevent deterioration in operator performance.

Tractor noise typically exceeds most or all of the hearing conservation criteria, as may be seen in the following section. This excessive noise may cause increased operator stress and permanent hearing loss (57).

Hearing tests of college freshmen at South Dakota State University and the University of South Dakota suggested that one-tenth to one-fifth of the young men from farm homes may have had their hearing damaged by noisy farm machinery. This figure was about four times larger than the national average (47). Hearing acuity of tractor operators in Iowa was worse above 1000 Hz than the acuity of the general population. The operators had even greater losses at 4000 Hz, especially in the 30 to 39 and 40 to 49 age groups. However, tractor noise had not impaired the hearing of all operators tested (46).

Tractor Noise

Sources of Noise

Before an attempt is made to control tractor noise, the factors needing to be controlled must be identified. Knowledge of the various sources of the noise, individual source characteristics, and relative magnitude and importance of each source relative to the total noise is a prerequisite for determining noise control methods (56).

Mechanically-driven fluid systems, such as engines, are inherent acoustic wave generators because of spatial motion of their mechanical parts. In the immediate vicinity of a moving part, periodic compression and expansion of the fluid (gaseous) medium creates an acoustic wave that can propagate and radiate (69). Should the parts be caused to vibrate at their natural frequencies, the noise may be magnified (14). A noise source might not emit significant airborne sound itself, but may rather act as an energy source for vibrations which are transmitted through structural members and converted to airborne sound at a resonant point (51). The transmission of energy through other parts of the tractor will probably alter the frequency spectrum of the sound from that produced by the source because of differences in properties of the transmitting media. Because of these changes, a full-scale investigation of tractor noise must include a separate consideration of all the likely sources and measurement of their noise output (40).

Rowley (56) allocated tractor noise to the following sources: exhaust, intake, fan, and mechanical. He arbitrarily lumped all noise sources except the first three into the mechanical category, i.e., combustion, gear, valve, pump, ignition, cam, bearing, etc. Engine noise, for example, would contribute to each noise source.

<u>Relative Contribution of Noise Sources</u>

A large portion of the noise emitted by a tractor originates at the engine and its accessories (40). Generally, even with mufflers in place, the exhaust is the major source of noise. However, on some tractors it is improbable that noise could be reduced to meet hearing conservation levels by decreasing exhaust noise only (40, 56).

Rowley (56) isolated various sources of noise on a PTO-loaded tractor. These sources in order of importance were:

1. Exhaust - low frequencies from the engine fundamental firing frequency and a small flat peak in the 600 to 2400 Hz range.
2. Fan - noise centered at the blade passage frequency.
3. Mechanical - a frequency range of 600 to 4800 Hz.
4. Intake - (least important) frequency components at the engine fundamental and its harmonics, 75 to 600 Hz.

Priede (55) presented a different hierarchy on noise sources for a diesel engine operated at 1500 rpm. In this case exhaust noise (unsilenced) predominated by about 10 dB over the audio-frequency spectrum, air intake ranked second, and noise emitted by the engine structure (mechanical noise) was the lowest noise category. Fan noise was not mentioned by Priede.

The Major Noise Source

Exhaust noise is a major component of tractor noise even when a standard-equipment exhaust muffler is employed. It is caused when high-velocity gases are rhythmically moved through valve porting by pressure differential across the port (37). These high velocities are attenuated in the manifold and piping, but some energy is transformed into a pressure wave which is propagated as a sound wave superimposed on the much slower gas flow.

Noise from an unsilenced exhaust generally appears as a spectrum containing a series of peaks at the fundamental firing frequency of the engine (exhaust valve opening frequency) and harmonics of this frequency. High frequency components are also present as a result of turbulence and eddies where high gas flow velocities occur. The noise peaks at the fundamental firing frequency result mainly from the combustion process and pulses of exhaust gas emitted by the valves. These pressure pulses have an amplitude of about one-half atmosphere. The majority of the dischange from each cylinder occurs in a time interval of less than one-tenth of each cycle (two revolutions in a four-cycle engine) (22). The pressure pulses are the impulses causing gas flow through the system.

Theoretically any engine design factor which increases gas velocity through the porting or improves coupling between the cylinder and manifold (increases volume flow) will also increase noise (56). For a given engine, increases in speed and load will increase noise output (40).

Results of Previous Tractor Noise Studies

Numerous tractor noise survey reports have been found in the literature and are summarized below. Because a tractor has dimensions larger than the wavelength of many frequencies that it emits, has more than one noise source, and its operator may be in the near sound field,[2] a tractor does not act as an ideal point source. Instead of radiating sound energy equally in all directions, a tractor may radiate up to 41 times more acoustic power[3] in one direction than another. About two-thirds of the total acoustic power emitted by a tractor occurs in the 35 to 140 Hz frequency range (17). Some tractors produce a considerable increase in noise at the operator station when operating without an exhaust muffler, especially at the middle and high audio frequencies. However, this increase in noise does not always result in much difference in the overall noise level.

The octave band spectrum at the operator station of a 40-horsepower diesel tractor under typical loading is shown in Table II (70). The data indicate that low and medium frequency components predominate.

In a 1962 test of 12 tractors (11-diesel and 1-gas) loaded by a belt-driven dynamometer, Hutchings and Vasey (40) measured overall

---

[2]The near field is within three or four diameters of the largest source dimension from the source. Here the SPL varies with position around the source because of nearness to its various elements and superposition of sound from nearby elements. Beyond the near field is the far field where sound energy decreases inversely proportional to the square of the distance from the source (60).

[3]Acoustic power = Intensity x Area of surface sourrounding the source (44).

sound pressure levels. These levels ranged from 100 to 113 dB for tractors developing 27 to 48 belt horsepower at rated speed. They observed no correlation between horsepower and overall SPL.

TABLE II

OCTAVE BAND SPECTRA AT OPERATOR STATION

Exhaust Configuration	Overall Level, dB	Octave Band Level, dB							
		63	125	250	500	1000	2000	4000	8000 Hz
Without muffler	101	87	95	98	89	84	81	78	69
With muffler	92	87	88	84	78	82	76	70	63

Source: (70), page 20.

Related tests in 1957 by Lierle and Reger (46) and in 1965 by Jensen (41), Jones and Oser (42), and Simpson and Deshayes (62) showed the trend in tractor noise over time. The 1957 tests on 11 tractors (2 rear, horizontal exhausts; 6-diesel and 5-gas) measured octave band levels at the operator station while the tractors pulled a variety of implements. The 300 to 600 and 600 to 1200 Hz bands had average levels of 95.0 and 90.5 dB, respectively (46). The total loudness of these tractors varied from 85 to 190 sones with a mean of 175 sones, as calculated by Jensen (41) for comparison with the total loudness of 1965-model tractors (21 tractors from 7 manufacturers) operated at 75% drawbar load and 4½ mph. The average total loudness of the two tractor groups was almost identical even though the average horsepower was nearly two times larger for 1965 models. Another test of 1965 tractors (42) surveyed the overall and octave band SPL at the operator's left ear while he was sitting and standing. The overall levels are presented in Table III. The octave band analysis showed that levels in the speech range (250 to 3000 Hz) exceeded 95 dB[4] in all tests (62). A summary of the octave analysis at 75% load is shown in Table IV.

Tables III and IV show that agricultural tractors produce noise that may result in permanent operator hearing loss for extended exposure. The mean SPL over the 500 to 2000 Hz bands for each fuel type were: diesel--90.1 dB, gas--87.8 dB, and LP gas--85.6 dB.

The study of 1965 tractors showed no direct relationship between SPL and horsepower. However, doubling the load (50 to 100% of rated load[5]) resulted in nearly a two-fold increase in sound energy output, evident by a 3-dB increase in SPL for both the overall and octave band levels.

Mainly because of the engine and exhaust location with respect to the operator, 62% of the tractors in Table III had a higher SPL and 27% had a lower SPL for a standing operator (42). Passing the exhaust gases 12 feet in front of the tractor reduced noise at the operator station by 2 to 8 dB (40). Further inquiry into the effect of exhaust outlet location (vertical stack) with respect to the operator by Schmer (59) showed noise levels to be highest with the stack closest to the operator and lowest at the farthest position for a given stack height. Exhaust outlet relocation mainly affected lower frequencies, i.e., 31.5 to 500 Hz. The low frequency loudness indicies (from total loudness

---

[4]This value appears to be in error since other data given in Simpson and Deshayes (62) indicate that 85 dB would be a more appropriate value.

[5]Rated load is sufficient load to cause the engine to run at the manufacturer's rated speed (59).

TABLE III

OVERALL NOISE LEVELS OF VARIOUS TRACTORS

Tractor	Overall Level, dB					
	Full Load		75% Load		50% Load	
	Sitting	Standing	Sitting	Standing	Sitting	Standing
Oliver 1600 - Gasoline	105.5	106.5	106.5	106.0	105.5	105.0
David Brown 990 - Diesel	102.0	104.5	100.0	102.5	99.5	99.5
Allis-Chalmers 190 XT - Diesel	103.0	101.5	102.0	100.5	99.5	98.5
Ford Commander 6000 - Diesel	109.0	107.0	103.5	102.5	101.0	99.0
Massey-Ferguson MF 150 - Diesel	106.5	109.0	105.5	107.5	106.5	104.5
Nuffield 10/42 - Diesel	102.0	104.0	101.5	103.5	101.0	102.0
Ford 3000 8-Speed - Gasoline	106.0	108.5	105.5	108.0	104.0	106.0
Ford 3000 8-Speed - Diesel	104.5	107.0	104.5	106.0	104.5	105.5
Allis Chalmers XT 190 - Gasoline	101.5	100.0	100.5	98.5	99.0	98.0
Ford 5000 Select-O-Speed - Diesel	102.0	104.0	100.5	101.0	99.5	99.0
Ford 5000 - Diesel	102.5	104.0	101.0	102.5	98.0	99.5
Massey-Ferguson 175 - Diesel	102.0	102.5	100.0	100.5	98.0	98.5
Ford 4000 Select-O-Speed - Diesel	107.0	108.5	107.0	108.0	106.0	107.0
Ford 4000 8-Speed - Diesel	107.0	108.5	106.0	107.5	105.5	106.0
Massey-Ferguson MF 180 - Diesel	102.5	105.5	99.5	102.5	99.0	101.0
John Deere Model B - Gasoline	97.0	98.0	96.5	97.5	96.5	96.5
Ford 3000 4-Speed - Gasoline	106.5	108.5	104.5	107.0	103.0	105.0
Ford 4000 8-Speed - Gasoline	107.5	109.5	106.5	108.0	105.5	106.0
Ford 4000 Select-O-Speed - Gasoline	107.5	109.5	107.5	109.0	106.0	106.5
Farmall 1206 Turbo Charged - Diesel	108.5	107.0	105.5	106.0	102.0	104.5
Farmall 656 - Gasoline	113.0	114.0	112.5	112.5	110.0	110.0
Nuffield 10/60 - Diesel	99.0	99.5	99.0	99.5	98.5	98.5
Ford 3000 Select-O-Speed - Diesel	105.0	107.0	104.5	106.5	104.0	105.5
Ford 2000 4-Speed - Gasoline	103.5	106.0	103.0	105.5	101.5	102.5
Ford 2000 8-Speed - Gasoline	103.5	106.0	103.0	105.5	101.5	103.0
Case 930 - Gasoline	111.0	110.0	109.0	109.0	107.0	107.0
Massey-Ferguson 135 - Gasoline	104.0	100.5	100.5	97.5	99.5	97.5
David Brown 880 - Diesel	108.0	109.5	106.0	107.0	104.5	105.0
Ford 6000 Select-O-Speed - Gasoline	107.0	106.0	104.0	102.0	100.5	98.0
John Deere 2510 - Diesel (Synchro-Range)	101.0	101.0	100.0	99.0	98.0	97.5
John Deere 2510 - Diesel	103.0	102.5	101.0	101.0	Tractor Broken	
Massey-Ferguson 165 - Gasoline	104.5	107.0	104.0	105.5	103.0	105.0
Ford 3000 Select-O-Speed - Gasoline	105.5	108.5	105.0	108.5	104.0	106.0
John Deere 60 - Gasoline	98.5	100.0	97.0	99.0	96.0	98.0
Farmall 656 - Diesel	112.0	112.5	109.0	108.0	105.5	105.0
Massey-Ferguson MF 165 - Diesel	105.0	108.5	102.0	106.0	100.5	105.0
Massey-Ferguson 135 - Diesel	106.0	102.5	105.0	101.5	104.0	101.0
Ford 3000 4-Speed - Diesel	105.0	108.0	105.0	106.5	104.5	105.5
Kubota - Gasoline	100.0		99.5		96.0	
Case 941 - Liquid Petroleum	109.5	108.5	108.0	107.0	106.5	105.5
Case 841 - Liquid Petroleum	107.5	109.0	107.0	108.0	105.0	106.0
Case 831 - Diesel	104.5	107.0	101.5	102.5	98.5	99.0
International 424 - Diesel	106.0	106.5	104.5	104.0	102.0	102.0
Case 931 - Diesel	108.0	106.5	106.5	104.5	104.5	103.0
Allis-Chalmers 190 - Diesel	100.0	97.5	99.0	97.0	98.5	98.5
Minneapolis-Moline M670 - Liquid Petroleum	104.0	105.0	105.0	105.0	104.0	103.5
Minneapolis-Moline M670 - Diesel	105.0	105.0	103.5	104.0	103.0	103.0
Minneapolis-Moline - Gasoline	105.5	106.5	105.0	106.0	105.0	106.0
International 424 - Gasoline	104.5	109.0	104.0	108.0	103.5	107.0
Allis-Chalmers 190 - Gasoline	102.0	100.0	101.0	100.0	100.0	99.0
Case 841 - Gasoline	107.5	110.0	109.0	110.0	107.0	108.0
International 4100 - Diesel	103.5	101.0	101.0	101.0	101.0	101.0
Massey-Ferguson 1100 - Diesel	100.5	100.0	98.5	99.0	97.5	97.5
Allis-Chalmers 190ST - Liquid Petroleum	101.0	100.0	96.0	94.0	90.0	91.0
John Deere 4020 - Diesel	111.0	111.0	108.0	107.5	106.0	104.5

Source: (62), page 348.

analyses) decreased rapidly as the outlet was moved away from the operator. However, noise decreased at a diminishing rate as the outlet was moved farther away. Horizontal and vertical movement had essentially the same effect. After exhaust relocation for noise reduction, the 1000 to 2000 Hz band became the peak level of the measured spectrum; whereas, the peak level occurred in the 125 to 250 Hz band when the stack was at its original location.

TABLE IV

OCTAVE BAND SPECTRA AT 75% LOAD

Source	Octave Band Level, dB									
	31.5	63	125	250	500	1000	2000	4000	8000	16000 Hz
Mean of all tractors	80	98	100	94	91	90	86	84	80	72
Minimum of all tractors	72	78	90	85	82	78	74	65	61	50
Maximum of all tractors	96	107	108	100	102	97	98	95	90	80

Source: (42), page 147.

Ryland and Turnquist (57) reduced noise at the operator station from 92.6 sones for a tractor under 75% PTO load at 1900 rpm without cab to 76.5 sones with the addition of a metal cab insulated with 1-inch fiberglass sound absorbing material. However, the standard cab (without insulation) had a detrimental effect, as its total loudness was 150.0 sones. This was twice as loud as the insulated cab. Similar results were obtained in an agricultural noise survey by Matthews (50), see Table V. United States Steel (66) also developed a cab designed for noise control. Under full tractor load this cab reduced the total loudness from 113 sones (without cab) to 43 sones.

TABLE V

NOISE LEVELS WITH AND WITHOUT CABS

Test Condition*	dBA			Total Loudness, sones		
	Min.	Mean	Max.	Min.	Mean	Max.
Tractor with cab	82	95	108	41	160.5	280
Tractor without cab	74	88	102	28	98.5	169

*Tractors were both wheel and crawler types, and were performing a variety of tasks. Cabs were both metal and canvas.

Source: (50), page 166.

Control of Exhuast Noise

General Considerations

Usually significant noise reduction at the operator station can be accomplished by more effective exhaust muffling at the expense of increases in: (a) muffler cost, (b) power loss from back pressure, (c) muffler size, or (d) audibility of other noises (41). A successful silencer design must reduce overall exhaust noise to acceptable limits of subjective loudness and have minimum effect on engine performance. Both of these requirements have yet to be adequately defined. Also, theory does not account for noise generation within the silencer or sound radiation from vibration resonances of the exhaust system (22).

In general, cross-sectional area of the muffler determines the amount of possible noise attenuation; whereas, internal design and length determine the frequencies that are attenuated. The silencer design depends upon noise-source characteristics of the engine, muffler location in the exhaust system, and required amount of reduction in total loudness. Muffler location and type plus engine speed range determine whether resonances occur in the system (56). Except in short exhaust pipes, one of the most difficult problems is silencing sound from pipe resonances excited by engine firing frequencies (61).

The resonant frequencies of a straight pipe with one end closed are odd multiples of four times the pipe length:

$$f_n = \frac{(2n - 1)c}{4\ L} \qquad (2\text{-}1)$$

where: f_n = n-th resonant frequency, Hz.

c = speed of sound, ft/sec.

L = pipe length, ft.

According to conventional accoustic theory, the pipe responds as if closed on one end because the engine acts as a very high impedance sound source (68).

When the exhaust noise approaches the same level as extraneous tractor noise (other than from the exhaust), both become equally important in determining the tractor noise spectrum. Then exhaust noise must be reduced in greater increments to reduce the overall noise level by equal amounts. Thus, the noise reduction gained by a given muffler is dependent upon the relative intensities of the extraneous and exhaust noises (26).

Exhaust system effects on engine performance are back pressure and increased valve temperature; both have the same causes. They are the result of changes in exhaust gas velocity and flow losses (friction) that occur as the gases flow through the system. As back pressure increases, more engine power is required to push the gases through the system. A very small decrease in sound level is associated with increased back pressure. Therefore, it does not cause silencing; and high back pressure, in itself, does not result in good muffling (28). Because of complicated pressure and flow patterns at the engine end of the system, a reduction in average pressure at the manifold will not necessarily reduce power loss. The timing of the reduction is important. Thus, removal of a muffler may decrease power because flow characteristics of the system were altered; even though average pressure may decrease, pressure at the valve port might increase near the end of the exhaust stroke. This pressure increase causes the cylinder to retain more combustion residuals to dilute the fresh intake charge, and the result is a power loss (63).

Conventional and Active Silencers

Tractor exhaust mufflers are generally a reactive or reactive-absorptive type. A reactive muffler, usually composed of expansion chambers and resonators, attenuates a portion of the noise spectrum by reflecting noise out-of-phase back toward the engine. If in-phase, amplification occurs. Absorptive materials convert sound energy into heat by damping vibration and creating turbulence in the gas flow. This is effective at higher frequencies, but the adverse environment limits the longevity of such packing materials (56, 58). Conventional mufflers reduce high frequency noise considerably, but are not very effective at low frequencies (39). Low frequency noise is much more costly to attenuate than high frequency noise because larger silencer size is required (7).

An active means of attenuating noise by acoustic interference, developed by Olson (52), was applied to a tractor engine by Chen (16). This silencer electronically generated a noise which was nearly equal in amplitude and opposite in phase to the noise components to be cancelled. The combination of these two noises resulted in destructive interference if their path difference and time delay was equivalent to one-half the wavelength of each frequency (63).

Active silencing has been considered as: (a) the superposition of the cancellation and original wave forms, or (b) the creation of an acoustic impedance shunt for the original acoustic resistance, i.e., absorption of sound rather than cancellation (19, 52). Superposition is the most readily applied method to determine the effect of amplitude and phase variation (from that of ideal cancellation) on the net sound produced. At a given frequency the resultant pressure is the vector sum of the original and cancellation pressures (10). This assumes that the two waves are independent (19).

In order to obtain a 20-dB reduction at one frequency, the cancellation wave must have an absolute magnitude of at least 90% of the original frequency component and only a very small difference in phase shift from 180° (34). A 10° variation from 180° phase shift between signals of equal amplitudes will result in only a 15-dB attenuation (10). Although Chen did not determine or control the phase relation, he observed about a 3-dB attenuation in the overall noise level at the operator station by a trial-and-error positioning of the cancellation noise source. Most of this occurred in the 63 and 125 Hz bands; none was observed above 500 Hz (16).

Basic Silencer Theories and Their Limitations

Several theories are presented in the literature to mathematically describe flow and pressure in exhaust systems. These may be generally classified as: (a) acoustic filter theory, (b) shock tube theory, (c) finite-wave theory, (d) method of characteristics, or (e) direct finite-difference solution of the system differential equations. The latter category is difficult for even the most simple systems and will not be considered here. Cundiff (21) describes the first three theories while the method of characteristics is described by Benson, et al (4) and others (9, 32, 71). Some limitations of these theories are presented below.

Accoustic Filter Theory: This theory assumes:

1. Sound pressures are small (less than 110 dB) compared to average absolute gas pressure so that nonlinear effects are negligible.
2. No reflected waves exist in the tail pipe of the muffler.
3. The sound pressure waves are plane waves (one-dimensional).
4. Viscosity and temperature (heat conduction) effects on sound propagation are negligible.
5. The physical system dimensions are small compared to the wavelengths of interest (7, 21).
6. The gases are not flowing through the system (22).

These assumptions allow the system to be described by linear differential equations. In other words, this theory says that the intensity of sound transmitted through a pipe is directly proportional to the intensity at the pipe inlet for all frequencies (33). Ease of applying this theory is counteracted by frequent violation of its assumptions. For example, pressure pulses in an exhaust system have an amplitude of about one-half atmosphere, or 185 db, and the gas travels at an average velocity of 200 to 300 ft/sec (22, 63).

Shock Tube Theory: Also known as single pulse theory, its assumptions relative to an exhaust system are:

1. Disturbances from one exhaust pulse have decayed almost completely before the next one occurs (pulses are separated by about 18 feet in a 4-cylinder, 4-cycle engine operated at 3000 rpm).
2. Energy is dissipated only by: entropy rise in the gas, interaction of shock waves traveling in different directions, and interaction of reflected shock waves and steady flow through an area change.
3. Flow of gases is unsteady in pipes of constant cross-section, and is steady at pipe area changes.
4. The unsteady flow can be described by one-dimensional flow in a shock tube (a shock front develops in the exhaust pipe).

The first assumption allows examination of methods to reduce the amplitude of one pressure pulse as a means to attenuate noise at the exhaust outlet (22). A shock front will develop under certain conditions in an exhaust system since a compression wave tends to steepen to a shock front (rarefactions tend to flatten out). For example, a pressure pulse with a uniform increase of 7 psi over 1-millisecond (msec) will form a shock front after traveling for 2 msec at atmospheric pressure and temperature. This will occur more rapidly at higher temperatures and pressures. Except for a short pipe the unsteady flow behavior is thus similar to that in a shock tube (25). However, dissipation due to friction and interference of multiple reflections are not considered since the theory assumes the pulses are discharged into an empty tube (9).

Finite Wave Theory: Finite wave theory is an extension of acoustic filter theory to include the nonlinearity of finite amplitude waves (71). A propagating wave of finite amplitude experiences changes in waveform shape because each point on the wave profile has a different velocity than its neighboring points. As the distance from the source increases, waves steepen on the leading face of a compression wave and on the rear face of a rarefaction wave (72). Weak shock fronts may develop for compression waves in the exhaust system, as mentioned previously.

The gas flow is assumed to be adiabatic, isentropic, one-dimensional, frictionless, and to satisfy perfect gas laws. At area changes, the system differential equations (continuity, momentum, and energy) are simplified by neglecting partial derivatives with respect to time since they are much smaller than those with respect to distance (21). The isentropic assumption neglects effects of heat transfer and wall friction in the exhaust pipe. These effects both attenuate the pressure wave and cause an entropy gradient along the pipe. Gas temperature is primarily affected by heat transfer; the temperature decrease causes a reduction of the speed of sound as the gas moves toward the pipe outlet. Friction has the opposite effect (72, 73). These waveform changes are additional to those of finite amplitude wave propagation without friction and heat transfer. They are sometimes incorporated into theoretical analyses by assuming that, at any instant and position, the friction and heat transfer rate are those of steady flow at the same Reynolds number. The differential euqations of the wave motion are then best solved by the method of characteristics (1). Temperature discontinuities between successive cylinder discharges or across shock fronts are also not considered in this theory. The discontinuity may cause a partial closed-end reflection of the pressure waves (73).

Method of Characteristics: As the name indicates, this is not really a theory, but a method of solving the system equations graphically. The equations must be first order, nonlinear differential equations. Hence, the effects of heat conduction, diffusion, and viscous shearing within the fluid must be neglected. Other assumptions are similar to finite wave theory with the exception of replacing the isentropic assumption with the Reynolds' analogy relating heat transfer and coefficient of friction (73). This method is the only practical procedure for simulating the complete pressure-time history of exhaust pulses (including reflections). However, it is rather costly in computer time, difficult to apply, and still makes simplifying assumptions about flow conditions at area changes (24). Pressure-time histories can only be predicted at some distance downstream from area changes because of the complex changes in flow patterns at such locations. This distance must be evaluated experimentally as well (5).

Each of the above theories requires experimental evaluation of several coefficients to obtain solutions for a given exhaust system. Simplifying assumptions limit their application and accuracy, but difficulty of use remains. Perhaps actual measurement of the pressure-time history in an exhaust system would be a more realistic approach to evaluate the effect of pressure wave modifications on noise produced at the exhaust outlet.

Transfer Function Analysis

In order to relate exhaust system pressure to noise at the exhaust outlet, the effect of the system on the pressure wave must be determined. If the pressure-time history at a given point is considered the input to the remaining downstream system and the noise history is considered the system output, the relation between them in the frequency domain is known as the transfer function of the system. Since the pressure wave is periodic (repeating at the engine firing frequency), the frequency response (rather than the transient response) of the system will be determined. This response is linear only if pressure and noise frequency spectrums are directly related. The system dynamic characteristics can be determined without knowing the system details required by the theories mentioned previously. Only the system input and output must be known to experimentally evaluate the transfer function.

First the input and output time functions, $f(t)$ and $g(t)$, must be converted from the time domain to the frequency domain by use of the Fourier integral. The time functions are assumed to: (a) have a finite number of maxima, minima, and finite discontinuities (Dirichlet conditions), (b) have a finite area under the curve, i.e., $\int_{-\infty}^{\infty} |f(t)|dt$ is finite, (c) be real, and (d) be casual, i.e., zero for time less than zero. Then the Fourier integral transform converts the time function $f(t)$ into a complex frequency function $F(\omega)$:

$$F(\omega) = \int_{-\infty}^{\infty} f(t)e^{-i\omega t}dt$$

$$= \int_{0}^{\infty} f(t)e^{-i\omega t}dt \qquad \text{since } f(t) = 0 \text{ for } t < 0. \quad \text{But:}$$

$$e^{-i\omega t} = \cos(\omega t) - i\,\sin(\omega t). \qquad \text{Therefore:}$$

$$F(\omega) = \int_{0}^{\infty} f(t)\cos(\omega t)dt - i\int_{0}^{\infty} f(t)\sin(\omega t)dt$$

$$= R(\omega) + i\,X(\omega)$$

$$= A(\omega)e^{i\phi(\omega)} \tag{2-2}$$

where: $R(\omega)$ = real frequency component.

$X(\omega)$ = imaginary frequency component.

$A(\omega) = \sqrt{R(\omega)^2 + X(\omega)^2}$ = amplitude spectrum.

$\phi(\omega) = \tan^{-1}\dfrac{X(\omega)}{R(\omega)}$ = phase spectrum.

$i = \sqrt{-1}$.

The time function $f(t)$ has now been converted to a complex frequency function having an amplitude and phase spectrum. Repeating this process for $g(t)$ yields the result: $G(\omega) = B(\omega)e^{i\theta(\omega)}$.

The transfer function $H(\omega)$ for a linear[6] system with constant parameters[7] is the Fourier transform of the unit impulse function $h(t)$ of the system (3, 38, 53):

---

[6]A system is linear if its output to a sum of inputs equals the sum of its outputs produced by each input individually, and if its output produced by a constant times the input equals to the constant times the output produced by the input alone (53).

[7]Constant coefficients, only, must appear in the system differential equation. If the system response to $f(t)$ is $g(t)$, then $g(t-t_1)$ is the response to $f(t-t_1)$ (3).

$$H(\omega) = \frac{G(\omega)}{F(\omega)} = \frac{B(\omega)e^{i\theta(\omega)}}{A(\omega)e^{i\phi(\omega)}}$$

$$= \frac{B(\omega)}{A(\omega)} e^{i(\theta(\omega)-\phi(\omega))} \qquad (2\text{-}3)$$

Thus, the system transfer function can be evaluated directly from knowledge of the amplitude and phase spectrums of the input and output. By taking the inverse Fourier transform, the unit impulse function can be determined for a linear system. However, the inverse transform is not valid for a nonlinear system because the linearity assumption has been violated. This is of no consequence, though, since the transfer function and not the unit impulse function is desired to relate pressure and noise in an exhaust system.

CHAPTER III

DEVELOPMENT OF PROBLEM

Objectives

Two areas that could be studied were indicated by the background information of Chapter I. The following objectives were selected from these areas based on the review of literature and available instrumentation.

1. Determine the relationship between exhaust system pressure and noise at the exhaust outlet for specific systems and thereby determine the portions of the exhaust system pressure-time history which are the main contributors to noise at the exhaust outlet.
2. Using the above knowledge, apply the principle of mechanically-induced acoustic interference to reduce the noise attributed to the pressure-time history.

System Configuration and Variables

The exhaust system used in this study was that of an International Harvester 1948 Model "H" tractor. This unit was selected because of its availability and ease in loading with a PTO dynamometer. On a particular day, the engine, with a standard exhaust system, operated at its rated speed of 1650 rpm when at full throttle and subjected to a 280-inch-pound PTO torque. The torque load was held at this value for all subsequent tests. However, the engine speed varied slightly from test to test because of modifications in the exhaust system and atmospheric changes. Rated speed under load was the operating condition chosen since noise increases with speed and load (40).

Variables measured were noise and pressure. Noise measurements were made at the exhaust outlet to minimize the influence of other noise sources on the readings. A preliminary comparison of exhaust noise from the standard system and from the system with a straight pipe replacing the muffler indicated that noise frequencies below 400 Hz were essentially unaffected by the muffler, see Figure 1. These frequencies were major contributors to the total exhaust noise. Thus, in this study, the analysis of noise and pressure was mainly concerned with frequencies below 600 Hz. The dashed vertical lines in Figure 1 indicate changes in the frequency range of the Type 2107 frequency analyzer.

Exhaust system dynamic pressure was sensed at the inside wall of the piping to avoid affecting the system characteristics. The pressure taps were located at least two pipe diameters from changes in pipe or muffler cross-section, except at the outlet end (where they were 2 inches from the outlet). These locations reduced the effect of area changes on the pressure observed.

Pressure and noise were measured for several exhaust system configurations: standard, no muffler, straight pipe replacing the muffler, single-cylinder straight pipe, and standard system with wave cancellation. The transfer function of each system, to relate pressure and noise, could then be determined, if desired. In addition, the single-cylinder exhaust indicated the effect of pipe length on system pressure, see Appendix B.

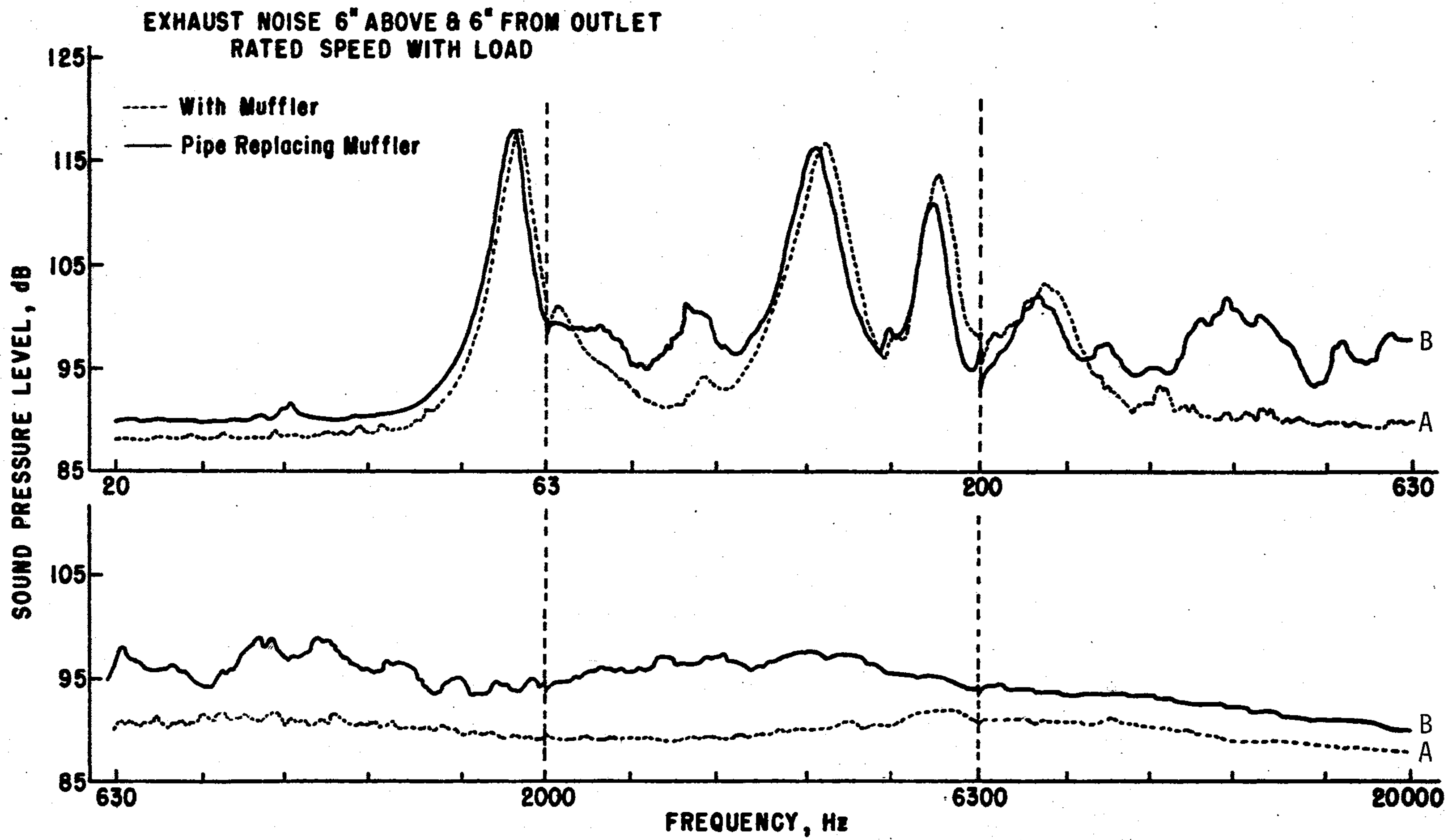

Figure 1. Narrow Band Frequency Analysis of Exhaust Noise.
A - Standard System (Test No. 1);
B - Pipe Replacing Muffler (Test No. 3)

Temperature-time history effects on the pressure were not considered in this study.

Theory of Wave Cancellation

Noise reduction by acoustic interference essentially requires the superposition of a pressure wave upon the original wave in the exhaust system. As noted in the review of literature the superimposed wave should be nearly equal in amplitude and 180° out of phase with each frequency component to be cancelled in the original wave.

Consider one frequency component to be cancelled by a wave containing that component delayed 180° in phase:

$$P_o(t) = A \sin(2\pi ft)$$

$$P(t) = (A + e)\sin(2\pi ft - \pi) = -(A + e)\sin(2\pi ft)$$

where: $P_o(t)$ = original wave of frequency f as a function of time t.

$P(t)$ = cancellation wave.

A = wave amplitude, which could be a function of time.

e = amplitude difference between the two waves.

The principle of superposition yields:

$$P_o(t) + P(t) = -e \sin(2\pi ft) \tag{3-1}$$

Thus, the resultant amplitude is small if the two waves have nearly the same amplitudes.

Mechanical generation of a cancellation wave (obtained from the original wave) requires a delay of 180°. That is:

$$P(t) = P_o(t) \text{ delayed } 180^o,$$

$$= A\sin(2\pi ft - \pi)$$

$$= A\sin(2\pi f(t - \Delta t)). \qquad \text{Thus:}$$

$$2\pi f\Delta t = \pi \text{ radians} = 180^o, \qquad \text{or}$$

$$\Delta t = \frac{\pi}{2\pi f} = \frac{1}{2f}, \qquad \text{but} \tag{3-2}$$

$$f = \frac{V_0}{\lambda_0} = \frac{V_1}{\lambda_1}$$ from acoustic theory (65). Therefore:

$$\Delta t = \frac{\lambda_0}{2V_0} = \frac{\lambda_1}{2V_1} \tag{3-3}$$

where: Δt = time delay between waves, sec.

V = local speed of sound ± the gas velocity through the system, ft/sec.

λ = wavelength, ft.

Subscripts 0 and 1 refer to the original wave and the cancellation wave, respectively.

The above time delay can be obtained by dividing the flow of the original wave, then lengthening the flow path of one branch before recombining the waves. The length differential, ΔL, required between branches is:

$$\Delta t = \frac{L_1 - L_0}{V_1} = \frac{\Delta L}{V_1}, \qquad \text{or:}$$

$$\frac{\lambda_0}{2V_0} = \frac{\Delta L}{V_1}$$ from equation (3-3). Therefore:

$$\Delta L = \left(\frac{V_1}{V_0}\right)\frac{\lambda_0}{2} \tag{3-4}$$

When the wave propagation velocities are equal, equation (3-4) indicates that the required delay is obtained if the cancellation wave travels a distance $\lambda_0/2$ farther than the original wave.

In this study, a side-branch (quarter-wave) resonator was used to mechanically generate the delayed cancellation wave. This resonator was a straight pipe which formed a 90° branch off the main exhaust pipe. It was closed at the far end. Part of the pressure wave in the main pipe was transmitted down the side-branch. At the far (closed) end the wave was reflected without a change in sign[1] and it returned to the branch to combine (interfere) with the wave transmitted through the main pipe. The length of the branch determined the delay of this returning cancellation wave. The distance travelled by the wave was twice the length, L, of the side-branch; therefore, for proper cancellation:

$$2L = \Delta L$$

$$= \left(\frac{V_1}{V_0}\right) \frac{\lambda_0}{2} \qquad \text{from equation (3-4). Or:}$$

$$L = \left(\frac{V_1}{V_0}\right) \frac{\lambda_0}{4} \qquad (3\text{-}5)$$

A side-branch of length determined by equation (3-5) is designed to cancel the frequency component $f = V_0/\lambda_0$ in the pressure wave.

---

[1]A positive pressure is reflected as a positive pressure, where the zero pressure reference is the average pressure in the pipe.

CHAPTER IV

EQUIPMENT AND PROCEDURES

Measurement of Variables

Exhaust system dynamic pressure was measured by two piezoelectric pressure transducers (Kistler Instrument Corporation, Model 701A) mounted in water-cooled adaptors for protection against the high temperature environment.[1] The adaptor was flush-mounted with the inside wall of the exhaust pipe.

The Model 701A transducer was selected because of its high resolution (limited only by the amplifier noise) and capability for water-cooling. These and other important considerations for transducer selection are discussed in the literature (13, 23, 48).

An undesirable characteristic of piezoelectric transducers is acceleration sensitivity: 0.03 psi/g for the Model 701A (45). To reduce the influence of vibration on the transducer output, the exhaust system was isolated from the tractor by a flexible joint of asbestos gasket material and by external support, see Figure 2. The results of this vibration isolation are seen in Figure 3. Here the transducer was

---

[1]An iron-constantan 14 A. W. G. thermocouple junction, positioned 6½ inches above the exhaust manifold of the standard exhaust system and at the center of the exhaust pipe, attained a temperature of 1125°F when the tractor was operated at rated speed with load. Under the same conditions, a 20 A. W. G. chromel-alumel thermocouple imbedded in the pipe wall recorded an inside wall temperature of 680°F.

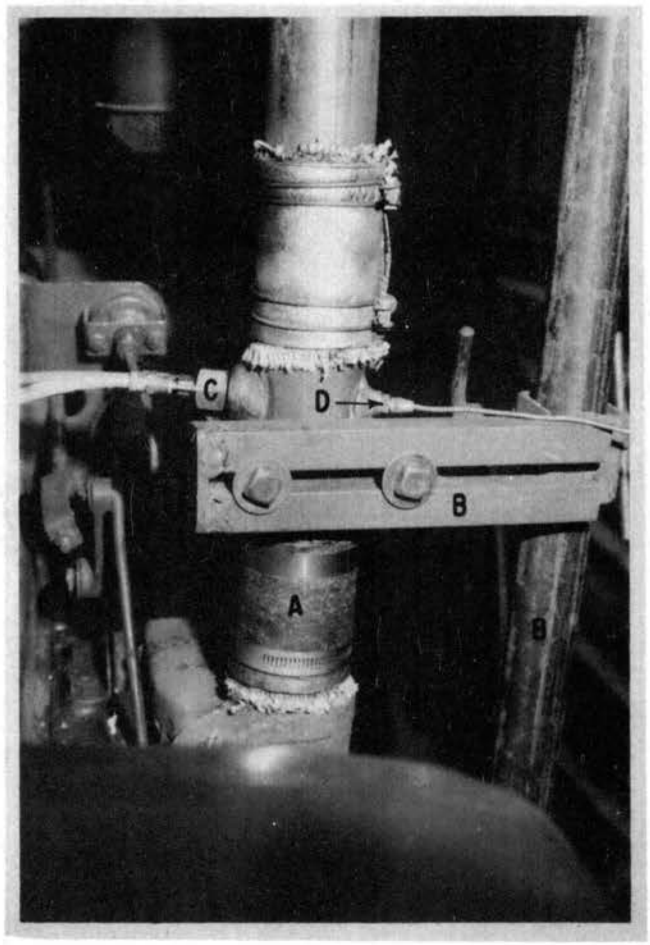

Figure 2. Vibration - Isolated Exhaust Pipe with Pressure Taps.
A - Flexible Joint;
B - External Exhaust System Support;
C - Pressure Transducer in Water-Cooled Adapter;
D - Manometer Pressure Tap

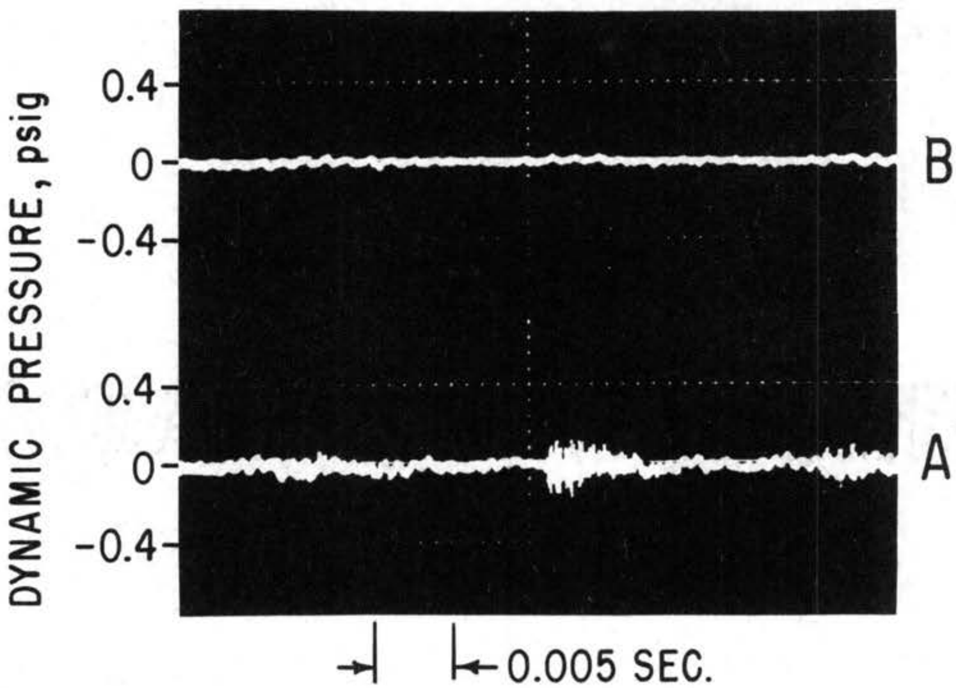

Figure 3. Pressure Transducer Output Due to Vibration.
A - Without Vibration Isolation;
B - With Vibration Isolation

located in a short length of pipe attached to the side of the standard exhaust system. The transducer output was caused solely by vibration from the tractor since there was no pressure input to the transducer. The output in Figure 3 - B was a "false pressure" which appeared as an error superimposed on the actual exhaust system pressures when measured in this study.

Each pressure transducer output was amplified by a Kistler Instrument Corporation Model 568 charge amplifier prior to being recorded on tape. The cable between the transducer and amplifier was short (5 ft) to minimize the detrimental effect of cable capacitance on signal-to-noise ratio (43). The amplifier gains were adjusted to yield amplifier outputs of 1-volt when the transducers were subjected to a dynamic pressure of 1-psi amplitude. The tape record of the pressures was calibrated with a 1-volt (rms), 250-Hz signal. This signal was obtained from the charge amplifier with a sine wave oscillator connected to its calibration input terminal.

Since piezoelectric transducers respond only to dynamic pressure, a U-tube manometer was used to measure the average pressure at the transducer location. The uneveness of the exhaust discharge pressure must be averaged to yield a reliable reading. Averaging was accomplished by connecting the manometer to a 1/8-inch copper tubing in series with a 10-foot length of 1/4-inch rubber hose, see Figure 2. This long, small diameter tubing eliminated standing waves in the line (27). Water was selected for the manometer fluid because average pressures were below 0.144 psig (4 inches of water). A disadvantage of the manometer is that large errors can occur when average pressures are much less than 40 inches of water (27). However, the pressures observed in this study were repeatable and appeared to vary realistically with changes in exhaust system configuration.

Exhaust system noise was sensed by a Brüel and Kjaer Type 4145, 1-inch diameter, condenser microphone coupled to their Type 2204 impulse precision sound level meter. The microphone, with Type UA-0207 windscreen, was positioned 6 inches beyond the exhaust pipe outlet and 6 inches from the O.D. of the pipe (measured along a radius directed toward the rear of the tractor) to isolate other noise sources (16,20). The microphone diaphram was perpendicular to a line joining it and the exhaust outlet.

Noise measurements may be affected by ambient conditions such as wind. Thus, tests were not conducted when the average wind velocity exceeded 10 mph. Other measurements considerations are discussed in the literature (12, 44, 54, 60).

The sound level meter gain was adjusted to give a meter deflection between zero and 10 dB for the slow meter response. The output of the sound level meter then passed through an attenuator box, which approximately halved the voltage, prior to being recorded on tape. This record was calibrated with a 124 dB, 250 Hz signal generated by a Type 4220 pistonphone coupled to the microphone.

A Sanborn-Ampex Model 2007 seven-channel tape recorder was used to store the pressure and noise analog data. The data were recorded on FM channels with a tape speed of 60 inches/second. Voice commentary and an engine rotational reference were also recorded.

The rotational reference was a voltage pulse emitted each time the number one cylinder passed the top-dead-center (TDC) position. A 1.5-volt "D" flashlight battery in series with a proximity switch produced the pulse when a metallic lobe on the flywheel passed the switch and actuated it. The position of the switch was adjusted to actuate at TDC with the aid of a stroboflash, triggered by the switch, and the TDC mark on the crankshaft pulley. The correct switch position was found to depend upon engine speed. Lyn, et al (48) suggested that this reference is accurate to 0.25° crank angle for no torsional deflection of the crankshaft, but multi-cylinder engines often have greater than 0.10° twist in the crankshaft (13). These errors may result in a timing error of about 0.04 msec at 1650 rpm. However, this was insignificant compared to the tape recorder interchannel time displacement error which was a maximum of 0.4 msec.

The procedure for recording the variables, pressure and noise, is briefly outlined below:

1. Locate tractor and PTO dynamometer at least 100 feet away from any buildings to avoid sound reflections. Allow the tractor to reach operating temperature.
2. Set up required instrumentation and select desired pressure transducer locations. Calibrate tape records for pressure and noise. Set proximity switch to indicate TDC on the number one cylinder.
3. Record atmospheric conditions and ambient noise level. Identify test on tape recorder voice channel.
4. Set desired tractor load and speed. Make a 4-minute recording of the variables.

This procedure was followed in tests of each exhaust system configuration listed in Table VI. In all tests noise was measured at the exhaust outlet.

TABLE VI

SUMMARY OF TESTS

Test Number	Exhaust System Configuration	Pressure Tap Location(s)	Average Pressure, Inches of Water
1	Standard manifold and muffler	7 inches above manifold flange	3.50
2	Standard manifold. Muffler removed (2-inch I.D. x 9-inch pipe above manifold)	7 inches above manifold flange	0.60
3	Standard manifold. Muffler replaced by an equal length of 2-inch I.D. pipe	7 inches above manifold flange	2.60
4	Single-cylinder exhaust pipe 1¼-inch I.D. x 44-inch length from engine block	6 inches beyond engine block	3.40
		36 inches beyond engine block	---*
5	Same as Test No. 4	6 inches beyond engine block	3.40
		24 inches beyond engine block	1.50
6	Standard manifold and muffler with 18-inch length of 2-inch I.D. pipe above manifold containing a side-branch 9½ inches above manifold	5½ inches below side-branch ℄	---**
		5½ inches above side-branch ℄	1.50

*This reading was not taken during the test.

**A manometer tap was not placed at this location.

Tests 1, 2, and 3 were performed to obtain a basis for comparing proposed exhaust system modifications. Test 2 was selected to provide base line data since it would probably be the loudest condition experienced by tractor operators. Comparison of Tests 1 and 3 would indicate the effective frequency range of the standard muffler (see Figure 1) and the frequencies which were main contributors to exhaust noise with and without a muffler. Test 3 and the single-cylinder study (Tests 4 and 5) were performed to determine the potential of these systems for wave cancellation. In addition, the speed of sound in an exhaust pipe could be readily obtained from the single-cylinder study.

The system for Test 6 was designed to provide silencing by wave cancellation based on the results of the simulation study of cancellation for the systems of Tests 1, 3, and 5. The side-branch resonator was selected as a phase-shifting device because of its simplicity. Other methods to mechanically produce phase shift were not pursued since the primary intent of this research was to develop a procedure for experimental and simulated analysis of internal combustion engine exhaust systems.

Preliminary Analog Analysis

After collecting the test data, a preliminary analog frequency analysis was performed. A Brüel and Kjaer Type 2107 frequency analyzer, its gain adjusted with the aid of the tape calibration signals, was used for one-third octave band (approximate) and 6% narrow band analyses of each pressure and noise record. The narrow band analysis could be used to identify frequency components in the noise or pressure spectrum (see Figure 1), while the one-third octave band analysis was used to calculate the total loudness of the exhaust noise.

The procedure for computing total loudness in sones is described in USA Standard S3.4 (67). It is commonly known as Stevens' Method. A loudness index, I, is obtained from a graph for each band level. These indices are then combined into one number, S_t, which represents the total loudness:

$$S_t = I_m + F(\Sigma I - I_m) \qquad (4\text{-}1)$$

where: I_m = the largest loudness index.

ΣI = the sum of loudness indicies for all bands.

F = 0.15 (for one-third octave bands).

Bands with high-loudness indicies are, thus, the main contributors to total loudness.

The total loudness procedure assumes the noise is diffuse (comes from all directions) and broadband (without sharp peaks in the spectrum). Both assumptions are violated here. However, comparisons are made only between spectra of similar character. Hence, this procedure is used to evaluate the relative effectiveness of the wave cancellation exhaust system.

Digital Analysis and Simulation

Digital analysis was selected because of its flexibility, the availability of an analog-to-digital conversion system, and the lack of proper analog analysis instrumentation.

A block diagram of the digital analysis system is shown in Figure 4. The analog tape record of test data was played back for preliminary amplitude analysis on the oscilloscope. These voltage amplitudes

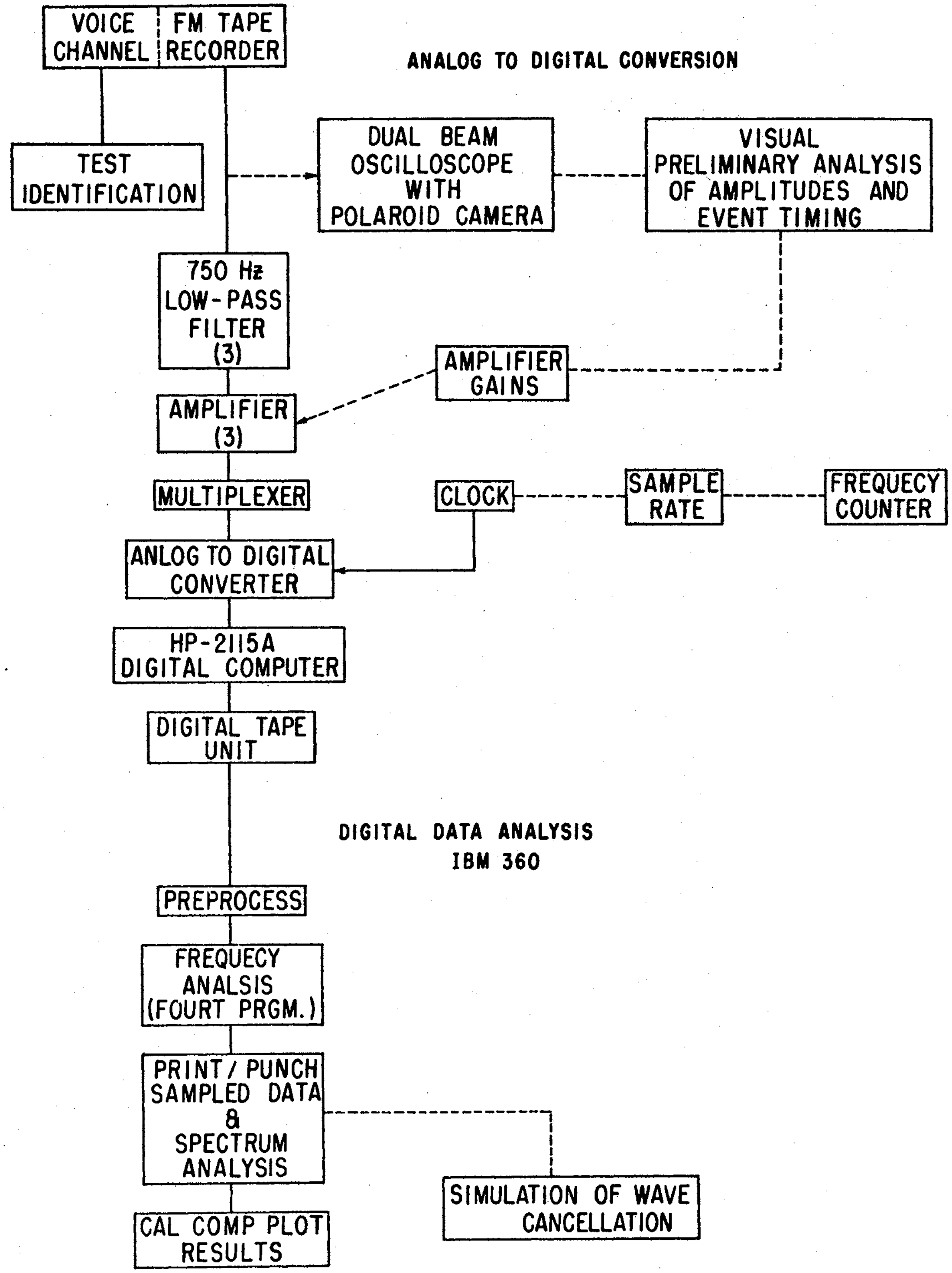

Figure 4. Block Diagram of Digital Analysis System

determined the required gains of the analog-to-digital converter (ADC) input amplifiers to give a ± 10-volt input to the ADC for maximum resolution.

Low-pass filters were used to prevent aliasing.[2] Their cut-off frequencies were set to remove any frequencies in the data greater than one-half the ADC sampling rate.[3] Since the sampling rate for each data channel was 1922 samples/second (sps), a cut-off frequency of 750 Hz insured adequate attenuation of the frequencies above 960 Hz. This sampling rate was the maximum attainable for the ADC system when using the multiplexer.

The multiplexer was a sample-and-hold device which allowed several channels of analog data to be digitized at one time while maintaining their interchannel time relationships. Unfortunately these time relationships were distorted by the FM tape recorder. This would appear in any phase relationship calculated from the digital data.

The noise and pressure variables were digitized for each test. A record of 167 sequential samples for each variable was written on digital tape by the Hewlett Packard computer and tape unit. All variables for each test were digitized simultaneously through the use of the multiplexer. The digital samples had a resolution about 0.01-volt since a 10-volt input to the ADC yielded a whole number digital output of 1023. An example of the calibration procedure to relate the numerical

---

[2]Aliasing is the appearance of a high frequency component as a lower frequency component folded about the Nyquist frequency because of insufficient samples to distinguish the two frequencies (29).

[3]The sampling rate should be at least 2½ times the highest frequency of interest in order to distinguish this frequency component. The sampling interval, or time between samples, is the reciprocal of the sampling rate (29).

amplitude of the digital samples to the analog pressure amplitude is presented in Appendix A.

The digitized data were analyzed on an IBM Model 360/65 computer. An IBM library subroutine, Cooley-Tukey Fast Fourier Transform--FOURT (11), was used to frequency analyze the first 128 samples of each data record after pre-processing (calibration). An integer power of two ($128=2^7$) was used for the record length because the fast Fourier transform procedure is highly inefficient for other lengths. Analysis of a square wave and a triangular wave, of periods comparable to the data, showed the 128-sample record length satisfactory in evaluating Fourier series coefficients. Coefficients calculated from records of 64 and 256 samples were also compared to the actual series coefficients for these wave forms. Final selection of the 128-sample record length was also determined by the capacity of the analog-to-digital converter.

A preliminary check of the subroutine indicated that its output must be multiplied by $\frac{2}{N} = \frac{2}{T} \Delta t$ to obtain estimates of the Fourier coefficients for the series representing the data; where: N = the number of samples (128), T = the record length in seconds, Δt = the sampling interval in seconds (0.00052). The data were assumed to be a periodic extension of the record length analyzed (8). However, 140 to 143, rather than 128 samples, formed a period or integer multiple of a period for the data. Thus, the frequencies at which the Fourier transform was evaluated (multiples of 1/T) differed from harmonics of the engine rotational frequency. For example, the rotational frequency was about 27.5 Hz, but the analyzed frequency was 30.0 Hz. Since deviation from the power-of-two record length may result in up to at least a 50 fold increase in computation time (11), it was concluded that the potential cost increase did not justify the improved accuracy of a 140-sample record length. An alternative solution would be reduction of the sampling interval, Δt, so that 128 samples covered a period or integer multiple of it. This would involve careful selection of the sampling rate and potential reruns of the digitizing procedure. In addition, the ADC sampling rate clock was very sensitive, making fine adjustments difficult.

The Fourier series coefficients obtained from subroutine FOURT were converted to polar form to yield magnitude (linear and logarithmic) and phase angle:

$$f(t) = \frac{a_0}{2} + \sum_{n=1}^{\infty} (a_n \cos n\omega t + b_n \sin n\omega t) = \sum_{n=-\infty}^{\infty} \bar{c}_n e^{in\omega t}$$

where: a_n and b_n = Fourier series coefficients.

$\bar{c}_n$ = complex Fourier series coefficients.

ω = $2\pi f$.

f = fundamental frequency.

i = $\sqrt{-1}$.

t = time.

$f(t)$ = a function dependent upon time.

The Euler definitions for sine and cosine can be used to obtain the relationships for polar form (35, 36):

$$\bar{c}_n = |\bar{c}_n| \, e^{i\phi_n}$$

where magnitude $$|\bar{c}_n| = \frac{1}{2}\sqrt{a_n^2 + b_n^2}\ , \text{ and} \qquad (4\text{-}2)$$

phase angle $$\phi_n = \tan^{-1}\left(-\frac{b_n}{a_n}\right) \qquad (4\text{-}3)$$

The linear magnitude given by equation (4-2) was converted to decibels by equation (4-4):

$$\text{dB} = 20 \log_{10} \frac{|\bar{c}_n| \text{ psi}}{4.10 \times 10^{-9} \text{ psi}} \tag{4-4}$$

where the denominator is a reference for peak pressures derived from the standard decibel rms reference of 0.0002 μbar.

The digital magnitude and phase spectrums for each data channel were punched on cards for graphing the spectrums on a Cal Comp plotter and for input to the simulation program. An example of an amplitude (magnitude) spectrum plot is seen in Figure 5. Its time domain plot is shown in Appendix A. The first three peaks in the spectrum occurred at harmonics of the engine firing frequency.

The transfer function analysis described in Chapter II, equation (2-3), was applied to relate pressure and noise for three of the exhaust system configurations studied. The amplitude spectrum of the transfer function was obtained from the ratio of the exhaust noise and pressure amplitude spectrums, where pressure and noise were considered the system input and output, respectively. The phase shift between the input and output could be obtained from the difference between their phase spectrums. However, calculation of this phase shift was not pursued because of the large error introduced by tape recorder interchannel time displacement. Hereafter the term "transfer function" will be used instead of the term "amplitude spectrum of the transfer function." An example of three transfer functions is shown in Figure 6.

The experimentally-derived transfer function was used to simulate noise reduction by wave cancellation. For simplicity, cancellation only by superposition of a time-delayed image of the original pressure

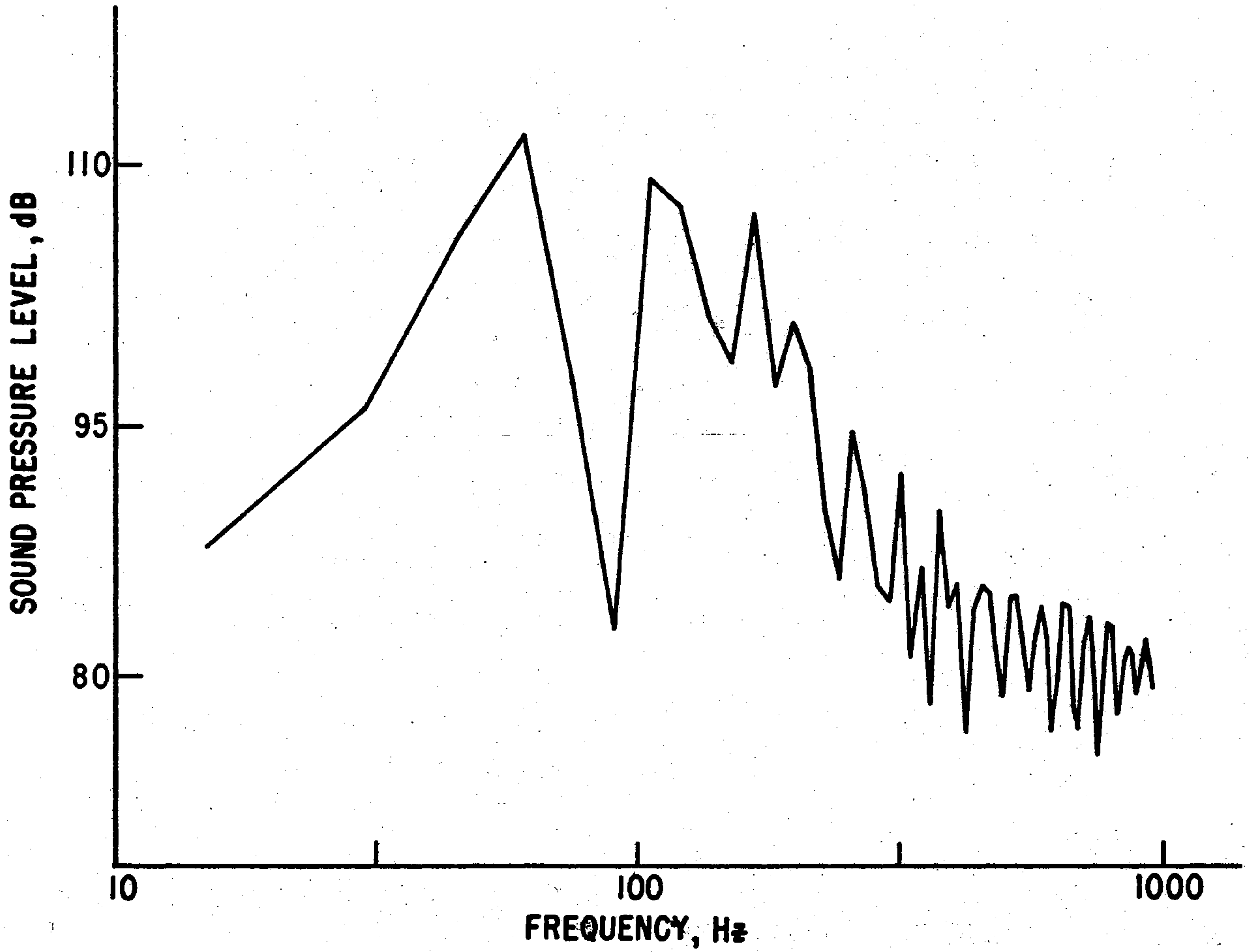

Figure 5. Digital Exhaust Noise Spectrum of Standard Exhaust System (Test No. 1)

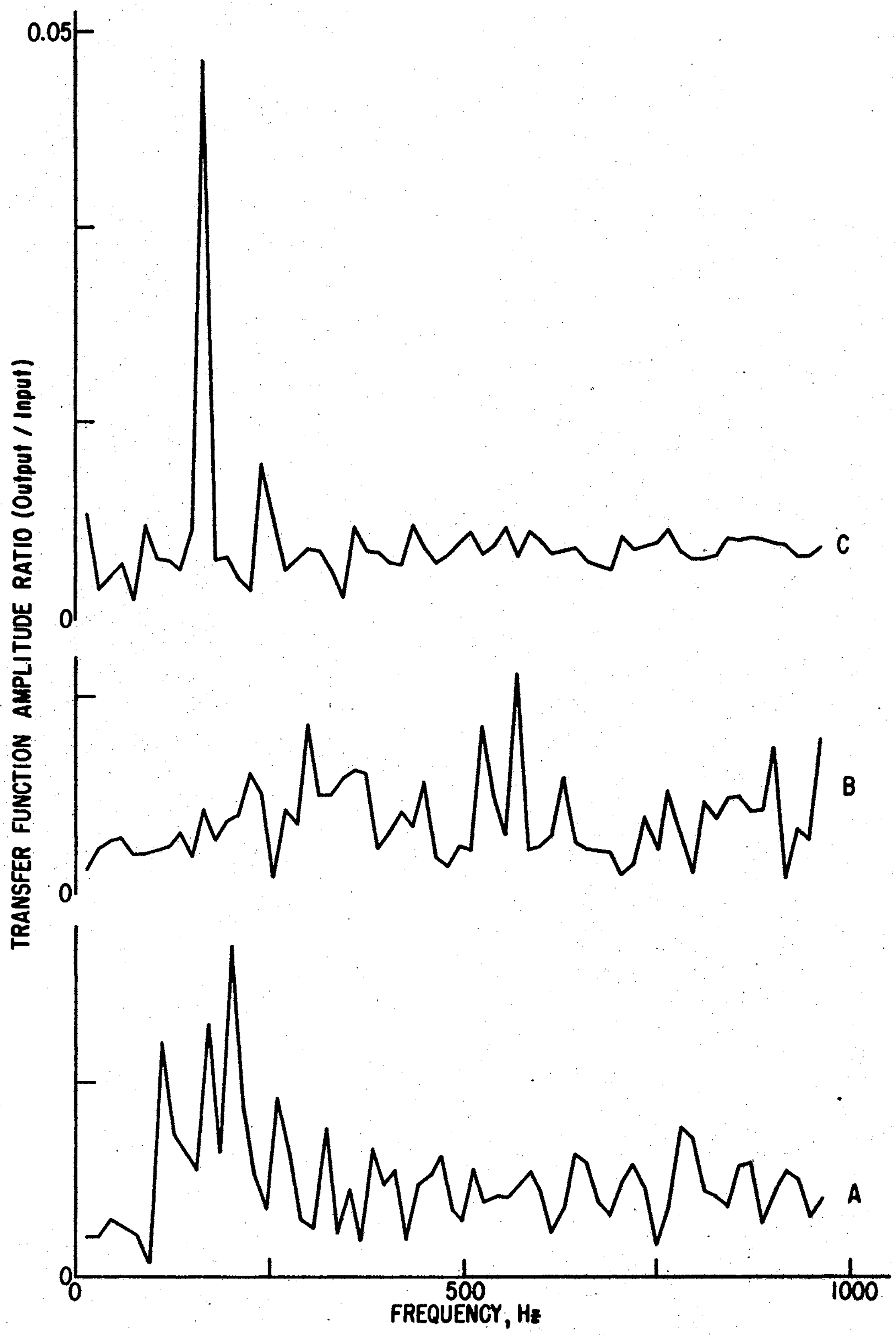

Figure 6. Amplitude Spectrum of Transfer Function.
A - Standard Exhaust System (Test No. 1);
B - Pipe Replacing Muffler (Test No. 3);
C - Wave Cancellation System (Test No. 6)

wave was simulated on the computer. The delay of the cancellation wave was such that its positive pressure peaks would coincide with negative peaks of the original wave, and vice versa, see curves A and B in Figure 7. This combination would correspond to 180^{o} phase shift if the waves were sinusoids. The intent of the cancellation was to minimize the pressure-time fluctuations resulting from the superposition. Then the transfer function of the system under analysis could be applied to obtain a projected noise spectrum from the resultant pressure spectrum.[4] The total loudness of this simulated noise spectrum was used to evaluate the effectiveness of wave cancellation for a particular exhaust system by comparing it to the total loudness of the same system without cancellation.

Experimental Wave Cancellation

The results of the simulated wave cancellation study, see Table IX in Chapter V, showed that the most noise reduction for the systems studied was obtained by applying the cancellation principle to the standard exhaust system. Thus, the system in Figure 8 was designed to experimentally evaluate the effectiveness of wave cancellation.

The side-branch resonator, described in Chapter III, generated the cancellation wave. The correct phasing of this and the original wave was obtained by adjusting the side-branch length with the plunger. The plunger was positioned to give the minimum dynamic pressure amplitude at the top pressure tap in the main pipe as seen on an oscilliscope display. For the test conditions, 280 inch-pound PTO torque and 1700

---

[4]The pressure spectrum was obtained by application of subroutine FOURT to the pressure-time history from the superposition.

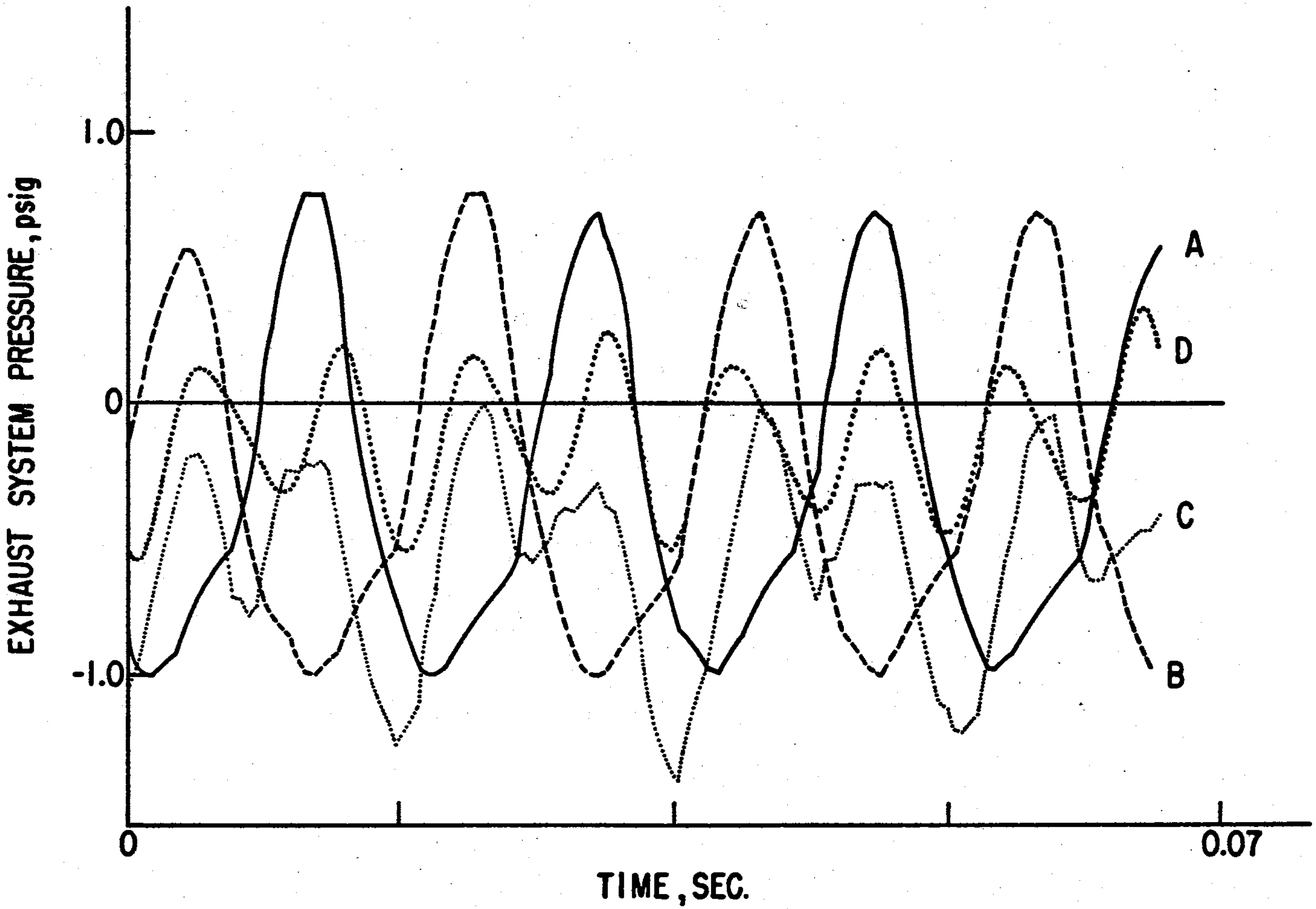

Figure 7. Exhaust System Pressure Waves for Simulated Wave Cancellation (Test No. 1).
A – Original Pressure Wave;
B – Cancellation Wave;
C – Resultant Wave;
D – Experimental Resultant Wave (Test No. 6)

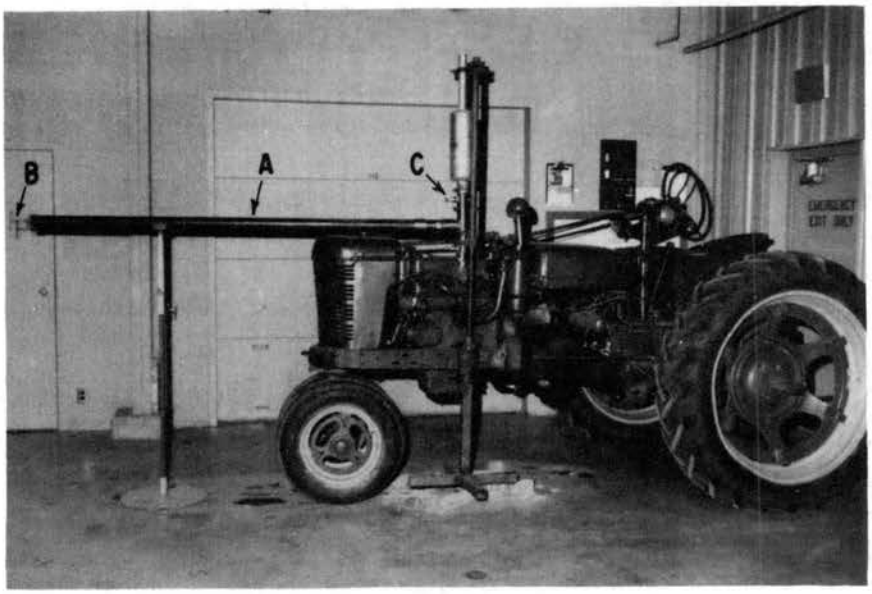

Figure 8. Wave Cancellation Exhaust System.
A - Side-branch Resonator;
B - Plunger;
C - Pressure Tap (upper)

rpm, this branch length was 62.5 inches. The corresponding length calculated from equation (3-5) was 65.7 inches, with the wavelength, λ_0, determined by the engine firing frequency and the average speed of sound in the exhaust system. The average speed of sound was 1245 ft/sec; as evaluated from the time required for a pressure wave to travel past a pressure transducer to the end of a straight exhaust pipe and back to the transducer again in the single-cylinder exhaust system shown in Figure 9. The measured speed of sound for this system varied over a range of 1112 to 1333 ft/sec. Further details of the single-cylinder study are presented in Appendix B.

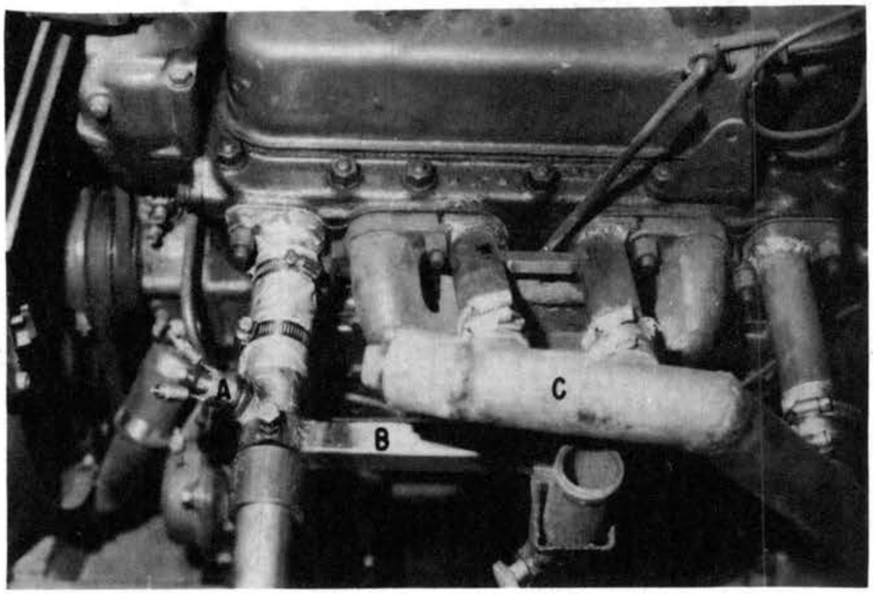

Figure 9. Exhaust Manifold for Single-Cylinder Study. A - Pressure Tap in Single-Cylinder Exhaust Pipe; B - External Support for Vibration Isolation; C - Exhaust System for Remaining Cylinders

CHAPTER V

RESULTS

Relation of Exhaust System Pressure and Noise

The exhaust system transfer function related dynamic pressure at a given location within the system to noise at the exhaust outlet. Recall that the analog test data were passed through a 750-Hz low-pass filter prior to being digitally sampled. Therefore, the transfer functions and other spectrums generated by computer analysis are representative only of low frequency exhaust pressure and noise.

Figure 6 shows the transfer functions that were of prime interest. These relationships were of value in projecting the effect of pressure wave modifications on exhaust noise. For the standard exhaust system, the frequency range of 90 to 250 Hz exhibited variability. Comparison of pressure and noise (system input and output) decibel vs. linear frequency plots indicated no clearly defined nonlinearity,[1] see Figure 10. However, above 300 Hz there was a cyclic variation in the output (SPL) spectrum which repeated every 60 to 75 Hz.

The system for Test 3 appeared nonlinear in the vicinity of 350 and 700 Hz. However, according to equation (2-1), the straight pipe used in this test had resonant frequencies which were odd multiples of

[1]Nonlinearities occur when a frequency component of the input appears in the system output with expanded harmonics of this frequency. If the input is more complex than a sinusoid, nonlinearity may be obscured by other portions of the spectrum (2).

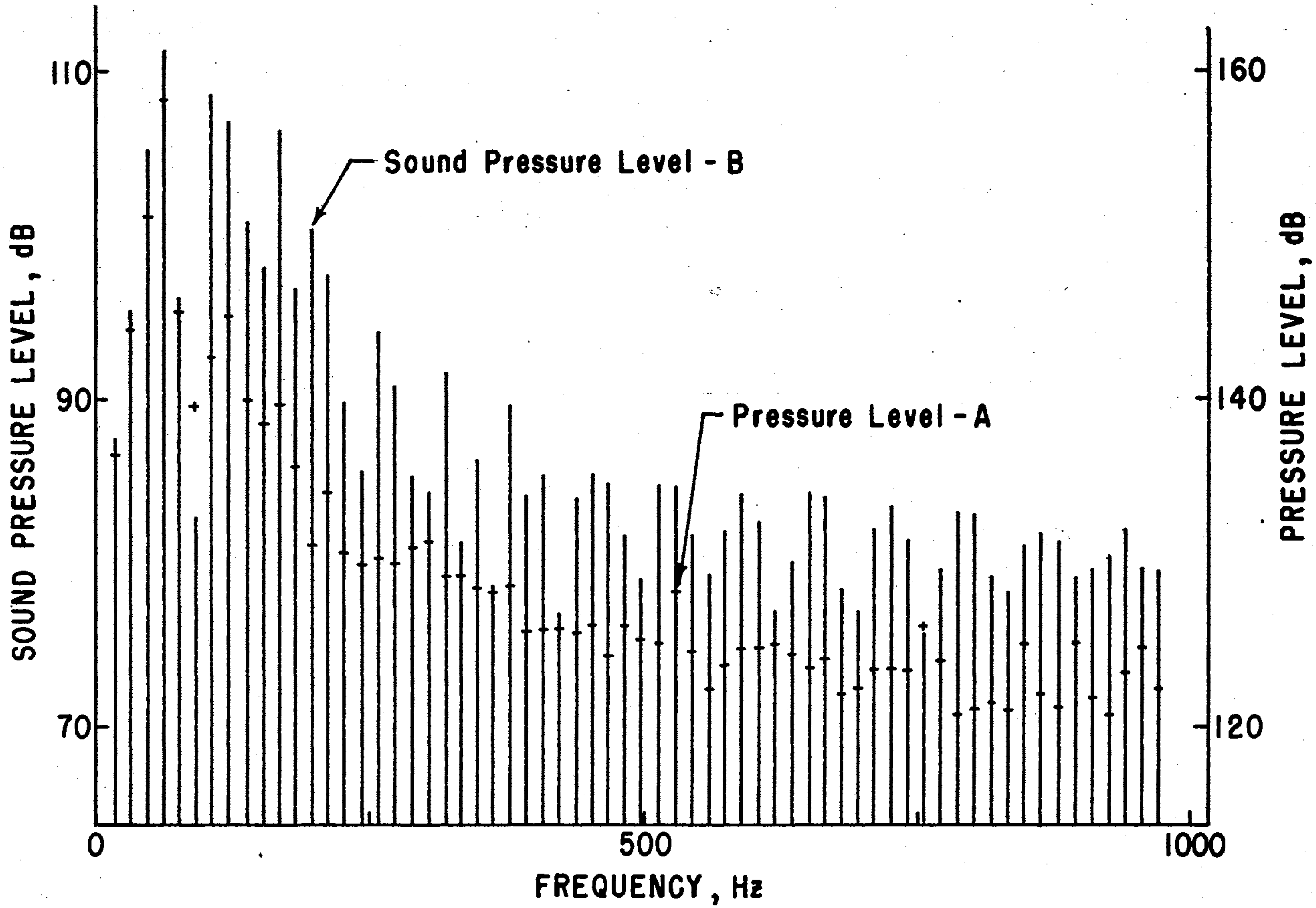

Figure 10. Linear Frequency Spectrums (Test No. 1).
A - Exhaust Pressure; B - Exhaust Noise

104 Hz (based on a speed of sound of 1245 ft/sec). Thus, 312 and 728 Hz were resonant frequencies for the pipe. These resonances caused the peaks in the transfer function (Figure 6 - B) which were initially considered nonlinearities. Such effects were not readily apparent at other resonant frequencies.

The large peak at 165 Hz in the transfer function of Test 6 (Figure 6 - C) occurred only in a single 15-Hz band. Hence, it was more likely caused by "noise" than by a system nonlinearity at that frequency. Here "noise" refers to a random lowness in the input spectrum (from the expected value) because of the fast Fourier transform averaging procedure used in the digital analysis (2). Further investigation of nonlinear effects was not attempted since it was not the intent of this study.

Identification of Major Noise Components

The major noise frequency components, in regard to loudness, were of interest for the standard exhaust system (Test 1) and for additional silencing improvements (Test 6). Identification of these components in Test 1 was the first step in an attempt to reduce exhaust noise. The most pronounced reduction in loudness will generally result from initial reduction of the major contributors of noise. Table VII presents the loudness indicies for Tests 1 and 6. These indicies were obtained from an analog one-third octave band analysis and the standard procedure for computing loudness (67). The data were not filtered before this analysis.

The highest loudness indicies of Test 1 were for the frequencies 63 through 250 Hz and above 2500 Hz. The lower range included the first

TABLE VII

EXHAUST NOISE LOUDNESS INDICIES

1/3 Oct. Band Center Freq., Hz	Test No. 1		Test No. 6	
	Sound Pressure Level, dB	Loudness Index, I	Sound Pressure Level, dB	Loudness Index, I
40	99.0	20.0	84.0	7.0
50	109.0	53.0	85.5	9.0
63	114.5	90.0	94.0	18.0
80	108.0	58.0	93.5	19.0
100	106.5	55.0	98.0	28.0
125	116.0	125.0	107.5	62.0
160	110.5	90.0	105.0	58.0
200	108.0	78.0	95.5	30.0
250	105.5	70.0	91.5	24.0
315	99.0	46.0	89.5	23.0
400	95.5	38.0	86.0	18.0
500	93.5	35.0	84.0	18.0
630	92.5	35.0	83.0	18.0
800	93.0	39.0	83.5	20.0
1000	93.0	42.0	83.0	20.0
1250	92.5	42.0	82.0	20.0
1600	92.0	45.0	83.0	24.0
2000	91.0	46.0	81.0	22.0
2500	91.5	52.0	82.0	25.0
3150	92.0	58.0	84.0	31.0
4000	92.5	65.0	88.0	45.0
5000	93.5	75.0	89.0	54.0
6300	95.0	90.0	89.5	60.0
8000	94.0	90.0	88.0	58.0
10000	93.0	75.0	86.5	45.0

three harmonics of the engine firing frequency. It also contained the maximum noise index. A major portion of the exhaust noise from the standard system was in this low frequency range. However, the standard muffler was not effective in this range as shown by Figure 1. The data from Test 6 are discussed in a later section of this chapter.

Simulation of Wave Cancellation

Wave cancellation was simulated for the exhaust systems of Tests 1, 3, and 5 described in Table VI. The results are shown in Table VIII.

TABLE VIII

EXHAUST NOISE TOTAL LOUDNESS

Test Number	Total Loudness, sones			
	Analog Analysis (Actual Test)		Digital Analysis (40-800 Hz)	
	40 - 10000 Hz	40 - 800 Hz	Actual Test	Simulated
1	333.0	231.0	99.2	83.2*
2	443.0	291.8	124.0	---
3	421.6	239.8	106.6	100.9
4	226.0	138.0	117.8	---
5	236.8	139.1	100.3	113.0
6	166.1	105.5	51.2	---

*The simulated cancellation for Test 1 is comparable to the actual Test 6.

The total loudness was calculated from equation (4-1). The analog portion was from a one-third octave band analysis while the digital section was a "pseudo" one-third octave band analysis. These digital bands were formed by averaging the sound pressure levels (at least two) of the 15 Hz bands from the digital analysis which fell in or near each one-third octave band. Although the analog and digital total loudness in Table VIII are not of the same magnitude, they show the same trends between tests and their ratio within each test is 2.1 to 2.4, except for the single-cylinder tests. The single-cylinder data had a period four times longer than the other data. Thus, the resolution of the FOURT program was reduced at the low frequencies (since the record length was not increased proportionately) and the resulting noise spectrums did not have the variability seen in the other tests.

The simulated total loudness showed that cancellation by superposition of a time-delayed image of the pressure wave was detrimental in Test 5. Some improvement was seen in Test 3, but the greatest reduction in loudness occurred for Test 1. The output of the simulation program for Test 1 is shown in Table IX. Also included are the experimental results of Test 6 for comparison purposes.

The "Original Experimental Decibels" columns of Table IX are the frequency spectrums of data from Test 1. The "Experimental Decibels" columns under "Wave Cancellation" are from Test 6. The output of the simulation program appears in the remaining two columns. Recall that the simulated cancellation wave was a time-delayed image of the original pressure wave, see Figure 7. The resultant pressure wave, f(t), of this superposition is then:

$$f(t) = a(t) + a(t + \tau)$$

and its Fourier transform is:

$$F(\omega) = A(\omega) + e^{i\omega\tau}A(\omega) \qquad \text{from the shifting theorem (53).}$$

$$= (1 + e^{i\omega\tau})A(\omega) \qquad (5\text{-}1)$$

TABLE IX

SIMULATED AND EXPERIMENTAL WAVE CANCELLATION
RESULTS FOR TEST NO. 6

FREQUENCY	ORIGINAL		WAVE CANCELLATION			
	EXPERIMENTAL DECIBELS		SIMULATED DECIBELS		EXPERIMENTAL DECIBELS	
(HZ)	PRESSURE	SPL	PRESSURE	SPL	PRESSURE	SPL
0.0	163.7	93.6	169.0	98.9	155.9	105.7
15.0	142.6	93.4	138.9	89.8	125.4	84.5
30.0	150.4	101.3	132.6	83.6	136.0	84.8
45.1	157.2	111.2	151.0	105.0	134.5	86.1
60.1	164.3	117.1	157.2	110.1	140.1	93.7
75.1	151.4	102.1	134.5	85.2	134.5	79.6
90.1	145.5	88.8	141.7	85.0	132.9	90.9
105.2	148.5	114.6	156.6	122.7	145.7	100.0
120.2	150.9	112.9	152.4	114.3	156.9	111.0
135.2	145.9	106.7	145.4	106.2	143.4	95.9
150.2	144.4	103.9	142.3	101.7	141.6	99.2
165.3	145.7	112.4	146.8	113.4	121.1	94.6
180.3	142.0	102.7	140.4	101.1	145.6	99.6
195.3	137.2	106.3	115.5	84.6	137.0	91.6
210.3	140.3	103.5	136.8	100.0	139.4	90.4
225.4	136.7	95.7	132.2	91.2	137.4	85.5
240.4	136.1	91.5	128.0	83.4	130.7	93.1
255.4	136.3	100.0	125.3	89.0	131.6	90.4
270.4	136.1	96.6	131.7	92.2	131.2	83.8
285.5	137.0	91.2	134.2	88.5	130.3	84.6
300.5	137.4	90.3	134.5	87.4	129.4	85.0
315.5	135.4	97.6	129.9	92.1	129.5	84.8
330.5	135.3	87.2	130.1	82.1	128.8	81.3
345.6	134.5	92.2	129.7	87.4	128.9	74.9
360.6	134.2	84.5	127.6	77.9	126.9	84.8
375.6	134.6	95.4	132.0	92.8	127.7	83.1
390.6	132.0	90.1	122.3	80.4	126.6	81.8
405.6	131.9	91.3	120.6	80.0	126.2	79.9
420.7	132.1	82.9	125.3	76.2	128.5	81.8
435.7	131.7	89.8	126.7	84.9	123.9	82.0
450.7	132.3	91.3	125.7	84.7	125.9	81.5
465.7	130.4	90.8	131.8	92.2	125.0	78.7
480.8	132.1	87.7	129.1	84.7	124.8	79.7
495.8	131.4	85.0	123.6	77.2	123.3	79.7
510.8	131.1	90.5	124.1	83.5	124.9	82.3
525.8	134.1	90.6	124.1	80.6	122.9	77.9
540.9	130.6	87.7	123.7	80.8	123.6	79.6
555.9	128.4	85.3	111.3	68.3	121.3	79.2
570.9	129.7	87.9	122.1	80.2	124.9	79.7
585.9	130.8	90.1	126.2	85.5	122.2	79.7
601.0	130.8	88.4	125.4	82.9	122.1	78.7
616.0	131.1	83.0	124.0	75.9	123.6	78.6
631.0	130.5	86.0	122.9	78.5	123.1	78.4
646.0	129.6	90.2	125.6	86.2	123.0	78.8
661.1	130.1	89.9	127.8	87.6	123.1	77.0
676.1	128.1	84.4	117.3	73.6	122.9	76.0
691.1	128.3	83.1	114.8	69.6	122.4	74.9
706.1	129.5	88.0	125.2	83.6	121.9	78.9
721.2	129.5	89.4	123.7	83.6	122.0	77.5
736.2	129.5	87.3	123.4	81.2	122.5	78.4
751.2	132.0	81.7	130.1	79.9	122.5	78.8
766.2	130.2	85.5	126.9	82.3	122.7	80.5
781.3	126.8	89.1	123.4	85.6	121.7	76.9
796.3	127.2	88.8	114.9	76.4	121.8	76.1
811.3	127.5	85.1	124.3	81.9	122.4	76.6
826.3	127.1	84.1	121.7	78.7	122.0	76.8
841.3	131.1	87.0	126.5	82.4	121.6	78.4
856.4	128.0	87.7	121.9	81.6	121.5	78.0
871.4	127.2	87.2	115.4	75.4	121.9	78.7
886.4	131.2	85.1	127.8	81.6	121.3	78.0
901.4	127.8	85.5	125.0	82.6	121.8	78.0
916.5	126.8	86.2	124.3	83.7	121.9	78.1
931.5	129.4	88.0	125.6	84.2	121.9	76.7
946.5	130.9	85.5	127.9	82.6	121.6	76.4
961.5	128.4	85.4	121.0	78.0	121.7	77.4

where: $a(t)$ = original pressure wave.

$a(t + \tau)$ = cancellation wave.

$a(\omega)$ = Fourier transform of $a(t)$.

τ = the delay of the cancellation wave ≈ 0.0104 sec.

The magnitude of $F(\omega)$ is the magnitude of the original wave times the magnitude of $(1 + e^{i\omega\tau})$. Thus, the simulated pressure spectrum is the original spectrum multiplied by a sinusoid with amplitude varying from 0 to 2 and a period of τ. Since the frequencies analyzed by the FOURT subroutine seldom approach the frequencies where these maxima and minima occur, the cyclic variations between the original and simulated pressure spectrums are not readily apparent in Table IX. These variations may also be blurred by system nonlinearity (2). The levels at zero frequency, however, indicate the amplitude-doubling with about a 6-dB increase for the cancellation wave. The level at zero frequency is indicative of the average level, but may be in error for the experimental analyses because of zero-shift (dc bias introduced by the tape recorder and ADC input amplifiers).

The simulated exhaust noise spectrum was obtained from the product of the simulated pressure spectrum and the transfer function for the system of Test 1. This spectrum showed improvement over the original spectrum in all but three frequency bands. These bands were in the vicinity of maxima for $|1 + e^{i\omega\tau}|$. Comparison of noise levels from simulated and experimental wave cancellation systems showed the experimental levels to be less than the simulated levels, as would be expected from the data of Table VIII.

Experimental Wave Cancellation

The simulation results led to the selection of the standard exhaust system for experimental adaptation of the wave cancellation principle. This system was analyzed as Test 6. Comparison of the digital total loudness of this system (51.2 sones) with the simulated loudness (83.2 sones) indicated that simulation can be used to project the effect of exhaust system modifications. The discrepency between these two values was caused by at least two factors: (a) the system transfer functions were not identical (see Figure 6) as was assumed in the simulation, and (b) the side-branch resonator probably did not create the type of superposition used in the simulation because of nonlinearities and wave interference effects. The differences between the experimental and simulated resultant waves are seen in Figure 7. Only their frequency components at zero frequency were affected by the difference in average levels shown in this figure (similar to the effect of dc bias mentioned previously).

The improvement in silencing by the addition of wave cancellation is shown in Tables VII and VIII. In the 63 through 250 Hz range the indicies were less than one-half those of the standard system. Therefore, the contribution of this frequency range to the total loudness was decreased by more than a factor of two. In Table VIII the total loudness for Tests 1 and 6 indicated that exhaust noise was only half as loud after the addition of wave cancellation to the standard system. This was a substantial improvement in silencing.

CHAPTER VI

SUMMARY AND CONCLUSIONS

Tractor noise typically exceeds hearing conservation limits. Its major source is usually the engine exhaust system. Unfortunately, conventional theories for exhaust silencing are often inadequate and difficult to use. Hence, this study was initiated to determine the relation between exhaust system pressure and noise, and the frequencies which are the main contributors to exhaust noise. This information could then be used in the application of mechanically-induced acoustic interference to reduce exhaust noise.

The following conclusions can be derived from this study:

1. Exhaust system pressure and noise can be related by the amplitude spectrum of the experimentally-derived system transfer function. This relationship was essentially linear for much of the spectrum below 750 Hz, although nonlinearities could not be readily identified because of the complex spectrum of the system input.

2. The major noise frequency components were in the range of 63 to 250 Hz for the standard exhaust system. Since the first three harmonics of the engine firing frequency occurred in this range, the fundamental pressure frequency was a major cause of the exhaust noise.

3. Computer simulation of proposed modifications in existing systems was found to be practical when based on the experimentally-derived system transfer functions. However, a more complex simulation may be required for drastic modifications in the existing system and because of the $(1 + e^{i\omega\tau})$ effect on the simulated spectrum. The simulated analysis was comparable to the results obtained from experimental analysis. Simulation simplified the usual trial-and-error approach to muffler design because the proposed silencers were quickly evaluated on the computer without actual construction of the physical systems. Then the most promising silencer, of those simulated, was built for experimental testing.
4. The procedures developed in this study to evaluate existing and proposed exhaust systems were found to be adequate. The combination of transfer function analysis, computer simulation, and total loudness analysis should be an asset to muffler designers. The transfer function, derived experimentally from existing systems, determined the system dynamic characteristics without reliance on conventional exhaust system theories and their simplifying assumptions.
5. Mechanically-induced acoustic interference, or wave cancellation, proved to be a highly successful method of reducing exhaust noise for the tractor tested. When applied to reduce the fundamental pressure component, this silencing technique reduced exhaust noise to one-half its former loudness.

These conclusions led to the following suggestions for further work:

1. The foundations for noise reduction are the established hearing conservation criteria and maximum detrimental silencer effects tolerable by tractor manufacturers and owners. Although many limits have been proposed, additional definitive research on these human and mechanical limits would aid muffler designers and benefit tractor operators.
2. Wave cancellation with a side-branch resonator is an effective silencer for constant engine speed and load. However, speed and load vary for many tractor applications. Thus, investigation of means to govern the branch length with speed and to determine the effect of load on silencing could lead to wider application of this device. (Figures 11 and 13 show exhaust system pressure for various speed and load conditions.)
3. The side-branch resonator was chosen for simplicity since the primary intent of this research was to develop a procedure for experimental and simulated analysis of exhaust systems. In a detailed analysis of mechanical phase shifters, a more unique class of devices should be studied. Perhaps such things as mechanical analogs to electronic delay lines could be developed to obtain a more versatile and less cumbersome device than the side-branch resonator.

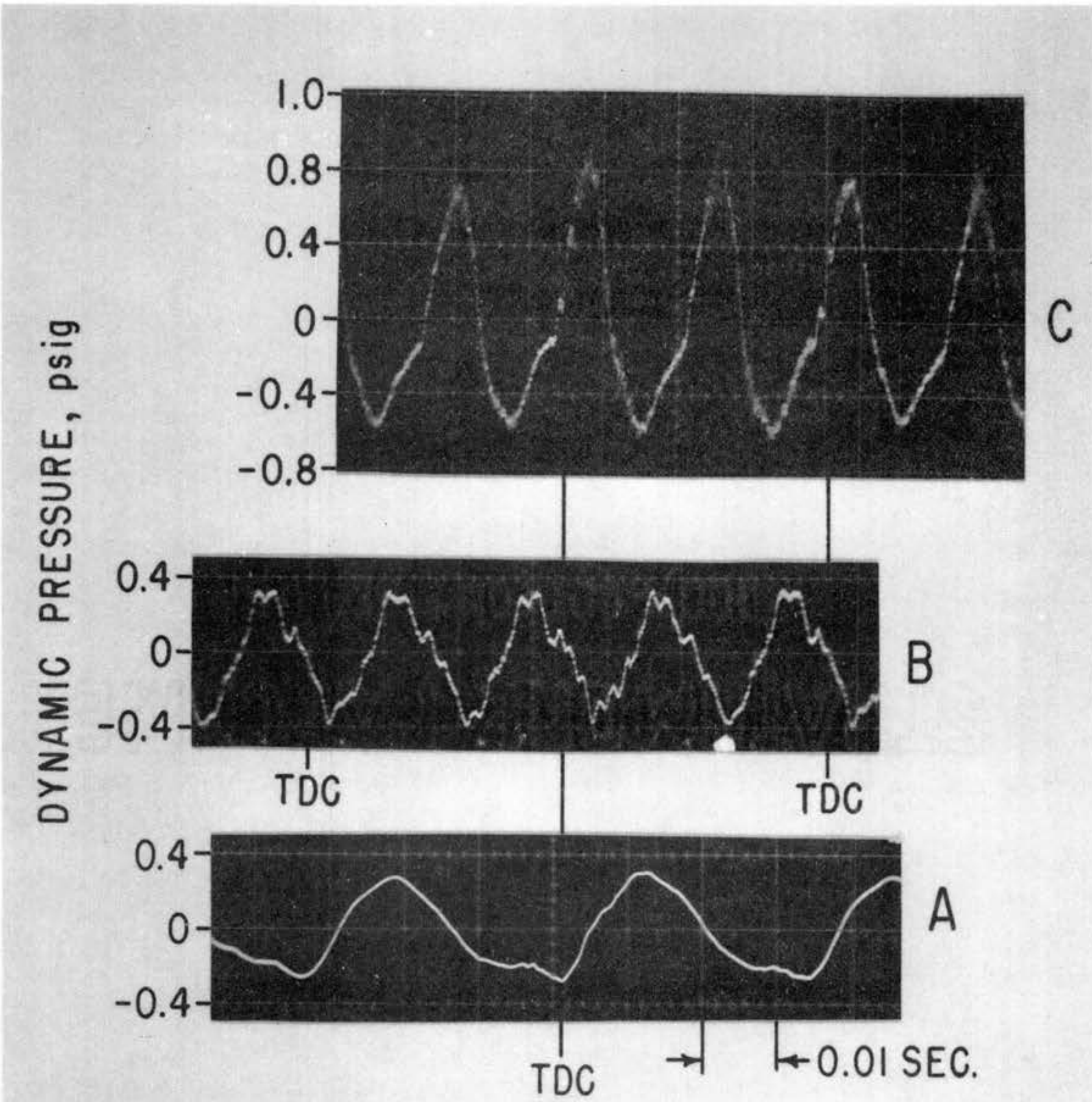

Figure 11. Pressure in Standard Exhaust System as a Function of Speed and Load.
A - Idle (870 rpm), No Load;
B - 1670 rpm, No Load;
C - 1670 rpm, PTO Load (Test No. 1)

BIBLIOGRAPHY

(1) Bannister, F. K. "Influence of Pipe Friction and Heat Transfer on Pressure Waves in Gases: Effects in a Shock Tube." Journal of Mechanical Engineering Science, Vol. 6, No. 3 (1964), 278-292.

(2) Basore, B. L. Personal Consultation. Electrical Engineering Department, Oklahoma State University, Stillwater, Oklahoma, May 3, 1971.

(3) Bendat, J. S. and A. G. Piersol. Measurement and Analysis of Random Data. New York: John Wiley and Sons, 1966.

(4) Benson, R. S., R. D. Garg, and D. Woollatt. "A Numerical Solution of Unsteady Flow Problems." International Journal of Mechanical Sciences, Vol. 6 (1964), 117-144.

(5) Benson, R. S. and H. M. F. ElShafie. "Non-Steady Flow Through a Square-Edged Orifice in a Pipe." Journal of Mechanical Engineering Science, Vol. 7, No. 4 (1965), 482-495.

(6) Beranek, L. L. Acoustic Measurements. New York: John Wiley and Sons, 1949.

(7) Beranek, L. L., ed. Noise Reduction. New York: McGraw-Hill, 1960.

(8) Bergland, G. D. "A Guided Tour of the Fast Fourier Transform." IEEE Spectrum, Vol. 6, No. 7 (1969), 41-52.

(9) Blair, G. P. and J. R. Goulburn. "Pressure-Time History in Exhaust System of High-Speed Reciprocating Internal Combustion Engine." Society of Automotive Engineers, Paper No. 670477, May, 1967.

(10) Bleazey, J. C. "Electronic Sound Absorber." Journal of Audio Engineering Society, Vol. 10 (1962), 135-139.

(11) Brenner, N. "Cooley-Tukey Fast Fourier Transform -- FOURT." Geophysics Department, Massechusetts Institute of Technology, Cambridge, Mass., 1968.

(12) Brock, J. T. Acoustic Noise Measurements. Cleveland: Brüel and Kjaer Instruments, 1967.

(13) Brown, W. L. "Methods for Evaluating Requirements and Errors in Cylinder Pressure Measurement." Society of Automotive Engineers Transactions, Vol. 76, Part 1 (1968), 50-77.

(14) Brueggeman, L. T. "Analyzing and Reducing Motor Noise." Allis-Chalmers Engineering Review, Vol. 32, No. 4 (1967), 25-29.

(15) Campbell, J. "Measuring Noise." Machine Design, Vol. 39, No. 21 (1967), 216-224.

(16) Chen, Y. R. "Analog Simulation and Interference Approach to Tractor Exhaust Noise Control." (unpub. M. S. thesis, North Carolina State University, 1968).

(17) Chisholm, T. S. and P. K. Turnquist. "A Method for Characterizing the Noise Produced by an Agricultural Tractor in Terms of Acoustic Power and Directivity." American Society of Agricultural Engineers, Paper No. 68-109, June, 1968.

(18) Committee on Conservation of Hearing. "Guide for Conservation of Hearing in Noise." Supplement to Transactions of the American Academy of Ophthalmology and Otolaryngology, 1964.

(19) Conover, W. B. "Fighting Noise with Noise." Noise Control, Vol. 2, No. 2 (1956), 78-82, 92.

(20) Cordier, O. and G. Reyl. "The Noise Problems of Air-Cooled Diesel Engines -- Measures Towards Its Reduction with General Observations and Specific Results." Society of Automotive Engineers, Paper No. 680405, January, 1968.

(21) Cundiff, J. S. "Finite Wave Theory Evaluation of Internal Combustion Engine Exhaust for Noise Control." (unpub. M. S. thesis, North Carolina State University, 1969).

(22) Davies, P. O. A. L. "The Design of Silencers for Internal Combustion Engines." Journal of Sound and Vibration, Vol. 1, No. 2 (1962), 185-201.

(23) Davies, P. O. A. L. "Measurement and Recording of Transient Pressures in Unsteady Flow." Proceedings of the Institution of Mechanical Engineers, Vol. 180, Part 3G (1965-66), 1-7.

(24) Davies, P. O. A. L. and M. J. Dwyer. "Communications." Proceedings of the Institution of Mechanical Engineers, Vol. 179, Part 1 (1964-65), 391-394.

(25) Davies, P. O. A. L. and M. J. Dwyer. "A Simple Theory for Pressure Pulses in Exhaust Systems." Proceedings of the Institution of Mechanical Engineers, Vol. 179, Part 1 (1964-65), 365-375.

(26) Davis, D. D., Jr., G. M. Stokes, D. Moore, and G. L. Stevens, Jr. "Theoretical and Experimental Investigation of Mufflers with Comments on Engine-Exhaust Muffler Design." NACA Report 1192, 1954.

(27) Donaldson Company, Inc. "Backpressure Measurement (Use and Limitations of the Manometer)." Acoustic Technical Bulletin No. 11, Minneapolis, Minn., March, 1966.

(28) Donaldson Company, Inc. "Effect of Exhaust System on Engine Performance." Acoustic Technical Bulletin No. 9, Minneapolis, Minn., March, 1968.

(29) Enochson, L. D. and R. K. Otnes. "Programming and Analysis for Digital Time Series Data." SVM-3, Shock and Vibration Monograph Series, Shock and Vibration Information Center, U. S. Department of Defense, 1968.

(30) Glorig, A., W. D. Ward, and J. Nixon. "Damage Risk Criteria and Noise-Induced Hearing Loss." The Archives of Otolaryngology, Vol. 74 (1961), 413-423.

(31) Glorig, A. "The Effects of Noise on Hearing." Journal of Laryngology and Otology, Vol. 75, No. 5 (1961), 447-478.

(32) Goyal, M., G. Scharf, and G. Borman. "Simulation of Single Cylinder Intake and Exhaust Systems." Society of Automotive Engineers Transactions, Vol. 76, Part 3 (1968), 1733-1747.

(33) Harris, C. M., ed. Handbook of Noise Control. New York: McGraw-Hill, 1957.

(34) Hawley, M. E. "Acoustic Interference for Noise Control." Noise Control, Vol. 2, No. 2 (1956), 61-63, 94.

(35) Hodgman, C. D., ed. C. R. C. Standard Mathematics Tables, 12th ed. Cleveland: Chemical Rubber Publishing Company, 1959.

(36) Hsu, H. P. Outline of Fourier Analysis. New York: Simon and Schuster, 1967.

(37) Huang, B. K. and Y. R. Chen. "Computer Methods in Silencer Design." American Society of Agricultural Engineers, Paper No. 69-174, June, 1969.

(38) Huang, B. K. "Digital Simulation Analysis of Biological and Physical Systems." American Society of Agricultural Engineers, Paper No. 70-515, July, 1970.

(39) Huang, B. K. and C. W. Suggs. "Tractor Noise and Operator Performance." Transactions of the American Society of Agricultural Engineers, Vol. 11, No. 1 (1968), 1-5.

(40) Hutchings, M. J. and G. H. Vasey. "Preliminary Investigations of Noise Levels Around Agricultural Tractors." Journal of Agricultural Engineering Research, Vol. 9 (1964), 140-146.

(41) Jensen, J. K. "Are Tractors Noisy?" American Society of Agricultural Engineers, Paper No. 65-608, December, 1965.

(42) Jones, H. H. and J. L. Oser. "Farm Equipment Noise Exposure Levels." American Industrial Hygiene Association Journal, Vol. 29, No. 2 (1968), 146-151.

(43) Keast, D. N. Measurements in Mechanical Dynamics. New York: McGraw-Hill, 1967.

(44) Kinsler, L. E. and A. R. Frey. Fundamentals of Acoustics. New York: John Wiley and Sons, 1962.

(45) Kistler Instrument Corporation. Instruction Manual for Quartz Pressure Transducers Model 701A and H. Redmond, Washington.

(46) Lierle, D. M. and S. N. Reger. "The Effect of Tractor Noise on the Auditory Sensitivity of Tractor Operators." Annals of Otology, Rhinology, and Laryngology, Vol. 67 (1958), 372-388.

(47) Litke, R. "Farm Noise Can Impair Your Hearing." The Farmer, May 4, 1968, 76.

(48) Lyn, W. T., A. J. Stockwell, C. H. T. Wang. "Accuracy in Cylinder Pressure Measurement." Proceedings of the Institution of Mechanical Engineers, Vol. 180, Part 3G (1965-66), 8-16.

(49) Martin, L. S. "The Need for Establishing a Maximum Noise Level Standard." American Society of Agricultural Engineers, Paper No. 69-119, June. 1969.

(50) Matthews, J. "Measurements of Environmental Noise in Agriculture." Journal of Agricultural Engineering Research, Vol. 13, No. 2 (1968), 157-167.

(51) Naval Submarine Base New London. "Sources of Noise." Technical Memorandum No. 5-67, Groton, Connecticut.

(52) Olson, H. F. "Electronic Control of Noise, Vibration, and Reverberation." Journal of the Acoustic Society of America, Vol. 28, No. 5 (1956), 966-972.

(53) Papoulis, A. The Fourier Integral and Its Applications. New York: McGraw-Hill, 1962.

(54) Peterson, A. P. G. and E. E. Gross, Jr. Handbook of Noise Measurement, 6th ed. West Concord, Massechusetts: General Radio Company, 1967.

(55) Priede, T. "Noise Due to Combustion in Reciprocating Internal Combustion Engines." Advances in Automotive Engineering -- Part 3, (1965), 93-128.

(56) Rowley, D. W. "Control of Farm Tractor Intake and Exhaust Noise." Sound and Vibration, Vol. 1, No. 3 (1967), 15-23.

(57) Ryland, D. W. and P. K. Turnquist. "The Measurement, Analysis, and Reduction of Noise Produced by an Agricultural Tractor." American Society of Agricultural Engineers, Paper No. 68-640, December, 1968.

(58) Sanders, G. J. "Silencers: Their Design and Application." Sound and Vibration, Vol. 2, No. 2 (1968), 6-13.

(59) Schmer, G. L. "Effects of Tractor Exhaust Stack Location on Noise Levels at Operator Station." (unpub. M. S. thesis, University of Nebraska, 1967).

(60) Schneider, A. J. "Microphone Orientation in the Sound Field." Sound and Vibration, Vol. 4, No. 2 (1970), 20-25.

(61) Shepherd, C. D. "Exhaust System Design -- Art or Science?" Society of Automotive Engineers, Paper No. 690006, January, 1969.

(62) Simpson, E. W., Jr. and I. L. Deshayes. "Tractors Produce Ear Damaging Noise." Journal of Environmental Health, Vol. 31, No. 4 (1969), 347-350.

(63) Smith, P. H. and J. C. Morrison. The Scientific Design of Exhaust and Intake Systems. Cambridge: Robert Bentley, 1968.

(64) Suggs, C. W. and B. K. Huang. "Tractor Cab Suspension Design and Scale Model Simulation." American Society of Agricultural Engineers, Paper No. 69-118, June, 1969.

(65) Trimmer, J. D. "Sound Waves in a Moving Medium." Journal of the Acoustic Society of America, Vol. 9 (1937) 162-164.

(66) Tweedy, R. H. "Engineering a Modern Steel Cab." American Society of Agricultural Engineers, Paper No. 68-111, June, 1968.

(67) USA Standards Institute. "USA Standard Procedure for the Computation of Loudness of Noise." USA Standard S3.4, New York, March 26, 1968.

(68) Walker, J. C. and E. C. Lentz. "Chambered Pipe Silencing Concept of Exhaust Systems Design." Society of Automotive Engineers, Paper No. 829A, 1964.

(69) Wang, W. M. "Matrix Formulation in Acoustical Analysis of Mechanically Driven Fluid Systems." Journal of Acoustic Society of America, Vol. 41, No. 6 (1967), 1418-1423.

(70) Weston, H. R. "A Survey of Tractor Noise and the Effects on Hearing." Journal of Australian Institute of Agricultural Science, Vol. 29, No. 1 (1963), 15-22.

(71) Woollatt, D. "The Application of Unsteady Gas-Dynamic Theories to the Exhaust System of Turbocharged Two-Stroke Engines." Transactions of the American Society of Mechanical Engineers, Vol. 88, Sec. 1 (1966), 31-39.

(72) Wright, E. H. and K. F. Gill. "Theoretical Analysis of the Exhaust System of an Oil Engine." The Engineer, Vol. 218, No. 5666 (1964), 311-319.

(73) Wright, E. H. and K. F. Gill. "Theoretical Analysis of the Unsteady Gas Flow in the Exhaust System of an Engine." Journal of Mechanical Engineering Science, Vol. 8, No. 1 (1966), 70-90.

APPENDIX A

CALIBRATION OF DIGITAL DATA

APPENDIX A

CALIBRATION OF DIGITAL DATA

The ADC output was a sequence of whole numbers between ±1023. The extreme values occurred for ADC inputs of ±10 volts. To calibrate the digital output in terms of psi of exhaust pressure, for example, the ADC numerical output must be multiplied by the factor K:

$$K = \frac{10 \text{ volts}}{1023} \times \frac{1}{A \times R \times D} \times S$$

where: A = the voltage gain of the transducer amplifier.

D = the voltage gain of the ADC input amplifier.

R = the voltage gain of the analog tape recorder output.

S = the pressure transducer sensitivity, psi/volt.

A comparison of an analog signal and its ADC output after calibration is shown in Figure 12. The analog signal is from the preliminary amplitude analysis (unfiltered). This signal, when passed through a 750-Hz low-pass filter and the ADC, produced the digital output. Straight line segments connect the digital samples to form the digital curve. The similarity between these two curves is an indication that the ADC operated satisfactorily.

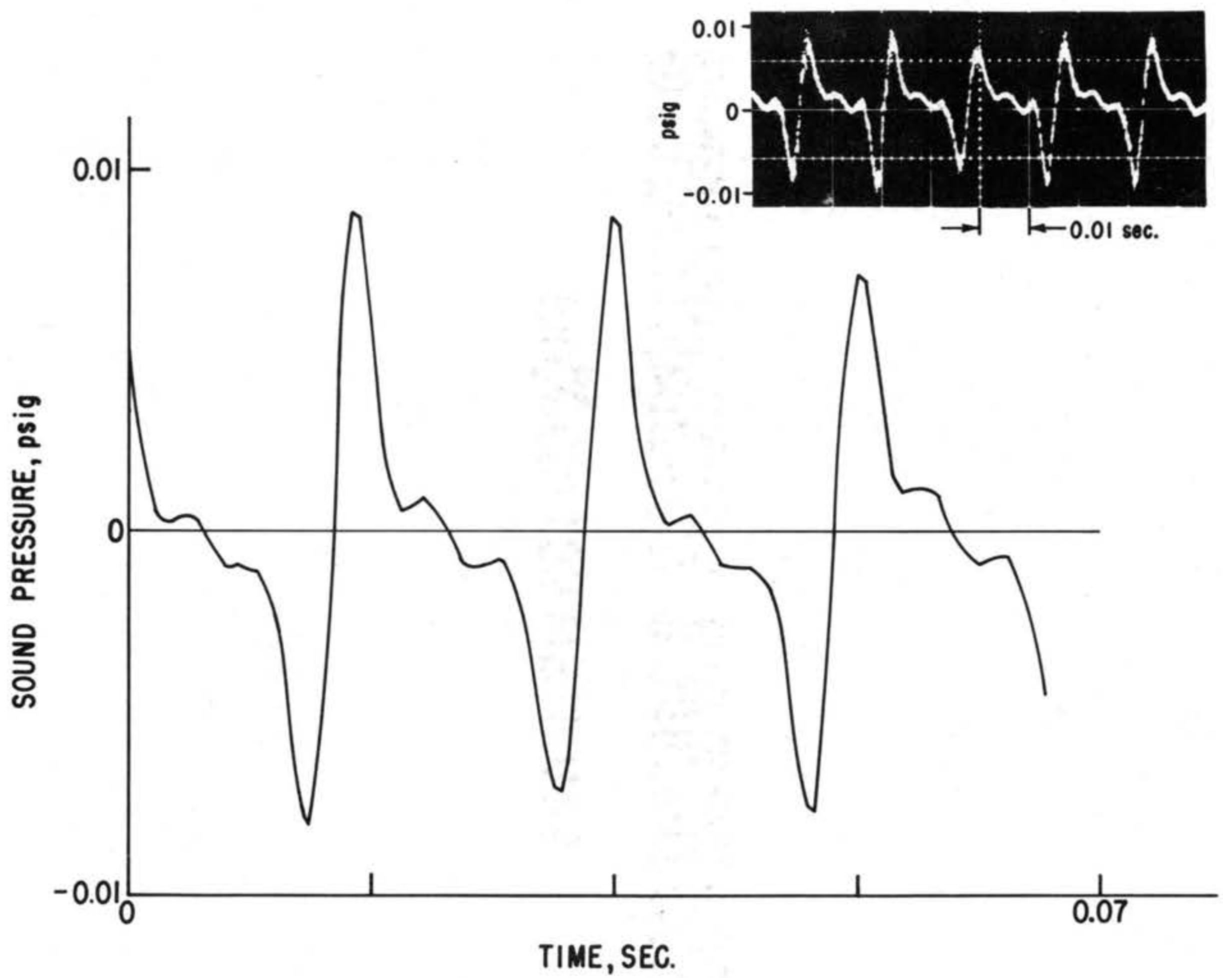

Figure 12. Comparison of Analog and Digital Representation of Test No. 1 Exhaust Noise Data

APPENDIX B

SINGLE-CYLINDER STUDY

APPENDIX B

SINGLE-CYLINDER STUDY

The single-cylinder exhaust system, shown in Figure 9, was used to experimentally evaluate the speed of sound in the exhaust system and to observe the effect of pipe length on the exhaust pressure wave. In addition, the effects of speed and load on waves in a specific system were recorded, see Figure 13. The first TDC position of the piston which occurred after the exhaust valve opened was used as a reference for layout of each group of oscillocope pictures in the following figures. This reference serves as an aid in comparing the pressure waves.

From Figure 13 it is apparent that increases in both speed and load cause increases in wave amplitude and the presence of higher frequencies. Of course, as speed increases the fundamental period decreases (firing frequency increases).

Figures 14 and 15 show interesting effects which are caused by varying exhaust pipe length for constant load and speed (rated speed with load). In Figure 14, the pressure tap was located 6 inches from the engine block. As the pipe was lenthened beyond this location, the dynamic pressure amplitudes increased (average pressure did too) and the rate of amplitude fluctuation decreased. The latter effect occurred because the train of reflections[1] from the initial compressive wave

---

[1]At the open end of the pipe the reflected wave has sign opposite to the incident wave, while at the engine (closed) end the reflected

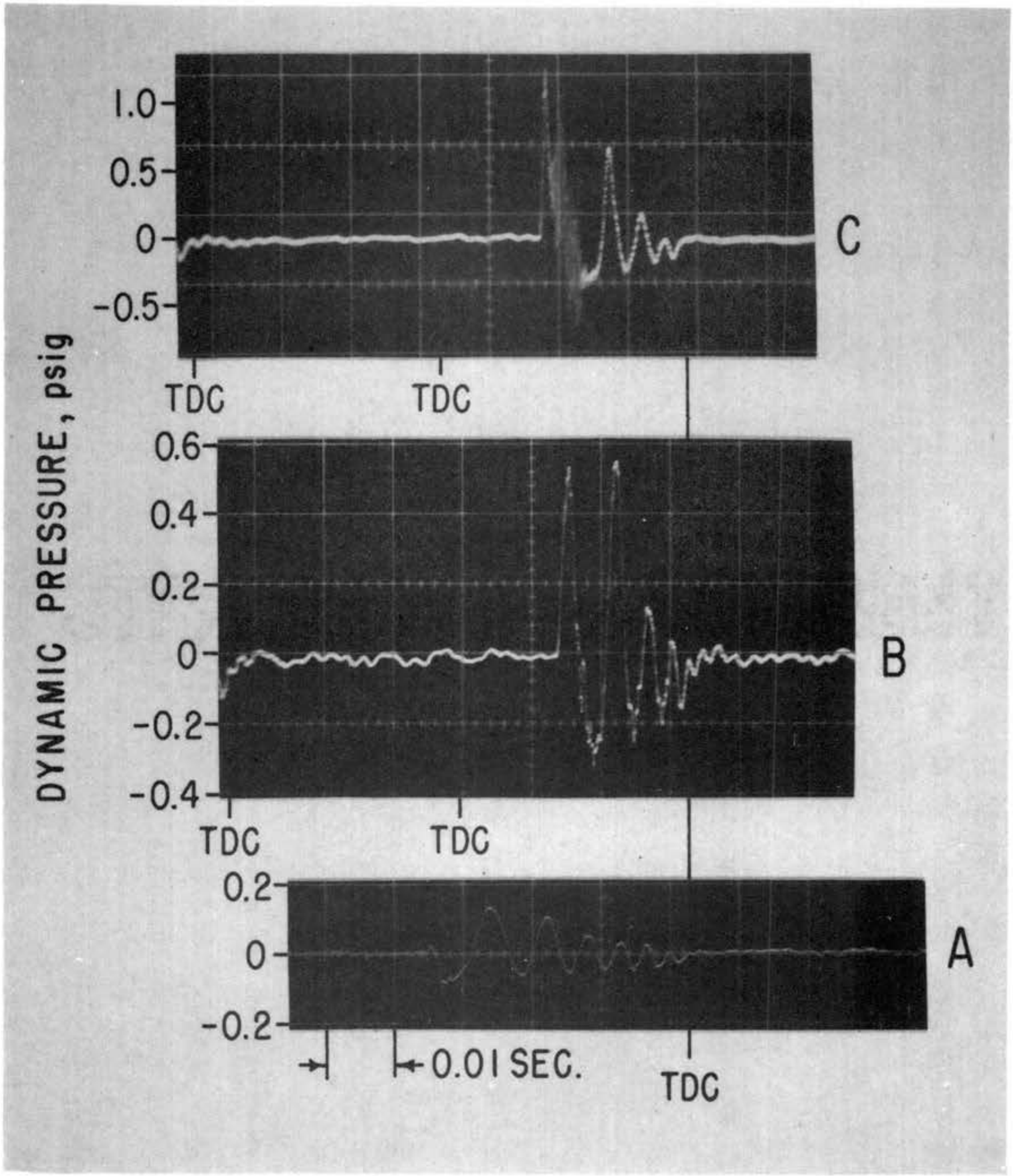

Figure 13. Pressure 4 inches from End of 8 3/4-inch Single-Cylinder Exhaust Pipe as a Function of Speed and Load. A - Idle, No Load; B - 1760 rpm, No load; C - 1668 rpm, PTO Load

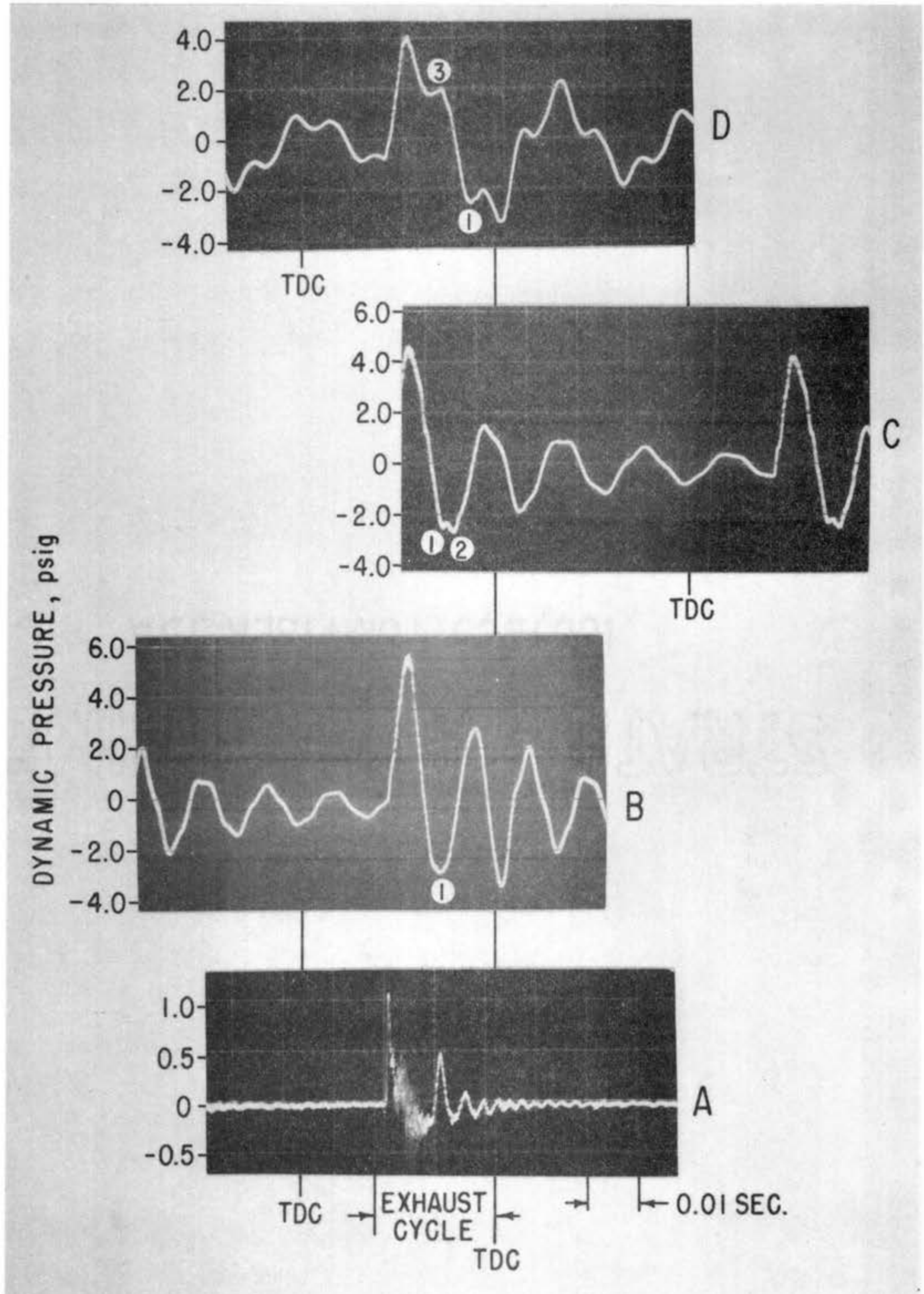

Figure 14. Pressure 6 inches from Block in Single-Cylinder Exhaust Pipe. A - 8-inch Pipe; B - 44-inch Pipe; C - 62-inch Pipe; D - 98-inch Pipe

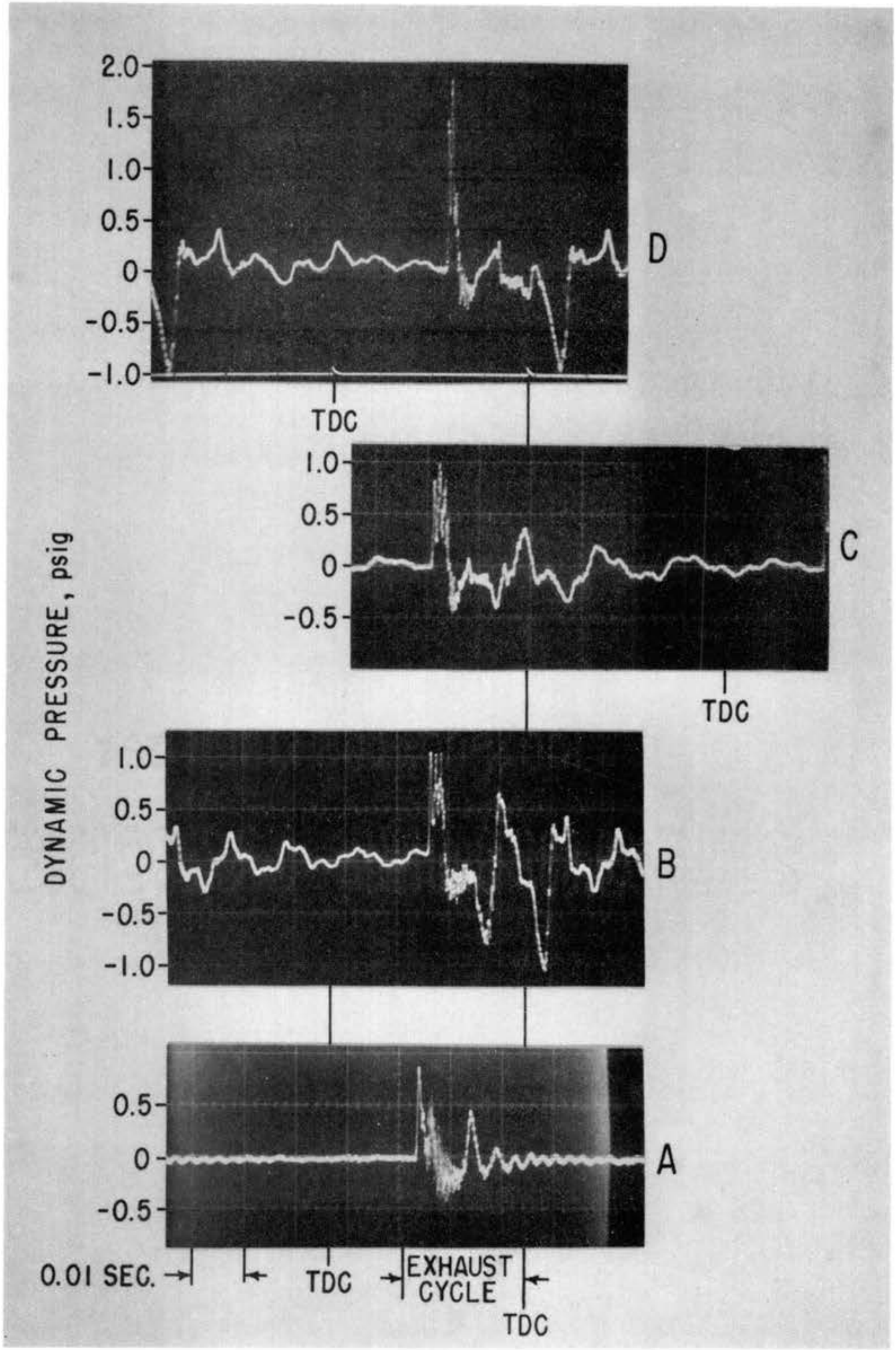

Figure 15. Pressure 2 inches from Open End in Single-Cylinder Exhaust Pipe. A - 8-inch Pipe; B - 44-inch Pipe; C - 62-inch Pipe; D - 98-inch Pipe

had longer distances to travel before passing the pressure transducer.

Figure 14 - A has high frequency "noise" after the initial pressure pulse. Possible explanations are: resonance of the pressure transducer, wave interference, and turbulence at the pipe outlet. Transducer resonance is probably at least partially the cause since the 0.055-inch diameter x 0.125-inch passage between the water-cooled adaptor and the transducer diaphragm will resonate at frequencies above 20000 Hz (45).

The number 1 identifies the first reflection at the open end. In Figure 14 - C, the first reflection at the cylinder end is indicated by the number 2. This reflection is obscured in B and D. The secondary positive pressure pulse in D, 3, may result from: (a) the piston pushing combustion residuals from the cylinder against the higher average pressure of the long pipe, (b) formation of a shock front, or (c) partial closed-end reflections caused by temperature discontinuities between successive cylinder discharges (73).

The speed of sound was determined by measuring the time between the initial pressure pulse and the return of the first open-end reflection and dividing it into the distance traveled. Inclusion of two directions of wave travel eliminated error due to exhaust gas velocity.

Figure 15 also showed an increase in pressure amplitude for increases in pipe length. However, the amplitudes of the reflections in B were larger than for C. Perhaps this was caused by the cylinder firing frequency exciting resonant frequencies of the pipe. But the

---

wave has the same sign as the incident wave. The initial wave experiences a series of reflections with gradual loss in amplitude until damped out or reinforced by another wave from the next engine exhaust cycle.

ratios of the fundamental resonant frequency[2] of each pipe and the cylinder firing frequency were 5.89, 5.03, and 2.92 for pipes B, C, and D, respectively. These ratios tended to disprove the above theory since they indicated forced vibration at resonance for system C rather than for B.

---

[2]Resonant frequencies were calculated from equation (2-1) using the measured speed of sound for the given pipe length. The cylinder firing frequency was 12.8 Hz.

VITA

Gerald Albert Stangl

Candidate for the Degree of

Doctor of Philosophy

Thesis: SIMULATED AND EXPERIMENTAL ATTENUATION OF TRACTOR EXHAUST NOISE

Major Field: Agricultural Engineering

Biographical:

Personal Data: Born in Hoven, South Dakota, July 21, 1944, the son of Mr. and Mrs. Albert J. Stangl.

Education: Graduated from Bowdle High School, Bowdle, South Dakota in 1962 as class valedictorian; received a Bachelor of Science degree in Agricultural Engineering in 1966 from South Dakota State University; received a Master of Science degree in Agricultural Engineering from South Dakota State University in 1968, as a National Science Foundation Graduate Trainee; completed the requirements for the Doctor of Philosophy degree at Oklahoma State University in July, 1971.

Professional Experience: Laboratory Assistant for the Agricultural Engineering Department, South Dakota State University, as an undergraduate; Engineering Trainee in Reliability Analysis Department, John Deere Waterloo Tractor Works, Waterloo, Iowa, summer 1965; Graduate Research Assistant and Teaching Assistant, Oklahoma State University, 1968 to 1971.

Professional and Honorary Societies: Student Member of American Society of Agricultural Engineers, Associate Member of Sigma Xi, Member of the honor societies: Alpha Epsilon, Blue Key Gamma Sigma Delta, Phi Kappa Phi, and Sigma Tau.