1

**PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT**

PRECISIONSM RESOLUTION TARGETS

PIONEERS IN METHYLENE BLUE TESTING SINCE 1974

1600 COUNTRY ROAD 6, BURNHAMME, NH 03027 USA
TEL 603 426 7887 FAX 603 426 7887 TLR 8 0000000

National Library
of Canada

Bibliothèque nationale
du Canada

Canadian Theses Service Service des thèses canadiennes

Ottawa, Canada
K1A 0N4

NOTICE

The quality of this microform is heavily dependent upon the quality of the original thesis submitted for microfilming. Every effort has been made to ensure the highest quality of reproduction possible.

If pages are missing, contact the university which granted the degree.

Some pages may have indistinct print especially if the original pages were typed with a poor typewriter ribbon or if the university sent us an inferior photocopy.

Reproduction in full or in part of this microform is governed by the Canadian Copyright Act, R.S.C. 1970, c. C-30, and subsequent amendments.

AVIS

La qualité de cette microforme dépend grandement de la qualité de la thèse soumise au microfilmage. Nous avons tout fait pour assurer une qualité supérieure de reproduction.

S'il manque des pages, veuillez communiquer avec l'université qui a conféré le grade.

La qualité d'impression de certaines pages peut laisser à désirer, surtout si les pages originales ont été dactylographiées à l'aide d'un ruban usé ou si l'université nous a fait parvenir une photocopie de qualité inférieure.

La reproduction, même partielle, de cette microforme est soumise à la Loi canadienne sur le droit d'auteur, SRC 1970, c. C-30, et ses amendements subséquents.

**AN EXPERIMENTAL STUDY OF
CO-AXIAL TAILPIPE MUFFLERS FOR
INTERNAL COMBUSTION ENGINES**

by

Alexander P. C. K. S. Lorimer, B.Eng.

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of
the requirements for the degree of

Master of Engineering

Department of Mechanical and Aerospace Engineering

Carleton University

Ottawa, Ontario

December 02, 1991

© copyright

1992, Alexander P. C. K. S. Lorimer

National Library
of Canada

Bibliothèque nationale
du Canada

Canadian Theses Service Service des thèses canadiennes

Ottawa, Canada
K1A 0N4

The author has granted an irrevocable non-exclusive licence allowing the National Library of Canada to reproduce, loan, distribute or sell copies of his/her thesis by any means and in any form or format, making this thesis available to interested persons.

The author retains ownership of the copyright in his/her thesis. Neither the thesis nor substantial extracts from it may be printed or otherwise reproduced without his/her permission.

L'auteur a accordé une licence irrévocable et non exclusive permettant à la Bibliothèque nationale du Canada de reproduire, prêter, distribuer ou vendre des copies de sa thèse de quelque manière et sous quelque forme que ce soit pour mettre des exemplaires de cette thèse à la disposition des personnes intéressées.

L'auteur conserve la propriété du droit d'auteur qui protège sa thèse. Ni la thèse ni des extraits substantiels de celle-ci ne doivent être imprimés ou autrement reproduits sans son autorisation.

ISBN 0-315-76014-1

Canada

Carleton University
Ottawa, Canada K1S 5B6

Books relating to the
history of the city of
Ottawa, which may
be of interest to the
University of Ottawa, are
available upon request
from Carleton University Library.

The undersigned recommend to
the Faculty of Graduate Studies and Research
acceptance of the thesis

**An Experimental Study of
Co-Axial Tailpipe Mufflers for
Internal Combustion Engines**

submitted by Alexander P. C. K. S. Lorimer, B.Eng.
in partial fulfilment of the requirements for
the degree of Master of Engineering

Thesis Supervisor

Chair, Department of Mechanical and Aerospace Engineering

Carleton University

April 15, 1992

ABSTRACT

The design and construction of an apparatus for the investigation of the acoustical effects of radial temperature gradients on the performance of internal combustion engine exhaust elements is presented. An exhaust system with variable elements and a flanged termination into anechoic space is acoustically excited in the frequency range of 500 Hz to 5000 Hz in the presence of a radially decreasing temperature gradient with no bulk gas flow. Sound pressures are recorded in the anechoic space, and insertion losses determined. The acoustic pressures in the outer tube of co-axial tube resonator tailpipes are mapped under uniform temperature conditions. The performance evaluation of co-axial tube resonators indicates that radial temperature gradients have a significant effect on the performance of some acoustical exhaust elements; up to approximately 20 dB more insertion loss at some frequencies with temperature gradients present compared to a uniform temperature condition for a two-hole co-axial tube resonator.

ACKNOWLEDGEMENTS

Thanks to:

**Acoustics and Signal Processing Section, Institute for Microstructural Sciences,
National Research Council of Canada for providing the opportunity for me to
work in their laboratories and for providing the facilities and materials to construct
the apparatus.**

**Dr. L. T. Russell of the Technical University of Nova Scotia for initial
encouragement to pursue graduate work in acoustics at Carleton University.**

National Research Council technician, R. St-Denis for construction assistance.

**S. E. Keith, PhD candidate at Carleton University (whose thesis work is being
carried out in the same NRC laboratory) for invaluable technical assistance at
both NRC & C.U.**

**Co-Supervisors, Dr. A.J. Brammer, NRC research scientist, and Dr. W.G. Richarz,
Professor in the Mechanical and Aerospace Engineering Department of Carleton
University for their technical assistance and steering.**

Anne Lorimer, my wife. for inspiring me to complete this work.

Table of Contents

Abstract	iii
Acknowledgements	iv
Table of Contents	v
List of Illustrations	vii
List of Plates	ix
Nomenclature	x
 I Introduction	 1
 II Muffler Elements: Definition and Performance	 5
2.1 Introduction	5
2.2 Uniform Tube	6
2.3 Acoustic Radiation from Tube Termination: Radiation Impedance	7
2.4 Reflection Coefficient at Tube Termination	8
2.5 Lumped Inertance of a Uniform Tube	9
2.6 Lumped Compliance of a Volume	10
2.7 Resonance	11
2.8 Co-Axial Tube Resonator with Wave Effects	12
2.9 Acoustic Filters	14
2.10 Performance Descriptors	17
 III System Design and Apparatus	 20
3.0 Introduction	20
3.1 System Concept	21
3.1.1 Exhaust Elements	21
3.1.2 Simulation of Exhaust Conditions	24
3.1.3 Acoustic Termination	26
3.2 Measurements	27
3.2.1 "Free-field" Sound Pressure	27
3.2.2 Sound Pressure Mapping	27
3.2.3 Temperature Measurement	28
3.3 Apparatus	29
3.3.1 High-power Electro-acoustic Driver	31
3.3.2 Acoustical Source Simulation	32
3.3.3 Exhaust System	33
3.3.3.1 Expansion Chamber	36
3.3.3.2 Co-axial Tailpipe	36
3.3.4 Anechoic Half-space Termination	42
3.3.5 Probe Tube Microphones	50
3.3.6 Thermal Simulation	54
3.3.7 Computer Data Acquisition	57

Table of Contents

continued

IV	Performance of Apparatus and Experimental Procedure	60
4.1	Performance of the Apparatus	60
4.1.1	Background Noise in the Anechoic Chamber	60
4.1.2	Standing Waves in the Transformer	62
4.1.3	Effect of Flanking Paths on Maximum Insertion Loss	64
4.1.4	Performance of Anechoic Chamber	67
4.1.5	Performance of Heating System	69
4.1.6	Performance of the Probe Microphones	72
4.2	Experimental Procedure	74
4.2.1	Insertion Loss Measurements	74
4.2.1.1	Uniform Temperature Insertion Loss Measurements	75
4.2.1.2	Radially Heated Insertion Loss Measurements	75
4.2.2	Mapping of Pressure Field Within Outer Co-axial Tailpipe	76
V	Results and Discussion	78
5.0	Introduction	78
5.1	Insertion Loss Measurements	78
5.1.1	Simple Exhaust Reference Tube	78
5.1.2	Cavity and Simple Tube Tailpipe	82
5.1.3	Cavity and Mid-Hole Co-axial Tube Tailpipe	88
5.1.4	Cavity with Top and Bottom Hole Co-axial Tube Tailpipe	89
5.2	Sound Pressure Mapping	90
5.3	Interpretation of Results	96
VI	Conclusion	97
Appendix A: Construction of the Anechoic Half-space with Tailpipe Termination		99
Appendix B: Design Calculations		110
Appendix C: Computer Programs: Data Acquisition, Processing, and Calculations		131
Appendix D: Exhaust System Performance Data		155
Appendix E: Apparatus: Performance Data and Sketches		175
References		200

List of Illustrations

Figure 2.1	Co-axial Tube Resonator	13
Figure 2.2	Acoustic Filter	
	a) Schematic Diagram	15
	b) Equivalent Circuit Diagram	15
Figure 3.1	Exhaust Elements of Equal Length	23
Figure 3.2	Acoustic Elements	30
Figure 3.3	Inner Co-axial Tube Holes	
	a) No-Hole	37
	b) Middle-Hole	37
	c) Top or Bottom Hole	38
	d) Top and Bottom Hole	38
Figure 3.4	Signal Information Flow Chart for Probe Tube Measurements .	59
Figure 4.1	Background Noise	61
Figure 4.2	Damping in the Transformer	63
Figure 4.3	Blocked Transformer Performance compared to	
	Background Noise	66
Figure 4.4	Maximum Measurable Insertion Loss	66
Figure 4.5	Impulse Response of Anechoic Chamber	68
Figure 4.6	Temperature Response of Middle Hole Co-axial Tailpipe	71
Figure 4.7	Temperature Response of Top and Bottom Hole	
	Co-axial Tailpipe	71
Figure 4.8	Probe Microphone Response of Middle Hole Co-axial Tailpipe	
	a) Amplitude	73
	b) Phase	73
Figure 5.1	Theoretical Insertion Loss of a Simple Tube Filter	80
Figure 5.2	Acoustic Power Spectral Density of Simple Tube Exhaust	
	a) At Room Temperature	81
	b) At 264° C	81
Figure 5.3	Insertion Losses of Cavity and Tailpipe Exhaust Systems	
	at Room Temperature	84
Figure 5.4	Insertion Losses of Cavity and Tailpipe Exhaust Systems	
	at 230° C	85
Figure 5.5	Insertion Losses of Cavity and Tailpipe Exhaust Systems	
	at 265° C	86
Figure 5.6	Comparison of Calculated versus Measured Insertion Loss	
	at Room Temperature	
	a) Cavity Muffler with Simple Tailpipe	87
	b) Cavity Muffler with Mid-hole Co-axial Tailpipe	87
Figure 5.7	Probe Microphone Response of Middle Hole Co-axial Tailpipe	
	at 700 Hz	
	a) Amplitude	92
	b) Phase	92

List of Illustrations

continued

Figure 5.8	Probe Microphone Response of Top and Bottom Hole Co-axial Tailpipe at 700 Hz	
	a) Amplitude	93
	b) Phase	93
Figure 5.9	Probe Microphone Response of Middle Hole Co-axial Tailpipe at 2.6 kHz	
	a) Amplitude	94
	b) Phase	94
Figure 5.10	Probe Microphone Response of Top and Bottom Hole Co-axial Tailpipe at 2.6 kHz	
	a) Amplitude	95
	b) Phase	95

List of Plates

3.1	Engine in Exhaust Test Cell	22
3.2	Tailpipe Termination in Test Cell Wall	22
3.3	a) Simple Tube Exhaust Apparatus	35
	b) Muffler and Tailpipe Exhaust Apparatus	35
3.4	Co-axial Tube Coupling Holes in Sliding Sleeves	41
3.5	Acoustic Apparatus from Driver to Tailpipe	41
3.6	Large-scale Anechoic Chamber	43
3.7	Anechoic Chamber from Outside	49
3.8	Anechoic Chamber from Inside	49
3.9	Co-axial Tailpipe with Probe Microphone Tubes and Taps	53

NOMENCLATURE

<i>a</i>	acceleration of air mass, <i>m</i>
<i>C</i>	arbitrary constants
<i>c</i>	local sound speed
<i>d</i>	diameter of hole or tubular element
<i>e</i>	natural exponential base, 2.718281...
<i>F</i>	force of moving air mass, <i>m</i>
<i>IL</i>	insertion loss
<i>J₁</i>	1 st order Bessel function of the 1 st kind
<i>j</i>	imaginary unit vector, $(-1)^{1/2}$
<i>k</i>	wave number
<i>l</i>	axial length of an acoustic element
<i>l'</i>	axial distance from the tailpipe measured acoustic pressure and its termination
<i>M</i>	2 x 2 matrix
<i>m</i>	mass of air moving under force, <i>F</i>
<i>p</i>	acoustic pressure
<i>R</i>	complex reflection co-efficient
<i>R</i>	specific acoustic resistance of an exhaust element
<i>r</i>	radial cylindrical co-ordinate
<i>S</i>	area of cross-section normal to axis, <i>z</i>
<i>T</i>	instantaneous temperature
<i>TL</i>	transmission loss
<i>t</i>	time variable, thickness of material
<i>u</i>	acoustic particle velocity
<i>V</i>	volume of cavity
<i>v</i>	acoustic mass velocity ($v = u \times \rho \times S$)
<i>X</i>	specific acoustic reactance of an exhaust element
<i>Y</i>	characteristic impedance of waves in a tube

NOMENCLATURE

continued

Z	specific acoustic impedance of an exhaust element
z	axial Cartesian co-ordinate
β	pre-exponential temperature constant
γ	ratio of specific heats of gaseous medium
ζ	specific acoustic impedance
θ	steady state temperature
ρ	density of gaseous medium
τ	thermal time constant
ψ	phase angle of reflection co-efficient
ω	radial frequency

Subscripts

<i>cavity</i>	a volume whose boundary area is very large compared to its openings
<i>co-ax cavity</i>	annular outer cavity of a co-axial tube exhaust element
<i>compliance</i>	exhaust element whose lumped impedance is analogous to a mass of air whose displacement is opposed by the compression of a volume of air
<i>hole</i>	communication port between the main tube and the annular cavity of a co-axial tube exhaust element
<i>inertance</i>	exhaust element whose lumped impedance approaches that of a small hole in a flat plate
<i>n</i>	numerical index of exhaust elements
<i>o</i>	referring to ambient
<i>rad</i>	referring to radiation into atmosphere from exhaust termination
<i>src</i>	referring to the acoustic source of a system
<i>wall</i>	boundary between the main tube and annular cavity of a co-axial tube exhaust element

I Introduction

During engine operation, the pressure in the cylinder as the exhaust port or valve opens serves to drive the spent combustion gas from the cylinder and through the exhaust system to the atmosphere. It is this fluctuating gas flow, suddenly exposed to the much lower atmospheric pressure at the outlet of the exhaust system, that is responsible for much of the exhaust noise, similar to the bursting of a balloon [1,2].

In single cylinder engines, the gas discharged from the cylinder includes an acoustic signal travelling at the local sound speed in the gas, while the bulk flow of the exhaust gas travels at a much slower velocity. In multi-cylinder engine exhausts, the bulk flow approaches a steady velocity downstream of an exhaust manifold as the alternate cylinders expel their respective pulses.

Reducing the noise without greatly inhibiting the flow of exhaust gas from the system is the primary objective of typical silencers [3]. This noise problem has classically been addressed by the use of conventional mufflers. Conventional mufflers are of two types: 1) Absorptive, and 2) Reactive. Recent advances in microprocessor technology have made active noise control feasible, that is, the control of unwanted sound by electro-acoustically generating a sound in anti-phase with the original sound to minimize the combined field at some location in space. Active noise control is effective for reducing periodic noise at low frequencies. Combined with a complementary absorptive or reactive silencer, exhaust noise may be reduced greatly at most frequencies [1,4].

In an absorptive (or resistive) silencer, the gas and acoustic pulse pass through sections of the exhaust system lined with acoustically absorbing material. Typically, high frequency components of the noise are absorbed, resulting in a

noise emission characterized by low and mid frequencies (initially, the acoustic pulse contains energy at all frequencies). Unfortunately, the corrosive environment within the silencer, with hot gases flowing intermittently at high axial velocities, leads to "blowing out" the absorptive material. This results in a constantly deteriorating performance [5,6].

Acoustically reactive silencers are typified by expansion chambers, interconnecting pipes, which may be perforated, and finally, a tailpipe. The acoustic impedance experienced by an acoustic pressure wave depends on its boundaries. A change in this acoustic impedance can result in a portion of the wave being reflected back upstream. Unfortunately, one primary means of achieving impedance changes is through general cross-sectional area changes; i.e., expansion chambers. This requires that the outside dimension of the silencer be large to be effective. These systems can be effective at many frequencies, although the overall size limits low frequency performance. In addition, the use of several stages of these impedance changes, while attenuating at many frequencies, often results in high flow impedance, and thus the mechanical performance of the engine is compromised. Where overall size permits, reactive mufflers are typically effective at low and mid frequencies. In addition, some are very effective as tuned narrow band attenuators; eg. Helmholtz and other types of resonators, and need not be restrictive to flow.

Temperature gradients exist in many muffler elements. However, it does not appear that the acoustic effects of radial temperature gradients have been investigated nor exploited [7,8]. Given a temperature distribution across a duct of heated flow in a cool environment, even within a simple tube, a radially decreasing temperature gradient exists. This may account for some of the difficulties with prediction models which assume the mean temperature across the duct is also uniform. That is, whereas many models assume plane wave theory within certain wavelength limitations, if the gas is hotter in the core of the flow than at locations

radially outward, it follows that the wave velocities will be greater in the core, thus leading to departures from plane wave propagation.

Radially uniform, but axially varying temperatures have been investigated [9] in exhaust systems. The differences between lumped approximations of axial temperature and analyses of continuous axially varying temperature are subtle and not very significant according to Munjal through private communication.

Irrespective of the silencing method applied, the gas flow and remaining noise must still issue from the system via a tailpipe. The acoustic energy remaining after the noise control elements described above, in conjunction with airflow generated noise of the exhaust system, can be sufficient to excite the natural length modes of tailpipes. Therefore, an improved tailpipe element is required to silence the noise associated with the final stage of exhaust transport.

The objective of this work is to explore the performance of co-axial tailpipe muffler elements both with and without radial temperature gradients. The elements of interest are to be suitable for use

- i) by single cylinder internal combustion engines,
- and/or ii) in complement to active noise control systems.

In this thesis, a means of experimentally modelling the exhaust system is developed to explore the performance of tailpipe silencing elements. The work includes the design and construction of an apparatus for the investigation of acoustic phenomena in small engine exhaust systems in the presence of radial temperature gradients. The potential for axial temperature gradients, and for the inclusion of flow are incorporated in the design of the facility. The apparatus includes a single cylinder internal combustion engine exhaust simulator, to which

may be attached elements of exhaust systems. After qualifying the acoustical and thermal performance of the facility, the acoustical performance of a series of co-axial tailpipes is explored under various thermal conditions.

II Muffler Elements: Definition and Performance

2.1 Introduction

In classical acoustic filter theory, an analogy is drawn between the acoustical parameters and an equivalent electrical circuit. The muffler elements are treated as acoustic filters with acoustic impedances, and the complex acoustic pressure and acoustic mass velocity are analogous to complex voltage and current, respectively. This analogy allows the system analysis to be carried out using familiar electrical circuit theory [1,10].

In applying the electro-acoustic analogy, several simplifying assumptions are made to facilitate the analysis. Firstly, the medium is assumed to be stationary. This frees the analysis from flow effects, such as convection of acoustic waves downstream. Secondly, the wave propagation is assumed to be one-dimensional in space, that is, only plane waves propagate in the axial direction, z . This allows the analysis to be governed by the one-dimensional wave equation:

$$\frac{\partial^2 p}{\partial t^2} - c_o^2 \frac{\partial^2 p}{\partial z^2} = 0 \quad (1)$$

where p is the acoustic pressure.

With the electro-acoustic analogy relations between the state variables of acoustic pressure and acoustic mass velocity, the wave equation can thus be solved. The acoustic performance of the muffler elements can be calculated with the known state variables.

As with electric circuit theory, the effect of a single acoustic element in itself is not as important as the effect of an element in combination with the remainder of the system. Once the characteristic impedance of each element is known, the specific impedance of the system, as referred to a particular location, will determine the acoustic filter performance of the entire system at that point.

2.2 Uniform Tube

The most fundamental element in an acoustic filter system is a simple tube. To simplify the analysis even further, we shall limit the discussion to uniform round pipes, as are commonly used in exhaust systems. For the case of plane waves, which requires that the inequality: $kr < 1.84$ be fulfilled, (k and r are defined below), the characteristic impedance, Y_o , of the tube represents the ratio of the acoustic pressure, p , to the acoustic mass velocity, v , associated with a progressive wave. For a tube of infinite length, it can be shown that (see pages 44 - 47 [1]):

$$Y_o = \frac{c_o}{S} \quad (2)$$

where $k = \omega / c_o$ is the wave number

r is the internal radius of the tube

ω is frequency (radians per second)

c_o is the local speed of sound

S is the cross-sectional area of the tube

Solving the wave equation with arbitrary constants C_1 and C_2 , and making use of the characteristic impedance, the state variables, p and v are determined to be:

$$p(z,t) = (C_1 e^{-jkz} + C_2 e^{jkz}) e^{j\omega t} \quad (3)$$

and

$$v(z,t) = \frac{1}{Y_o} (C_1 e^{-jkz} - C_2 e^{jkz}) e^{j\omega t} \quad (4)$$

The specific acoustic impedance at a particular location of the tube, $\zeta(z)$, depends on the downstream conditions. For the case of an anechoic termination (i.e. no reflected wave), the specific acoustic impedance throughout the tube is:

$$\zeta(z = 0 \rightarrow \infty) = Y_o \quad (5)$$

For the case of a rigid termination, the specific impedance at the termination is,

$$\zeta(z = l) = \infty \quad (6)$$

and at any other point is

$$\zeta(z) = -j Y_o \cot(kz) \quad (7)$$

For all other conditions, an analysis of the downstream element impedances is required.

2.3 Acoustic Radiation from Tube Termination: Radiation Impedance

For an open-ended tube, the atmosphere imposes an impedance on the upstream system. For the special case of a flanged termination, that is, termination of a pipe in an infinite plane (also termed radiation into a half-space),

an analytical expression for the complex radiation impedance of plane waves exists [11]. The expression for radiation impedance is in the form:

$$Z_{rad} = R_{rad} + j X_{rad} \quad (8)$$

where

$$R_{rad} = Y_o \left\{ 1 - \frac{2 J_1(2kr)}{2kr} \right\} \quad (9)$$

and

$$X_{rad} = Y_o \left\{ \frac{4}{\pi} \frac{2kr}{3} - \frac{(2kr)^3}{3^2 5} + \frac{(2kr)^3}{3^2 5^2 7} - \dots \right\} \quad (10)$$

At frequencies low enough that $kr < 0.5$, these reduce to:

$$R_{rad} = Y_o \frac{(k^2 r^2)}{2} \quad (11)$$

and

$$X_{rad} = Y_o (0.85 kr) \quad (12)$$

Empirical expressions have been developed for other types of termination conditions, however, they are quite complex and not required here.

2.4 Reflection Coefficient at Tube Termination

The reflection coefficient at an open end of a tube is related to the specific acoustic impedance, Z , and the characteristic acoustic impedance, Y_o , by:

$$R = \frac{Z - Y_o}{Z + Y_o} \quad (13)$$

For the reflection from a termination into a half-space, the amplitude is reduced to approximately [1]:

$$|R| \approx 1 - 0.14 k^2 r^2 \quad (14)$$

and the phase to

$$\psi \approx \pi - \tan^{-1} (1.2 k r) \quad (15)$$

Therefore, with the half-space termination, nearly all of the acoustic energy is reflected with a nearly 180 degree phase change, except at high frequencies.

2.5 Lumped Inertance of a Uniform Tube

For a tube of length much less than the wavelength of sound, the difference in magnitude and phase between the particle velocities at its ends will be small. Thus, the mass, m , of air is said to move as a lumped mass with a uniform velocity. Applying Newton's second law of motion:

$$F = m a$$

or, since $\frac{dv}{dt} = \rho_o S l$,

$$S (p_1 - p_2) = (\rho_o S l) \frac{1}{\rho_o S} \frac{dv}{dt} \quad (16)$$

where

S is cross-sectional area of the tube

p_1 and p_2 are the upstream and downstream acoustic pressures

l is the tube length

ρ_a is the gas density

$$\frac{dv}{dt} = j \omega v = a, \text{ acceleration} \quad (17)$$

The characteristic impedance is then

$$Z_{inertance} = j \omega \frac{l}{S} \quad (18)$$

2.6 Lumped Compliance of a Volume

If the mass of air moving in a short tube, for example, the neck of a cavity, is coupled to a finite volume, V , then the volume in the cavity is reduced by the volume displacement of the lumped neck mass, ΔV . If it is assumed that this contraction occurs adiabatically [12], then:

$$p = p_o \gamma \frac{\Delta V}{V} \quad (19)$$

where γ is the ratio of specific heats

p is the instantaneous pressure

p_o is the ambient pressure

It can be shown that [1]:

$$\Delta V = \frac{-v}{\rho_o j \omega} \quad (20)$$

therefore:

$$p = \frac{\rho_o \gamma v}{V j \omega \rho_o} \quad (21)$$

and thus the acoustic impedance of the cavity is given by:

$$Z_{\text{compliance}} = \frac{c_o^2}{j \omega V} \quad (22)$$

2.7 Resonance

Each of the acoustic elements has resonant frequencies. When the incident acoustic wave is at these (natural) frequencies, the acoustic wave is transmitted through the element with little if any attenuation, and indeed may be amplified. Theoretically, these resonances occur at frequencies for which the absolute value of the imaginary component of the acoustic impedance are minima. Therefore, the theoretical resonant frequencies can be calculated by solving the expression for the imaginary component of impedance for a minimum [10]. For the uniform tube with both ends open, which characterizes a conventional tailpipe, the resonances occur where the minima occur for the impedance of the pipe and radiation impedance combined. For plane wave propagation, resonances occur at approximately harmonic frequencies whose half-wavelengths correspond to the pipe length. These resonances are thus termed half-wave resonances. For the case of a tube with one end open, and the other end rigidly closed, the

resonances occur at approximately odd multiples of the quarter wavelength corresponding to the length of tube. These are thus quarter-wave resonances.

2.8 Co-Axial Tube Resonator with Wave Effects

One acoustical filter element that is of particular interest, especially in this work, is the co-axial tube resonator with one hole along its axis. For the element in the figure, it has been shown [13] that the impedance of the resonator referenced from the main tube at the location of the hole including wave effects in the outer tube is

$$Z_{Co-Ax\ cavity} = \left\{ j \omega \frac{l_{hole}}{S_{hole}} + \frac{\omega^2}{\pi c_o} \right\} \cdot j Y_{cavity} \frac{1}{\tan k l_a + \tan k l_b} \quad (23)$$

where d_{hole} = diameter of hole

S_{hole} = cross-sectional area of hole

S_{cavity} = cross-sectional area of annular cavity

Y_{cavity} = characteristic impedance of the cavity, a_o / S_{cavity}

t_{wall} = main tube wall thickness

l_{hole} = effective hole length, $t_{wall} + 0.85 d_{hole}$

The performance of this element, amongst others, is explored in this work.

Figure 2.1: Co-axial tube resonator. Note that l_a need not equal l_b .

2.9 Acoustic Filters

The acoustic elements described in the sections 2.2 through 2.6 as well as 2.8 can be arranged in combinations to produce acoustic filters, or in this work, exhaust systems. These filters then have acoustic characteristics dependent on those of the individual elements, as well as the effect of combining them. A typical example is shown in Figure 2.2a, where tubes of radii r_1 and r_3 , and lengths l_1 and l_3 are attached at either end of a third tube of radius r_2 and length l_2 , where r_2 is greater than either r_1 or r_3 . This is a typical exhaust pipe-expansion chamber-tailpipe exhaust system.

The relations developed for the impedances of each of the basic elements of the acoustic filter are determined, and the analogous electro-acoustic circuit is developed. (See Figure 2.2b.) The specific (overall) acoustic impedance evaluated at the source can be determined for a given system. Employing the analysis techniques of electric circuits, the performance of the filter can be evaluated in terms of the radiated power and the source power.

A technique for simplifying the analysis is taken directly from electrical circuit analysis; that is, the Transfer Matrix Method, or Four-Pole Parameters. This method relates the state variables upstream and downstream of an acoustic element through a 2 x 2 matrix. The matrix is determined from the impedance relations developed in the previous sections. With reference to Figure 2.2b, the source acoustic pressure and acoustic mass velocity are related to the radiated acoustic pressure and acoustic mass velocity by a 2 x 2 matrix, i.e.:

$$\begin{bmatrix} p_{sc} \\ v_{sc} \end{bmatrix} = \begin{bmatrix} 2 \times 2 \text{ Matrix for} \\ \text{exhaust system} \end{bmatrix} \begin{bmatrix} p_{rad} \\ v_{rad} \end{bmatrix} \quad (24)$$

Figure 2.2a: Typical Acoustic Filter consisting of three elements plus source and radiation.

Figure 2.2b: Electro-acoustic Analogous Circuit for the Acoustic Filter in Figure 2.2a.

This matrix is the matrix product of the individual 2 x 2 matrices for each of the elements in the circuit. Symbolically, these are:

$$[M_{sys}] = [M_{src}][M_3][M_2][M_1][M_{rad}] \quad (25)$$

The three essential types of transfer matrix required in this analysis are [1]:

Uniform Tube:

$$[M_{tube}] = \begin{bmatrix} \cos k l & (j / Y) \sin k l \\ (j / Y) \sin k l & \cos k l \end{bmatrix} \quad (26)$$

Lumped In-Line Element (e.g. Short Tube):

$$[M_{series}] = \begin{bmatrix} 1 & Z \\ 0 & 1 \end{bmatrix} \quad (27)$$

Lumped Shunt Element (e.g. Cavity):

$$[M_{parallel}] = \begin{bmatrix} 1 & 0 \\ 1 / Z & 1 \end{bmatrix} \quad (28)$$

Approximate calculations for the Cavity Muffler - Simple Tailpipe and Cavity Muffler - Co-axial Tailpipe are found in the Appendix B and compared to the measurements in section 5.1. In those calculations, the source impedance is assumed to be infinite.

2.10 Performance Descriptors

The following are three most commonly used descriptors for the performance of acoustic filters:

- a) Insertion Loss, IL
- b) Transmission Loss, TL
- c) Level Difference, LD or Noise Reduction, NR

Insertion Loss

Insertion loss is defined as the difference between the acoustic power level radiated without the filter and that with the filter. It can be shown that the insertion loss of an acoustic filter for a homogeneous gas is [1,13]:

$$IL = 20 \log_{10} \left[\sqrt{\frac{R_{rad,1}}{R_{rad,2}}} \left| \frac{Z_{src}}{Z_{src} + Z_{rad,1}} \right| \left| \frac{v_{src}}{v_{rad,2}} \right| \right] \quad (29)$$

where the numerical subscripts 1 and 2 refer to the system without filter and with filter, respectively. Where the tailpipe termination is maintained for the with filter and without filter cases, the radiation impedances will be equal in both cases, and therefore, the insertion loss reduces to:

$$IL = 20 \log_{10} \left[\left| \frac{Z_{src}}{Z_{src} + Z_{rad,1}} \right| \left| \frac{v_{src}}{v_{rad,2}} \right| \right] \quad (30)$$

If the source impedance is sufficiently large that $Z_{src} + Z_{rad} \approx Z_{src}$ this may be further simplified to:

$$IL = 20 \log_{10} \left[\left| \frac{v_{src}}{v_{rad,2}} \right| \right] \quad (31)$$

The ratio of v_{src} to v_{rad} , velocity ratio, can be determined from the transfer matrix of the acoustic filter system (which includes the source), specifically, the second-row, second column element of the transfer matrix.

If the source impedance is high, the source acoustic mass velocity will be approximately constant, in the comparison of a series of acoustic filters, and then the only variable remaining in the insertion loss expression is v_{rad} . Therefore, since the measurable radiated acoustic pressure can be related to the radiated acoustic mass velocity by the radiation impedance, the calculated insertion loss resulting from one filter to the next can be compared to the measured insertion loss.

In this work, it is more relevant to compare the insertion loss of an acoustic filter system to that of a simple filter, a tube. Thus the effective insertion loss can be defined:

$$IL_{eff} = IL_{filter} - IL_{tube} \quad (32)$$

IL_{eff} can be determined from the above equation and the calculated values of IL_{filter} and IL_{tube} based on the transfer matrix method. Alternatively, IL_{eff} can be calculated from the measured acoustic pressures radiated from the filter and the tube, and the following relation, which presumes that the radiation impedance is constant:

$$IL_{eff} = 20 \log_{10} \left[\left| \frac{P_{tube}}{P_{filter}} \right| \right] \quad (33)$$

Transmission Loss

The second descriptor, *TL*, is independent of the source and termination¹.

It is defined as the difference between the acoustic power level incident upon the exhaust element (or holistically, the system) and that which is transmitted to the anechoic termination. This descriptor is therefore unsuited to this study, in which the radiation of noise from the tailpipe is to be included.

Level Difference

The third descriptor is the Level Difference, which is defined as the difference in sound pressure levels between two locations in the exhaust system. This definition applies without specific requirements of the source or termination. However, as with transmission loss, it requires measurement within the system, which is more difficult to accomplish than the exterior measurements of insertion loss.

¹The independence from the termination is due to the mathematical requirement of an anechoic termination.

III System Design and Apparatus

3.0 Introduction

For a single-cylinder engine, the exhaust port/valve is closed from approximately two-thirds to three-quarters of the cycle [15]. The pulse of exhaust gas, which is discharged from the cylinder with significant acoustic energy at the firing frequency and harmonics, travels to the exit of the tailpipe where various frequency components of it are reflected [1]. As it is the goal of this work to measure the performance of tailpipe systems with the purpose of attenuating the standing waves within the tailpipe, the closed port configuration is chosen. Thus, a no flow condition is modelled.

The no-flow system will have an acoustic signal input at the location of the modelled closed port. The system must readily accommodate different exhaust system elements for comparative evaluation. A radially decreasing temperature gradient from the axis of symmetry of the pipe, and a means of measuring it, are essential for the work of this thesis. The apparatus is to have the ability to measure certain aspects of acoustic performance of the exhaust systems. These will include insertion loss measurements, and a means of mapping sound pressures within parts of the exhaust system for subsequent analysis of the acoustic phenomena existing within the exhaust system. In this thesis, the part of the exhaust chosen to be mapped is the outer annular region of a two chamber tailpipe. The ability to adapt to a flow-included model has been included in the design.

3.1 System Concept

The various configurations of apparatus were established by initially considering the simulation of an engine and exhaust system with a genuine engine and exhaust system and the measurement of insertion loss in the free-field. The final design configuration was determined by simplifying the model to suit the particular objectives of this thesis.

A real engine test setup exists at the National Research Council and comprises a single cylinder engine exhausting through simple muffling elements. The tailpipe terminates in the wall of the test-cell. (See Plate 3.1) Outside the wall, the tailpipe termination (a near infinite baffle) issues into the semi-free-field of atmosphere. (See Plate 3.2)

3.1.1 Exhaust Elements

The exhaust system proper includes the transition element between the exhaust port/valve and the exhaust elements under evaluation, the exhaust pipe itself (if present), a muffler element (if present), and a tailpipe (if present). The reference condition comprises only the transition and an exhaust pipe, Figure 3.1a, while the systems to be studied comprise the muffler and one of several tailpipes, Figure 3.1b. The overall lengths of all exhaust systems are to be equal, so that variable length is not a factor in the acoustical phenomena observed. The simulated exhaust elements are to be as similar to the real exhaust elements as possible, at least until internal pressure mapping is considered.

Plate 3.1

Plate 3.2

Figure 3.1: Exhaust Elements of equal overall length.

Conceptually, the system of particular interest comprises a transition, an expansion cavity (muffler proper), and a tailpipe with a pattern of holes on the circumferential surface in communication with an otherwise sealed annular chamber around the outside of the standard tailpipe. This tailpipe is the co-axial tube resonator. The hole pattern is to be alterable to some extent, and to include a no-hole configuration for comparison to simple tailpipes. In addition, as mentioned above, the expansion chamber is to be replaceable with a straight tube section to provide the reference condition of a simple tube.

3.1.2 Simulation of Exhaust Conditions

While the use of a genuine internal combustion cylinder would provide acoustic, thermal and mass flow input in the appropriate proportions, it was rejected as a source because of the instability of the exhaust gas parameters. Internal combustion gases are complex mixtures of noxious and corrosive gases as well as having a high particulate content and high temperatures. This harsh environment contributes to the rapid deterioration of sensitive acoustic sensors. In addition, it would be difficult to regulate a combustion process to maintain a constant set of parameters (temperature, gas constants) upon which acoustical effects may depend.

Therefore, a means of establishing a heated flow with the high pressure acoustic impulses of an opening port inside an otherwise idle engine was considered. This would entail using an electric motor to drive an engine into whose intake was piped a very hot gas. In order to simulate the acoustic signal produced upon opening the exhaust port, the pressure inside the cylinder would have to be extremely high (i.e. equal to the pressure produced upon combustion). This was seen as a difficult task within a limited budget, especially in trying to keep the gas "clean".

A heated flow system was considered where the acoustic signal was delivered by an electro-pneumatic driver. Such a system would entail a system with heated compressed gas being introduced to the exhaust system via an electro-pneumatic driver. The high pressure gas then flows through the driver where it is modulated by an electro-dynamically controlled opening. This would allow the introduction of virtually any type of acoustic signal from impulse to pure tone to random noise. This promising system is characterised by high harmonic distortion as well as high cost. (The least expensive electro-pneumatic driver, filtered and heated compressed gas system extra, was priced at \$11,000 US.)

Replacing the electro-pneumatic driver in this system with an electro-dynamic driver was not considered possible, since the acoustic signal would have to be introduced into the heated gas flow and no known driver would withstand the temperature. In addition, the driver would have to be placed entirely within the pressurized system lest it overload from a high static pressure on one side.

The above three designs have one difficulty in common; a compressed gas system with a heating stage prior to being introduced to the cylinder and the resulting flow rate would require a large heat input.

In light of the last point, the requirement for flow was re-assessed. An analysis of the time histories of single cylinder engines' exhaust pressures revealed that, only in the phase of the engine cycle after the port has closed do the acoustic phenomena of concern in the tailpipe occur [14]. As a single cylinder engine only generates flow when the exhaust port is open, a closed port model requires no net flow. This observation simplified the design of the apparatus. However, the adopted design allows for the subsequent addition of flow at such time when the limitations of the flow-included designs can be overcome.

The no-flow configuration consists of a real exhaust system with a means of electrical heat generation within, and a means of introducing the acoustic signal from an electro-acoustic driver at the location of the closed port of the model. The effects of high temperature on the electro-acoustic driver must be minimized, as well as the acoustical effects of the coupling of the driver to the exhaust system. That is, the signal must be introduced, but the means must be "invisible" to the model.

The general concept of the engine and exhaust system thus established, the means for a standardized measurement system remained to be designed.

3.1.3 Acoustic Termination

In order that measurements of performance using the simulator may be compared with measurements on a real engine, the hemispherical free-field condition at the real tailpipe termination must be replicated, at least for the frequencies of interest. A common method for measuring the insertion loss of exhaust systems is to have the tailpipe terminate in the plane of a large baffle surrounded by a free field (anechoic half-space). The large baffle is also usually the means of isolating the exhaust noise from the engine casing noise. The sound pressure or intensity is then measured at a distance of, say, one metre from the opening and at 45 degrees to the axis of the tailpipe.

The large baffle termination only poses the problem of providing an adequate seal around the tailpipe to ensure isolation of the exhaust noise from the "engine" simulator noise. The anechoic half-space design initially considered acoustic ray paths. A semi-anechoic chamber that reflected the incident sound away from the baffle was considered, but would have required a shape that would be difficult to construct. In addition, the potential for future flow was to be

considered, therefore, an anechoic exhaust vent from the termination space was to be incorporated. An analysis of the frequencies expected from the exhaust system determined the size of the semi-anechoic chamber to be constructed.

3.2 Measurements

The measurements required are sound pressure at a distance of one metre from the tailpipe in a free field (at the frequencies of interest), temperature measurement at two locations within the exhaust system, as a minimal assessment of the temperature gradient, and a means of mapping phase relations of acoustic pressures within a specific portion of the exhaust system.

3.2.1 "Free-field" Sound Pressure

This measurement requires that a microphone be mounted one metre away from the tailpipe termination and at 45 degrees and that acoustic reflections be limited to those from the "infinite" baffle. The microphone output shall then be processed into a narrow band frequency spectrum to observe any differences due to temperature changes. The spectrum recorded from a tailpipe under selected thermal conditions is then subtracted from that of a reference condition, and an insertion loss thus determined.

3.2.2 Sound Pressure Mapping

For the exhaust system configurations of interest, the phase relationships of acoustic pressure waves in the outer annular cylinder, near the ends, and near the holes coupling sound between the inner and outer co-axial tubes may be important to understanding the phenomena. It is essential in this measurement that the relative phase between the signals be determined. Therefore, this requires that

all the measurements be made simultaneously, or at least in simultaneous pairs. Furthermore, the pressure measurements must be made without interfering with the acoustic field in the volume between the two co-axial cylinders. As the annulus is expected to be quite small, and microphones are typically very fragile with respect to temperature, these measurements will have to be performed via probe tubes. It is not essential that the absolute frequency response of the probe tubes be known for comparative measurements. The signals from the microphones must be processed into narrow band frequency spectra in order that harmonics, and temperature induced shifts may be observed.

3.2.3 Temperature Measurement

Measuring the temperature in at least two radially different locations is required to estimate the radial temperature gradient under which acoustic phenomena may occur. In this case, the radial temperature gradient in the tailpipe is of concern; that is, from the inner tube to the outer co-axial tube. The temperature sensing devices must be such that they do not interfere to any significant degree with the acoustic phenomena: i.e., they must be small and mounted flush with surfaces. To reduce errors in gradient estimation, their axial locations should be identical.

3.3 Apparatus

The design and construction of each element of the system is described. First the acoustic simulation hardware, followed by the thermal simulation.

The acoustic system of elements is shown in the schematic of Figure 3.2. The orientation of the apparatus results from thermal design requirements. An acoustic signal (random noise, impulse, or pure tone) is produced by the high-power electroacoustic driver (#1 in Figure 3.2) from an electrical signal. The sound waves are coupled to the exhaust system via an acoustic transformer (#2), which possesses gradually increasing acoustic impedance followed by a narrow tube. The transition element (#3) is the actual coupling point of the transformer with the exhaust system. It directs the acoustic energy into the exhaust system, which consists of either an expansion chamber (#4) followed by a tailpipe (#5), or alternately the straight-tube (#6) (reference condition). The tailpipe is either a straight tube or has a simple pattern of holes connecting to a co-axial tube surrounding the tailpipe. The acoustic signal finally reaches the tailpipe termination, where all radiated energy not impinging upon the microphone (#9) is absorbed in the anechoic half-space (#8).

The acoustical system design is presented in order of acoustical signal flow, from the acoustical signal source to the anechoic half-space tailpipe termination.

Finally, the apparatus for monitoring the sound field and establishing the temperature gradients within the exhaust system is described.

Figure 3.2: Acoustic Elements. 1) Driver, 2) Transformer, 3) Transition, 4) Muffler, 5) Tailpipe, 6) Exhaust Pipe (replaces 4 and 5), 7) Infinite Flanged Termination, 8) Anechoic Chamber, 9) Microphone.

3.3.1 High-power Electro-acoustic Driver

The electro-acoustic driver (also known as a compression driver) was selected to produce as high a sound pressure as possible in the frequency range of interest, from 500 to 5000 Hz². The highest sound power level easily available within this frequency range was 123 dB. The driver, a JBL 4551, has a power handling capacity of 100 Watts. Since the application of the driver is similar to a plane wave tube, but without anechoic termination, in fact, a high impedance, the sound pressure level at the throat of the driver *in situ* should be similar, if not greater. As the sought after phenomena are expected to occur at sound pressure levels (SPLs) below 155 dB (i.e. linear wave effects), they are in the realm of linear acoustics, and therefore, any substantial SPL in the exhaust system will suffice. Note that the higher the SPL, the larger will be the dynamic range of the measurements, and therefore an effect can be more precisely measured.

Electro-Acoustic Driver Support Hardware

The power to drive the JBL 4551 compression driver was supplied by a Bryston 2B amplifier with gain control. This particular brand of amplifier is well known in the professional audio industry for its ruggedness and ability to handle low impedances and deliver the signal with very low distortion.

Any amplifier when switched on can generate a DC transient that can damage a driver. A 10 ohm dummy load was applied to the output terminals when the amplifier was switched on, and then after allowing a second or more for

²The sound pressure amplitude that can be produced by an electro-acoustic driver before the pressure wave becomes significantly distorted is approximately 155 dB [from a conversation with engineering staff at JBL]. The sound pressure producing capability of a compression driver is an industry standard measurement performed in a plane wave tube with an anechoic termination.

the transient to dissipate, a single pole, double throw switch disconnected the 10 ohm load while connecting the driver.

Further protection of the driver from unwanted low frequency signals was required, and the JBL recommended filter is a 6 dB per octave high-pass filter, with a half-power (cross-over) point at 500 Hz. A simple 2-pole filter was designed and constructed to be used on the low voltage side of the amplifier. A frequency response plot of this filter is in Appendix E.

3.3.2 Acoustical Source Simulation

Because the acoustic impedance upstream from the exhaust system must be very high to simulate the closed port, the acoustic signal must be "injected" into the transition. A small hole will have a high impedance compared to the transition, but there will be considerable loss in amplitude through the hole at most frequencies. The loss is a function of the impedance change, which, in turn, is a function of the area of the hole and the transition, and for a 1/8" hole into a 3/4" tube, the plane wave transmission loss is estimated to be 10 dB. A similar effect of 12 dB would be expected in the area change of the 2 inch throat diameter of the driver to the 1/8" diameter "injection" hole.

To minimize this loss, an acoustic transformer or horn is used to gradually reduce the diameter from 2 inches to 1/8 inch diameter. The horn should follow a smooth curve to avoid sudden impedance changes, and hence significant reflections. Given the difficulty in manufacturing a curved sided cone (e.g. exponential), a conical horn is used with minimal loss [1,10]. A 7 inch length for the transformer results in a cone angle of approximately 8°. Standing waves in the cone were expected, and therefore, damped by placing loose fibreglass in the cone.

The transition element connects the acoustical transformer to the exhaust system. The 1/8 inch "injection" hole (which required a 1-1/8 inch length to facilitate construction) is at the top edge of the side of the 3/4 inch transition tube. (See assembly drawing in Appendix E) This introduces the acoustic signal immediately adjacent to the top wall of the transition, at the model of the closed port/valve, ensuring a pressure maximum at the wall. Since the 1/8 inch hole allows the air particles to move through it, the particle velocity may not be zero (and hence the acoustic pressure not a maximum) at certain frequencies at the point where the hole opens into the tube. This will reduce the pressure maximum from its potential.

A means of establishing zero particle velocity at the opening of the hole is to use a feedback system. This would entail measuring the particle velocity at the hole opening, and adjusting phase of the signal at the driver to achieve zero velocity. This means was investigated, but particle velocity measurement requires very small hot-wire anemometers and support circuitry, and it was beyond the scope of this project. Given the size of the "injection" hole and length of tube, significant improvement would only have occurred at frequencies higher than are of interest, that is, only high frequencies would "see" the hole and respond to it as a hole, as opposed to a high impedance.

3.3.3 Exhaust System

The dimensions of the simple exhaust system were determined from calculations designed to separate the individual resonances of the system. Tailpipe dimensions similar to those used in a muffler for a real engine were desired. The axial standing wave and simple radial resonances of the expansion chamber, the transition resonance, and the overall system resonances were designed to occur at different frequencies than those of the tailpipe resonance. (See Appendix B for

these calculation results.) The overall system resonances were calculated by employing the appropriate lumped parameter model for the complex acoustic impedance of each element, and determining frequencies at which the imaginary component becomes zero, i.e. resonance [10]. The diameter of the transition and the tailpipe were held constant at 3/4", which is used in the real engine test setup to facilitate the evaluations of muffler elements in the engine. The axial and diametric dimensions of the expansion chamber were varied but held equal to one another throughout the calculations. The temperature was also varied, albeit uniformly throughout the system, from room temperature to 1000 K. The results of this analysis produced a system of approximate dimensions. The exact dimensions were determined by considering the frequencies of the first harmonics of these resonances throughout the temperature range. The dimensions were further adjusted so that the first harmonic of the chamber did not coincide with the fundamental tailpipe resonance, and the first harmonic of the tailpipe did not coincide with the transition fundamental resonance. (Calculation results are found in Appendix B) The resulting dimensions are:

Chamber: 4 inches, length and diameter

Transition: 1 3/8 inches, length; 3/4 inches diameter

Simple Tailpipe: 3.3 inches length; 3/4 inches, diameter.

The dimensions of the tailpipe outer annular chamber are discussed in Section 3.3.3.2. See Appendix E for assembly drawings of the exhaust system and elements. Plate 3.3a shows the simple tube exhaust apparatus, and Plate 3.3b shows the expansion chamber exhaust apparatus.

Plate 3.3a

Plate 3.3b

3.3.3.1 Expansion Chamber

The axial and diametrical dimensions were identical at 4 inches. The inlet and outlet diameters are 3/4" matching both the transition and tailpipe diameters so that no net change in acoustic mass flow occurs, and the characteristic impedance of the tubular sections is the same. This reduces the likelihood of axial cross modes at different frequencies. By having all modes exist at the same frequency at room temperature, separating these modes as a result of temperature gradients is simplified. A separation of mode frequencies then indicates that significant radial temperature gradients exist, even in the cavity. The dimension of the cavity was intended to excite a lower frequency resonance than that in the tailpipe.

3.3.3.2 Co-axial Tailpipe

The tailpipe consisted of two co-axial tubes, coupled acoustically by one or two holes in the wall of the inner tube (see Figures 3.3a to 3.3d). The inner tube was connected to the expansion chamber at one end, and the termination into anechoic half-space at the other. The inner tube terminations were thus modelled as flanged at both ends at the frequencies of interest. The outer tube was closed at both ends.

The dimensions of the tailpipe outer annular chamber and the hole patterns were determined from experiments using the engine setup. These revealed that a single hole at mid-pipe (Figure 3.3b) was effective at reducing

Figure 3.3a: Co-axial Tailpipe with no holes exposed (Simple Tailpipe).

Figure 3.3b: Co-axial tailpipe with one hole centrally located along the axis. (Mid-hole Tailpipe).

Figure 3.3c: Co-axial tailpipe with one hole located at one-fifth the axial length (Top Hole Tailpipe).

Figure 3.3d: Co-axial Tailpipe with one hole located at one-fifth the axial length, and another at four-fifths (Top and Bottom Hole Tailpipe).

the tailpipe fundamental under heated conditions. A single hole at approximately $1/5$ or $4/5$ the length of tailpipe from its inlet (Figure 3.3c), had some effect on the second harmonic, but a hole at both $1/5$ and at $4/5$ the length of the tailpipe together (Figure 3.3d) resulted in a marked reduction in both the fundamental and the first harmonic. Therefore, these three hole configurations were adopted, as well as the "no-hole" or simple tube tailpipe (Figure 3.3a) reference condition.

Hole size appeared to have less significant effect on the performance, and may be flow dependent. The acoustic phenomenon may depend on coupling effectiveness from tailpipe to annular chamber and for this reason, the holes were made as large as possible. In addition, the compliance associated with the volume of the outer tube and the inertance of the holes coupling the inner to the outer tube form a Helmholtz resonator. An exploration of wave resonance effects caused by a radial temperature gradient in the co-axial tailpipe requires that the Helmholtz resonance frequency be avoided because the performance of the tailpipes on real engines may only have been due to the excitation of Helmholtz or other lumped resonances. This is most readily achieved by designing the co-axial tailpipe so that the estimated Helmholtz resonator frequencies lie significantly below the fundamental (length) wave resonance of the inner tube. The remainder of the tailpipe design required attention to the measurement system, specifically, how to change the hole configuration while heated.

With the cavity muffler, there were four configurations of tailpipe to be compared. First, the reference, or no-hole. Next were two single-hole configurations. One had a hole at the axial centre of the tailpipe, while the other had a single hole near the upstream end. (it was felt that, in an attempt to determine if the axial hole position of a single hole was effective, the upstream and downstream locations, being symmetrically located, would behave similarly.) The final configuration had two holes; one at $19/24$ inches (20.1 mm) and the other at $2\ 3/4$ inches (69.9 mm) from the open-end termination.

To change the hole configurations from one to another, the inner tube wall was made from two thin sleeves (see plate 3.4). The inside one connected to the cavity muffler and termination flange, while the outer one was split longitudinally and keyed to the outer tube's outer wall. In this way a rotation of the outer tube will result in a rotation of the outer split sleeve of the inner tube wall. A 9.5 mm (3/8 inch) slot was machined in the inner sleeve, running nearly the full length of the tailpipe. Finally, a series of 6.4 mm (1/4 inch) diameter holes were machined in the outer sleeve to accommodate the hole patterns above. Thus, as the outer tube is rotated, the outer sleeve is also rotated and the outer co-axial tube is then in communication with the inner tube with different hole configurations. Plate 3.4 shows the inner and outer sleeves as well as the outer tube.

This complex piece of apparatus required exceptional seals at each end of the tailpipe and these were provided by "Viton O-Rings", which have been proved to survive real engine conditions as cylinder-transducer seals in previous work [4]. A means of effecting repeatable rotation on the outer tube while at elevated temperature was also required. This was addressed with a lever screwed onto the outer tube, and calibrated markings on the anechoic chamber's outer surface adjacent to the lever (for identifying the configuration being measured). Plate 3.5 shows the electro-acoustic driver, the transformer with water jacket, the transition, and the expansion chamber exhaust apparatus, suspended on cables.

Plate 3.4

Plate 3.5

3.3.4 Anechoic Half-space Tailpipe Termination

The termination space serves two functions; to provide a reflection-free environment for the frequency range of interest, and to isolate the free-field measurements from external (contaminating) noise sources. The reflection-free environment was achieved by considering both ray paths of acoustic energy, and absorption of acoustic energy incident on ultimate boundaries of the space. The isolation from external noise and vibration was achieved by constructing the space boundaries from material with high transmission loss, and erecting the entire structure on vibration isolating pads.

A predominantly anechoic environment will exist if all the walls of the space are lined with highly absorbent material to a depth equal to $1/4$ the wavelength of the lowest frequency for which an anechoic environment is required [5]. Furthermore, the anechoic character is enhanced if the shape of the space is such that any reflections are reflected into the sound absorbing medium. Large-scale anechoic chambers are typically ortho-parallelepiped in overall shape, but the absorbing material is fashioned into wedge shapes. This way, incident acoustic energy is reflected deeper into the material. Also, the wedge shape exposes more absorbing material to the incident acoustic energy. The material is typically fibreglass batt insulation, with an acoustically transparent fabric holding the insulation in place. It was decided that the anechoic half-space would be modelled after typical large-scale chambers as described in [5], and as found in the chambers at the NRC [16]. Plate 3.6 shows the anechoic chamber of the Physics division of the NRC.

Plate 3.6

Chamber Orientation

The tailpipe baffle consists of a hard reflecting planar surface. Thermal requirements of the design necessitate that this surface be the ceiling of the space, and therefore, the exhaust system axis would be vertical, above the space. Once heat was introduced, any temperature gradient in the exhaust system other than the intended radial gradient, would only be axial, i.e. the radial gradient would be uniform on all sides. This is further discussed in section 3.3.6 on thermal simulation.

Chamber Size

With the shape of the chamber determined to be an ortho-parallelepiped, the next consideration was the size. The minimum frequency of interest for this project is 500 Hz, with a corresponding wavelength of approximately 50 cm at room temperature. One quarter wavelength is thus 13 cm. However, if a flow-inclusive model was to be studied, the fundamental firing frequency would be of interest. At 6000 rpm (typical of single-cylinder engines under load), the frequencies of interest may extend to below 100 Hz, with a corresponding quarter-wavelength of 2 3/4 feet. Lining all of the walls of a space with 2 3/4 feet of absorbing material results in an overall dimension of 5 1/2 feet in each direction, without accounting for space inside for measurements.

As the measurement in the anechoic half-space requires the placement of a microphone 1 metre from the tailpipe penetration, and at 45 degrees, this defines the minimum open space required to be 0.6 metre by 0.6 metre by 0.6 metre. This setup may be sufficient acoustically because of symmetry, but with the inclusion of flow at a future date, the aero-acoustic and aerodynamic effects of the

absorbing material immediately adjacent to the tailpipe may be significant³. For this reason, the space should be 1.2 metre wide, 1.2 metre long, and 0.6 metre high, minimally, with the tailpipe outlet at the centre of the top face of this newly defined ortho-parallelepiped open space. Thus, the overall size of the anechoic space will be the size of the open space, plus the depth of absorbent material used in each direction. Finally, the three dimensions should be slightly dissimilar to reduce the coincidence of standing waves. Thus, the size was determined by the space available for the apparatus to be 2 m x 2.1 m x 1.75 m (6'6" x 7'0" x 5'9").

With the overall dimensions determined, and the minimum interior space established, the remainder was available for the application of absorptive material and structural considerations. The resulting depth of this material was then 50 cm. This gives a lower limit to the semi-anechoic chamber of approximately 170 Hz. This limit was more than adequate for this project.

Chamber Placement

The construction of the chamber comprises 2" x 4" framing for the floor and walls, with an edge suspended ceiling. (In order that the apparatus may be accessed from the top, a second floor was built over the ceiling on an edge supported 2" x 6" frame.) All panels comprise a layer of 3/4" plywood, followed by 1/2" Gypsum Wallboard, and 1/4" finishing plywood. This construction has an estimated sound transmission class⁴ (STC) rating of 36, effectively isolating the inside of the chamber acoustically from the outside for the expected conditions.

³Aero-acoustic tube sources are quadrupoles [4] and thus require free space in all directions orthogonal to the axis to be fully developed. Therefore, the open space should be the same in all four quadrants.

⁴Sound Transmission Class is a rating for partitions where a normalized STC curve is fit to the transmission loss data for the partition and the value at 500 Hz is reported as the STC value.

The chamber is mounted on a grade level poured concrete floor. Acoustically induced floor vibrations are expected to be minimal, however, the potential for road traffic-induced vibration exists, which could affect the alignment of the muffler system. As a precautionary measure, some degree of isolation is required. As the lower limiting frequency of interest is 500 Hz, the isolation should be effective above 500 Hz.

Employing a conservative estimate of the chamber weight, including hardware at 2000 pounds, the isolators chosen are 60 durometer neoprene, in the shape of 2 inch squares, and a thickness of 9.5 mm. The number of these isolators is approximately 200. They were evenly distributed on the floor at the site of the chamber, and it was then constructed on top. The minimum isolation efficiency is estimated to be 80 percent at 500 Hz [17].

A door is required for access to apparatus within the chamber. The sealing of the door is accomplished via two layers of resilient weather stripping.

Mounts for the microphone and other sensing equipment is provided by 1/4 inch diameter aluminum rods secured to the framework of the walls. The rods pass through the sound absorbent material, and extend into the space at least 5 cm.

The sound absorbent material is glass fibre. The glass fibre, in the form of 48" long by 16" wide by 4" deep batts, is held in place by acoustically transparent fabric (LaCoste Doubleknit)

In addition to providing a sturdy shield for the fibreglass, the fabric is very stretchy. This allows the fabric to be drawn tightly over the fibreglass and hold it in place. To destroy the smooth flat surface resulting from stretching the cloth tightly over the fibreglass, a pattern of locations on the wall was used as anchor

points for nylon fishing line. These lines were passed through the fibreglass batts, through the supporting fabric, and drawn taught, which in turn drew the fabric in towards the wall. This has the effect of producing an undulating surface. Thus, any reflections would be scattered in the otherwise anechoic sound field. The nylon lines also served to support the stack of fibreglass batts, and keep them from settling.

The fibreglass was installed on the two side walls and the rear wall of the chamber in this way. The fibreglass on the floor required the same degree of absorbent material, but had to be removable to permit access to the chamber. Also, since there is only half as much absorption in the floor to ceiling direction (because the ceiling is bare) the attenuation of standing waves in this direction will be reduced. For this reason, it was decided that the backing of the absorbent floor inserts should form a triangular prism. The fibreglass is held in place on the prismic insert by a sheet of the double-knit fabric, stretched over the fibreglass, and stapled to the back of the support.

Finally, the installation of absorbent material on the door side of the chamber was achieved in two stages. Firstly, the door was covered with a single 4 inch layer of fibreglass batt. These were stapled to the door and then covered with the double-knit fabric. Secondly, a stack of fibreglass batts, approximately 1 metre tall by 1 metre wide and 16 inches deep, was wrapped in the double-knit fabric and sewn closed. Three lengths of fishing line were fed from the bottom to the top of the stack in a line along the width. They were secured at the bottom to the fabric and at the top they were available as lifting handles. The fishing line was passed through three screw-eyes used to suspend the entire stack. Thus, the stack may be lifted into the space immediately in front of the door within the chamber. Plate 3.7 shows the exterior of the anechoic chamber and Plate 3.8 shows the interior, looking up at the flanged termination.

A detailed description of the construction and location of the anechoic half-space tailpipe termination, including diagrams and photos is given in Appendix A.

Plate 3.7

Plate 3.8

3.3.5 Probe Tube Microphones

The sensing of pressures within the tailpipe is necessary to investigate the acoustical behaviour therein. The region of primary interest for this project is the outer annular space of the outer co-axial tailpipe tube, and therefore, a means of pressure measurement within this space is required.

The smallest easily available quality microphone has a diameter of approximately 1/4 inch. This dimension is far too large to place in the annular space. A means of coupling into the sound field at a precise location, and detecting the sound pressure by a remote microphone is required. This requirement may be satisfied by a probe microphone, which consists of a fixed length, narrow diameter tube connected to a high quality microphone.

A probe tube that traversed the length of the space was rejected partly because of the intricate plumbing required to provide a seal in the heated environment while at the same time permitting motion, but primarily because to completely determine the waveforms present within the region, a measure of the phase with respect to location is required. This entails measuring the pressure signal in at least two places within the region synchronously. The plumbing required for two traversing, or one traversing plus a fixed probe would be equally intricate.

Rather than placing a probe entirely within the narrow annular region, a series of pressure taps were drilled through the outer surface of the outer tailpipe. To each of these holes was attached a small steel tube followed by a length of flexible teflon tubing. At the other end of each length of tubing, a microphone was mounted with an airtight seal. The series of microphones is then simultaneously sampled in groups of at least two.

A probe tube microphone forms an acoustic low-pass filter. Because of the low intrinsic absorption in air, standing waves corresponding to length resonances will result in the probe tube. Such resonances can mask small changes in the pressure field being probed and also impart phase changes in the signal.

One possibility of avoiding resonances is not to produce pressure fluctuations at the probe tube resonant frequencies. Given that the maximum frequency of interest is 5000 Hz, and considering the end conditions of the probe tube (one open, the other closed), the maximum tube length (corresponding to the $1/4$ wavelength of 5000 Hz) without resonances will be less than 2 cm.

Considering the impracticality of such a short tube, the other means of reducing the resonances is to increase the absorption of sound. The damping mechanism relies on the friction between the wall and viscous fluid, i.e. air. One approach is to lengthen the walls, i.e., the tube length. There are two drawbacks to this approach; one is that the length would be several feet long, which would support many harmonics in the frequency range of interest, and secondly, much of the acoustic energy at all frequencies would be greatly diminished because the viscous loss damping varies little with frequency [5].

Frequency selective damping may, however, be employed. In a standing wave, the velocity is zero, and hence the pressure is at its maximum at the end which is blocked. Conversely, at the open end, the velocity is at its maximum, while the pressure is a minimum. Since viscous damping involves absorption of kinetic energy by reducing the fluid particle velocity, applying a damping mechanism only at the locations of maximum particle velocity will reduce the resonances, but have less effect at other frequencies.

A suitable damping mechanism is the insertion of fine strands of wires into the tube at the open end. These effectively increase the viscous damping without changing the length.

The number of probes used was dependent on the number of probes that could be sampled at one time, which was determined by the computer hardware, and the space along the tailpipe for probe microphones. Since only the first and second order plane wave modes were of interest in the co-axial tailpipe, only five locations of pressure measurement along the tailpipe were required to identify the mode; that is, near the ends of the outer tube, at the centre, and half way between the ends and centre. The computer's analogue to digital converter board was capable of sampling three channels at a time. Therefore, the upper two and centre channels could be sampled simultaneously, followed by the lower two and centre channels. In this way, a common reference of the centre location may be obtained between the two sets of measurements, enabling the amplitude and phase profiles along the entire length of the tube to be reconstructed. Plate 3.9 shows a close-up of the probe microphone tubes attached to the outer tube of the co-axial tube tailpipe. Note the burnt epoxy, which later caused failure of the adhesion.

Plate 3.9

3.3.6 Thermal Simulation

The thermal simulation desired for the current study is to establish a radially decreasing temperature gradient within the axis-symmetric exhaust system. It should be provided uniformly over the length of at least each section of the exhaust system. The precision of the axial uniformity is not critical as the speed of sound is proportional to the square-root of the absolute temperature. However, a radial difference in temperature of approximately 100 °C between the inner and outer tailpipe chambers is desired.

The internal pipe temperature of the exhaust of a real engine of similar size has been monitored [4]. The exhaust temperature near the tailpipe exit was approximately 200 °C. Thus a radial temperature gradient characteristic of that occurring in real exhaust systems may be established by heating the inside to the 200° to 300° range. The actual temperature gradient will depend on the heat transfer properties of the system.

There are two means of establishing a radially decreasing temperature gradient: one is to cool the outer surface; the other is to heat the axis of the system. Cooling from the outside would require considerable refrigeration initially, but thermal steady state (which is required for reliable measurements) would result in no temperature gradient at all, unless heat was continually introduced within the system. Since the other means of establishing a radially decreasing temperature gradient involves introducing heat within the system, cooling the outside is then redundant.

The main heat transfer medium from the combustion chamber to the exhaust system is the gas itself. Therefore, it follows that the temperature distribution across the tailpipe section should be hottest at the centre of the

tailpipe. As a heated gas flow has been excluded at this time, the most logical means of heat introduction is via a line source along the exhaust system axis. This is accomplished using a small diameter electric heating rod.

The rod comprises a highly radiative sheath under which is a highly dissipative resistive element. The rod is fed electricity via a pair of high temperature leads to which a variable voltage is applied. The rod is suspended from the top of the transition element by a thin wire hook and eye. This enables the rod to be removed from the system while the acoustic elements are changed. In order to ensure that the rod is providing a uniform heat distribution in all radial directions, it is important that the rod always be co-axial with the system axis. This is achieved at the top by the suspension location. Further down the exhaust system, however, a locator ring is required to keep the rod centred in the tailpipe. This uniformity (i.e., a small angular temperature gradient) is enhanced by the orientation of the exhaust system on the anechoic chamber. That is, the exhaust system is mounted axially vertically, so that the radial gradients will all be approximately the same, and any axial gradient will result in a higher temperature near the transition than at the termination, as with real systems.

The power rating of the heating rod was determined through a one dimensional heat transfer model, first of the simple exhaust pipe, then of the expansion chamber with a simple tailpipe. For these calculations, typical emissivities of the Nickel-Iron-Chromium sheath and brass piping were determined from tabulated values [18]. These calculations are presented in Appendix B. The temperature of the rod was assumed, and then the heat diffusion equation was iterated until the heat dissipated by the rod to the system was equal to the heat dissipated by the system to the room. The rod temperature was varied until a temperature gradient of approximately 100 °C was calculated between the average air temperature in the pipe and the pipe wall temperature. Using the thermal power resulting from the calculation, the temperatures in the expansion chamber

were also calculated. The power rating was determined to be approximately 1000 Watts per metre of system length. The total length of the system from transition to termination is 8.675 inches, or approximately 0.22 metres. Therefore, a heater rod rated at 250 Watts over a length of six to eight inches is required.

The length of rod available from the only major supplier of such rods (Chromalox) is somewhat limited for an "off-the-shelf" item. It was not desirable to have the rods extend past the tailpipe termination, or even to the tailpipe termination, since with a typical tailpipe, the axial temperature gradient decreases sharply towards the end of the tailpipe [4,9]. Therefore, six and seven inch rods with 250 Watt heating capacity were obtained. The six inch rod was used in the present work. A variac (AC voltage potentiometer) with an adequate power handling capacity was obtained. However, the heater rod power dissipation capacities were rated at 240 VAC, while the variac was only capable of delivering AC line voltage up to 120 VAC. Therefore, as power is proportional to voltage squared, applying only half of the rated voltage of the heater gives only one quarter of the rated power. A step-up transformer in the ratio of 1:2 was installed in series after the variac. Thus, the variac gives control within its capacity of 0 to 120 VAC, while the transformer then doubles that voltage before applying it to the heater, giving 0 to 240 VAC control.

Heat Protection

With a heat dissipation rate of 250 Watts, the metal work of the entire acoustic system is expected to become quite hot. While the temperature will not affect the exhaust system, the conduction of heat to the loudspeaker was a major cause for concern. The transformer section (item # 3 in Figure 3.2) was surrounded by a tube with the ends sealed, and two "tygon" tubing teats on the circumference, forming a water jacket. Cold tap water was circulated through this space. In addition, a neoprene section between the transformer and transition, for

vibration isolation, also served to provide some thermal insulation. Because of the uniformity of the thermal energy introduction to the system, and the high thermal conductivity of the system, differential thermal expansion was not expected to influence the performance of the apparatus.

3.3.7 Computer Data Acquisition

The computerized data acquisition was achieved through the use of an IBM PC-AT compatible computer with add-on hardware. The hardware included a four-channel analog-to-digital converter, a manual multiplexer, anti-aliasing filters, microphone amplifiers, general purpose interface bus (GPIB), waveform generator, digital voltmeter, and thermocouples. Programs were written in QuickBASIC version 4.5 to operate the hardware, and process the acquired data. A listing of the source code for the acquisition program, and a flow chart of the processing procedure are found in Appendix C.

The flow chart in Figure 3.5 illustrates the path of signal information flow used for the probe microphone measurements. The computer program controls the waveform generator via the (GPIB) and initiates an electrical signal. This is passed through the 500 Hz high-pass filter, input to the power amplifier, then the compression driver. The resulting acoustic pressure in the outer co-axial tube is measured by the probe-tube microphones. These signals are switched through the multiplexer into three output channels, then amplified, and then input to the analogue-to-digital converter. The controlling program reads the digital representation of the microphone voltage into random access memory and then stores it on the computer's hard disk in a file for later processing.

The computer was also used to monitor the temperatures of the inner and outer co-axial tubes during the insertion loss measurements. The thermocouple

reference junction was placed in ice water (0°C), while the measuring junctions (sheathed in 1/16" diameter rod) were secured to the inner and outer co-axial tubes at mid length. The voltages produced by these thermocouples were measured by the digital voltmeter, with the inner tube voltage measured on the rear terminals, and the outer tube voltages measured on the front terminals. The voltages so measured were then read by the computer via the GPIB.

Figure 3.4: Signal information flow chart for probe tube measurements.

IV Performance of Apparatus and Experimental Procedure

4.1 Performance of the Apparatus

The performance of the apparatus is examined through several qualifying procedures. The background noise level in the anechoic chamber is established, below which no acoustic pressure is measurable. The acoustic transformer natural modes are damped to flatten the acoustic frequency response of the source. The maximum measurable insertion loss is determined by examining acoustic flanking paths. The anechoic character of the chamber is determined, which establishes the contribution of reflections to the free-field measurements. The general performance of the probe tubes is investigated, to determine possible amplitude and phase errors. Finally, the performance of the heating system is determined.

4.1.1 Background Noise in the Anechoic Chamber

The wall construction is estimated to have a Sound Transmission Class of approximately 36 [5.6]. This single-number descriptor indicates that, in the frequency bands from 125 Hz to 4000 Hz, the wall is effective at limiting the transmission of sound through the wall. The noise environment outside the box was not loud, though the Heating, Ventilation, Air Conditioning, and water system noise in the building was noticeable. The background noise inside the box was measured to be approximately 10 dB SPL (referenced to 20 microPascals) in each 8 Hz band, from 500 Hz up to the limit of the measurement (6.4 kHz). See Figure 4.1.

Figure 4.1: Acoustic power spectral density of the background noise in the anechoic chamber.

4.1.2 Standing Waves in the Transformer

The transformer element coupling the driver to the transition section was expected to contain standing waves. Some damping of these standing waves was achieved by placing acoustically absorbent material in the transformer. The sound absorbing material had the effect of reducing the sound pressure level at all frequencies by approximately 8 to 10 dB, but reduced the magnitude of the resonant peaks by an additional 5 dB. See Figure 4.2

Figure 4.2a: Acoustic power spectral density in the anechoic chamber with nothing (upper plot) and fibreglass (lower plot) in the transformer.

Figure 4.2b: Effect of placing fibreglass in the transformer (difference between curves in Figure 4.2a.). Note the harmonically related peaks, indicating that standing waves in the transformer have been reduced.

4.1.3 Effect of Flanking Paths on Maximum Insertion Loss

To explore the alternative paths whereby sound from the acoustic driver could reach the microphone within the anechoic chamber, the acoustic path through the engine simulator and exhaust system was blocked. Neoprene sheets were inserted, first at the entrance to the transition, then additionally at the termination, while the noise level in the box was measured for each case.

The blocking of the transformer/transition interface with two layers of neoprene effectively reduces the sound transmitted into the exhaust system, and thus sound will have reached the microphone by any of the following three paths:

- A) vibrationally transmitted through the tube; a flanking path
- B) airborne transmitted from the driver via a path outside the exhaust system; a flanking path.
- C) transmitted through the blocking material; a direct, attenuated path.

The first graph in Figure 4.3 shows that the acoustic energy measured inside the box is very small with the two sheets of neoprene at the transition; approximately 5 dB at the most above the background noise. This suggests that flanking paths are not of concern. In addition, no change from the sound pressure in Figure 4.3 was observed with the further blocking of the termination, suggesting that the 5 dB of energy which had arrived at the microphone did not travel via path C.

The insertion loss introduced by the two sheets of neoprene, shown in Figure 4.4 is approximately 25 to 60 dB across the spectrum. (It may, in fact, be

60 dB across the spectrum, but the signal to noise ratio in the chamber at some of the frequencies prevents verifying this.) This insertion loss is the maximum insertion loss measurable by the apparatus at the particular source strength, which was held constant throughout the measurements.

The most probable flanking path is the vibrational path along the tube.

Figure 4.3: Signal to noise (dB) with the acoustic transformer blocked with neoprene (sound pressures above the background).

Figure 4.4: Maximum measurable insertion loss (dB), background sound pressure spectrum subtracted from the open system sound pressure spectrum.

4.1.4 Performance of Anechoic Chamber

In the calculation of insertion loss, only differences between measured sound pressures are retained, and therefore, reflections, theoretically, are not evident in the insertion loss spectra. However, it is possible that the presence of reflections within the chamber could cause interference at the microphone.

To determine whether reflections are satisfactorily reduced by the fibreglass lining of the chamber, a small speaker was placed at the location of the tailpipe termination, and a pulsed signal was introduced. The response of the microphone 1 m from the termination at 45 degrees was monitored.

The time required for the acoustic signal to travel from the speaker to the microphone was estimated to be approximately 2.8 ms. The time required for the earliest reflection to reach the microphone (from the floor) was then estimated to be 8.4 ms. Referring to Figure 4.5, there are no significant reflections evident in the 25 ms following the direct response. Indeed there are no reflections at least 30 dB below the direct response at any frequency. A very low frequency (approximately 50 Hz) signal is observed with a very low amplitude. This likely corresponds to a standing wave in the chamber. Further figures in Appendix E show that the source and the response signals are fully coherent from 500 Hz and up, and that there is significant acoustic energy in the frequency response from approximately 1000 Hz and up.

Thus, the chamber acoustical character is sufficiently anechoic for the purposes of this work.

Figure 4.5: Impulse Response of the Anechoic Chamber

4.1.5 Performance of Heating System

The temperature response of the system is dependent upon the driving power of the heater, the system configuration, and the measurement location.

Representative thermal response curves for two co-axial tailpipes heated at half power (125 Watts) are shown in Figures 4.6 and 4.7. The form of the temperature response was assumed to follow the simple exponential function [19]:

$$T = \theta + \beta e^{-t/\tau}$$

where:

- t = time (minutes)
- T = temperature at time, t ($^{\circ}\text{C}$)
- θ = steady state temperature ($^{\circ}\text{C}$)
- β = pre-exponential constant ($^{\circ}\text{C}$)
- τ = thermal time constant (minutes)

Taking the time and temperature values at three points on the curve and iterating, a solution was found for the response function. This was performed for the temperature-time plots of each of the different configurations, for both heat-up and cool-down.

The time constants for these responses ranged from approximately 6 minutes in the inner tube to 12 minutes in the outer tube, with little dependence on the tailpipe configuration.

A first order exponential function approaches to within 2 percent of its steady state value after four time constants. This means that a reasonable estimate of the steady state temperature in the system occurs after approximately

24 minutes and in the outer tube after approximately 40 minutes [20]. Therefore, acoustic measurements at nearly steady state temperature requires up to a three-quarter-hour heating lead-in time.

Figure 4.6: Cavity and Middle-Hole Tailpipe temperature response curve for inner and outer co-axial tubes.

Figure 4.7: Cavity and Top and Bottom Hole Tailpipe temperature response curve for inner and outer co-axial tubes.

4.1.6 Performance of the Probe Microphones

The pressure mapping probe microphones in the outer tube of the tailpipe are uncalibrated, therefore their absolute response is essentially unknown. However, based on the experience of the NRC, their quality is reasonably consistent from one unit to the next. To correlate the microphone signals, a pure tone at the Helmholtz resonant frequency of the outer tube was used to drive the system. Thus, at this frequency, all of the microphones experienced the same amplitude and phase of pressure wave.

The geometrically simplest configuration of tailpipe is the middle-hole. The Helmholtz frequency of the middle hole configuration was calculated to be approximately 600 Hz. The response of the five microphones at this lumped resonance should be identical. As seen in Figure 4.8a, the amplitude response is not the same for all microphones, by up to 4 dB. With the phase response, Figure 4.8b, there are some differences, but limited to less than 10 degrees. The change in phase with frequency is expected to be similar for each probe tube microphone, as they are of identical construction. Therefore, except for an error of approximately 10 degrees, the phase response of the microphones should be adequate to indicate the types of phase changes expected with wave phenomena in the tube.

Figure 4.8a: Probe Microphone Amplitude Response (relative to microphone 3)

Figure 4.8b: Probe Microphone Phase Response (relative to microphone 3).

4.2 Experimental Procedure

Once the apparatus was functional, several series of measurements were performed on various muffler configurations. The configurations comprised an exhaust system with a simple straight tube as a reference, a cavity element that is insertable into the system after the transition, and the tailpipe with the outer annular chamber and various coupling holes.

4.2.1 Insertion Loss Measurements

The sound pressure spectrum radiated from the termination of the simple tube reference exhaust was recorded. This is the practical reference for insertion loss comparisons: the improvement of an acoustic filter system over a simple pipe.

The sound pressure spectrum radiating from the tailpipe termination of the system with a cavity in the upper section of the exhaust was measured next, with the simple tube tailpipe configuration. This is the classic muffler upon which the tailpipe configurations are intended to improve. It provides the reference for exploring the performance of the tailpipes themselves; any improvements of a system having a different tailpipe over this configuration indicate quite significant potential for that tailpipe.

The free-field sound pressure spectra of each of the systems with different tailpipe hole configurations were then compared to the reference spectrum of the cavity with the simple tailpipe system to give an insertion loss. (That is, equation (33) where the reference tube is replaced by a reference filter, the cavity muffler with simple tailpipe.) These data are found in Appendix D. The variation in configurations measured have holes between the inner and outer co-axial tubes at different axial locations. The axial locations are middle-hole, top-hole, and top

and bottom hole. The performance for each tailpipe configuration⁵, referenced to the simple tube exhaust is evaluated as per equation (33), and is presented in the following chapter.

4.2.1.1 Uniform Temperature Insertion Loss Measurements

The insertion loss measurement procedure described above was performed at room temperature, to provide the data for the condition of uniform temperature (radially and axially). The temperatures measured during these experiments had a variation from 14°C to 21°C, including both inner and outer tube measurements.

4.2.1.2 Radially Heated Insertion Loss Measurements

The insertion loss measurement procedure described in 3.2.1 was performed with the axial heating element operating at two different power settings. The first condition involved 50 percent power, resulting in the nominal temperature of 230°C, with a range of 210°C to 233°C, in the inner tube. The second condition consumed 67 percent of rated power, resulting in the nominal temperature of 265°C, with a range of 257°C to 270°C, in the inner tube. The radial temperature distribution changed very little with different hole configurations, and therefore, the different hole configurations were measured during the same heating run.

The temperature monitoring of the outer tailpipe was plagued by the thermocouple adhesive failing. The product used was initially epoxy, which

⁵Note: The "top-hole" tailpipe configuration did not result in significant differences from the "no-hole" tailpipe configuration at any of the temperatures. Therefore, these results are found only in the appendix.

browned and flaked off (failure). The second adhesive was ceramic cement, which was very cohesive, but did not adhere to the metal surface well, especially with the motion of rotating the outer tube to produce the different configurations. As this was never resolved, the outer tube temperatures are less precise than those of the inner tube, which did not suffer the jarring and rotation. However, a significant temperature gradient is assured from the initial thermal tests to determine the thermal response. (See section 4.1.5)

The insertion loss results of the heated systems in some instances are peculiar in that they have sudden changes in IL. An analysis of the location of the peaks in the spectra indicate that to a certain extent, these are explained by the error in temperature. The difference in temperature between different configurations at the same nominal temperature is indicated by the misalignment of the peaks, at both high and low frequencies, in the spectra. If the temperature effect is assumed to be linear, it is conceivable that the spectra could be frequency shifted slightly so as to align the peaks. However, it does not appear that the spectral shifts are linear; that is, the misalignment of peaks is not proportional to the measured temperature difference. Therefore, the spectra will not be frequency shifted for the insertion loss comparisons. For some of the spectra in the appendix, however, frequency shifting was used, and resulted in somewhat smoother looking insertion losses at frequencies where the misalignment was corrected. It is unclear whether this approach is appropriate, as the alignment of some peaks resulted in the increased misalignment of others.

4.2.2 Mapping of Pressure Field Within Outer Co-axial Tube

The pressure mapping was performed on the different configurations of the system, at room temperature. The heat cycling during the free field measurements had made reliable temperature measurements in the outer tube

difficult because of the thermocouple dislodging, and as it was, the bottom microphone (#5) had fallen off so frequently from poor adhesion that, although in place, it was not believed to provide reliable data. It was felt that cold measurements would suffice to determine that hot measurements would be possible once the adhesion problem could be remedied.

The pressure mapping was performed using pure tones rather than broadband noise to eliminate any doubt as to the frequency of concern, to simplify phase determination, and also to ensure that the quality factor of the signal was sufficient to excite resonances. That is, a broadband noise may not excite the resonances of a system to the same extent as pure tones. The response of the outer tailpipe microphones while being excited by a broadband noise may be different at a given frequency because of the presence of harmonics, which would not occur with pure tones.

The purpose of the pressure mapping is, at least initially, to determine the response modes of the outer tailpipe. This information may then be interpreted in terms of mechanisms.

V Results and Discussion

5.0 Introduction

The insertion loss results are presented for the various tailpipe configurations under various thermal conditions. These are followed by the pressure mapping results at room temperature. Finally, the scientifically important points stemming from these results are outlined.

5.1 Insertion Loss Measurements

All spectra experienced an upward shift in frequency as temperature increased. However, all spectra, including both reference spectra, experienced a distortion in the spectral shape as a result of increased temperature. This indicates that some non-linear effect occurred, an analysis of which is beyond the scope of this work.

The various measurements at elevated temperatures were difficult to obtain at identical internal temperatures, and therefore the nominal temperatures associated with a particular measurement (i.e. room temperature or cold, 230°C and 265°C) are only approximate. Room temperature varied from 14°C to 21°C. Temperatures of 210°C to 233°C are referred to as 230°C, and temperatures of 257°C to 273°C are referred to as 265°C.

5.1.1 Simple Exhaust Reference Tube

The theoretical insertion loss of the simple tube compared to no acoustic filter is calculated from the equations of Chapter 2 and is shown in Figure 5.1a. (The equation is only valid up to approximately 3 kHz due to the inequality $kr <$

0.5) The pipe resonances at odd harmonics of 380 Hz ($\lambda/4$) are clearly evident. The sound pressure recorded 1 metre from the tube termination at 45 degrees and at room temperature is shown in Figure 5.2. The spectral shape is indicative of the frequency response of the entire acoustic system comprising the driver, the transformer/transition, the pipe, and the termination, as the electrical input signal was flat, broadband, pseudo-random noise. Some of the peaks can be identified as relating to pipe resonances (near odd harmonics of 380 Hz)⁶. This spectral shape is seen to distort significantly at elevated temperature (see Figure 5.2b).

⁶Note that even in the calculated pipe response, the harmonics are not at constant multiples of the fundamental. This is due in part to the frequency dependence of the radiation impedance.

Figure 5.1: Calculated insertion loss of the simple tube exhaust referenced to no acoustic filter, at approximately room temperature (20°C).

Figure 5.2a: Acoustic power spectral density of the simple tube exhaust at room temperature (14°C to 21°C).

Figure 5.2b: Acoustic power spectral density of the simple tube exhaust at 264°C.

5.1.2 Cavity and Simple Tube Tailpipe

The sound pressures radiated from the cavity and simple tailpipe system were measured at room temperature, 233°C, and 257°C. The spectra are found in Appendix D. The insertion losses of this system under various temperature conditions compared to the straight tube are shown in the Figures 5.3 to 5.5.

For the cold test (Figure 5.3), the IL is greater than 20 dB in the ranges of 500 Hz to 1.2 kHz, 2.4 kHz to 3.3 kHz, and 3.7 kHz to 4.2 kHz. The significant peaks are at 1 kHz, 2.6 kHz, and 3.9 kHz. A large amplification occurs at just above 1.5 kHz (17 dB).

As the temperature is increased at the axis of the cavity and tailpipe (Figures 5.4 and 5.5), the spectrum shifts to frequencies approximately 30 percent higher. In addition, the amplification is reduced to approximately 12 dB, and it is also broadened to extend over approximately 500 Hz (from 2 kHz to 2.5 kHz). The second range of significant IL has changed shape from a rapid increase followed by a slow decline to a slow increase followed by a rapid decline.

Clearly, the deficiencies of this classical muffler are in the ranges (with frequencies referenced to the cold spectrum) from approximately 1.4 kHz to 2.2 kHz, and again from 3.3 kHz to 3.7 kHz.

In the frequency range of 1.4 kHz to 2.2 kHz, the major amplifying resonances at 1.55 kHz and at 2 kHz coincide approximately with the two theoretical plane wave modes of an idealized cavity and tailpipe system, that is, the cavity and tailpipe length resonances.

The theoretical insertion loss of the cavity muffler and simple tailpipe system referenced to the simple tube at room temperature were calculated using the equations of Chapter 2. This is compared to the measured insertion loss at the same temperature, and shown in Figure 5.6a. The theory agrees well with the measurements up to approximately 1500 Hz, and the general measured trend is evident in the calculated result above 1500 Hz. All of the maxima and minima of the calculated results coincide, in frequency if not amplitude, with those of the measured result up to approximately 2200 Hz. This is in spite of the onset of non-plane wave propagation within the cavity muffler, which is not accounted for in the calculation. (Purely plane wave propagation only occurs when $kr < 1.84$, or at frequencies less than 1980 Hz in the muffler at room temperature.)

Finally, the installation of absorbent material on the door side of the chamber was achieved in two stages. Firstly, the door was covered with a single 10 cm (4") layer of fibreglass batt. These were stapled to the door and then covered with the double-knit fabric. Secondly, a stack of fibreglass batts, approximately 1 m tall by 1 m wide and 0.4 m (16") deep, was wrapped in the double-knit fabric and sewn closed. Three lengths of fishing line were fed from the bottom to the top of the stack, one at each end and another in the centre. They were secured at the bottom to the fabric and at the top they were available as lifting handles. The fishing line was passed through three screw-eyes in the ceiling of the chamber and used to suspend the entire stack. Thus, the stack may be lifted into the space immediately in front of the door within the chamber.

This final absorbent treatment, although not drawn into an undulating shape as with the walls, is a limp, highly absorbent panel, providing substantial absorption.

In this way, approximately one half of a cubic metre of open space, anechoic on five of six sides, and significantly isolated from external disturbance is obtained.

Exhaust Systems vs Straight Pipe, Cold

Figure 5.3: Insertion Losses of the Cavity and Tailpipe Systems referenced to the Simple Tube Exhaust System, at room temperature (14°C to 21°C).

Figure 5.4: Insertion Loss of the Cavity and Tailpipe systems referenced to the Simple Tube exhaust system, at the nominal temperature of 230°C.

Figure 5.5: Insertion Loss of the Cavity and Tailpipe systems referenced to the Simple Tube exhaust system, at the nominal temperature of 265°C.

Figure 5.6a: Comparison of the calculated versus the measured insertion loss of the cavity muffler and simple tailpipe at room temperature.

Figure 5.6b: Calculated versus measured insertion loss for the cavity muffler and mid-hole co-axial tailpipe at room temperature.

5.1.3 Cavity and Mid-Hole Co-axial Tube Tailpipe

The sound pressure spectrum of the cavity muffler with the mid-hole co-axial tube tailpipe was measured at room temperature, 210°C and 270°C.

The measured cold insertion loss of this system referenced to the simple pipe is shown in Figure 5.3. The significant ranges of IL are the same as for the simple tailpipe, except that the first range is broader (from approximately 500 Hz to 1.4 kHz). There is a new significant peak (approximately 10 dB of insertion loss where it was less than 0 dB with the simple tailpipe) at approximately 1.8 kHz.

In the range of general low insertion loss, 1.5 kHz to 2.3 kHz, there is an overall improvement in insertion loss of approximately 10 dB up to 2 kHz, including at the local minima at 1.55 kHz. There is a slight loss of performance from 2 kHz to 2.5 kHz, including a marginally increased amplification at 2.1 kHz. The remaining high frequency portion of the spectrum of IL components is approximately identical to that of the simple tailpipe.

Aside from the general spectral shift due to increasing temperature (compare Figures 5.3, 5.4, and 5.5), the second amplification (at 2.1 kHz cold) is slightly larger and more pronounced. Amplification is now recorded, however, in the frequency range from 2.1 kHz to 2.6 kHz at 230°C, and at the extremes of this frequency range at 265°C. The second range of significant IL has changed shape as with the simple tailpipe.

Clearly, this configuration is a general overall improvement over the simple tailpipe with, however, a temperature dependent performance (although not linear).

The theoretical insertion loss of the cavity muffler and middle-hole co-axial tailpipe system referenced to the simple tube at room temperature was calculated using the equations of Chapter 2. This is compared to the measured insertion loss at the same temperature, and shown in Figure 5.6b. As with the cavity and simple tailpipe, the theory agrees well with the measurements up to approximately 1500 Hz, and the general measured trend is evident in the calculated result above 1500 Hz. All of the maxima and minima of the calculated results coincide, in frequency if not amplitude, with those of the measured result up to approximately 2200 Hz. This is in spite of the onset of non-plane wave propagation within the cavity muffler, which is not accounted for in the calculations. (Purely plane wave propagation occurs only when $kr < 1.84$, or at frequencies less than 1980 Hz in the muffler at room temperature.)

5.1.4 Cavity with Top and Bottom Hole Co-axial Tube Tailpipe

The sound pressure spectrum of the cavity muffler with top and bottom hole co-axial tube tailpipe was recorded at room temperature, 230°C, and 260°C.

The cold insertion loss of this configuration referenced to the simple pipe (Figure 5.3) shows a significant loss in performance from approximately 900 Hz to 1.3 kHz. Above this range, this configuration shows a general overall improvement, except for a marginal loss of performance from 2 kHz to 2.5 kHz. The insertion loss is, however, less than 10 dB in the frequency range from 1.6 kHz to 2.4 kHz, and in common with the other tailpipes, at frequencies near 3.5 kHz and 4.5 kHz.

At higher temperatures (Figures 5.4 and 5.5), the IL from 900 Hz to 1.3 kHz is less than when the system is at room temperature, but still greater than 10 dB. In fact, the IL remains positive at all frequencies, though the minima

approach zero dB. The deterioration in performance by up to 10 dB from 2 kHz to 2.5 kHz is only significant at the higher temperature.

The most significant improvement over the simple tailpipe is from approximately 1.3 kHz to 2.7 kHz, the range in which the simple tailpipe configuration is least effective. The marginally significant deterioration in performance (from approximately 20 dB to 10 dB of insertion loss) in the ranges of 900 Hz to 1.3 kHz and 2 kHz to 2.5 kHz, should be considered less important in view of the significant gains in performance in the range of extremely poor performance of the simple tailpipe.

At this time, the theory for the calculation of insertion loss of the two hole model is not well developed, therefore a calculated result is not presented here. Determining such a model is beyond the scope of this work, and would require a closer agreement of the theory with the measurements for the simpler elements as a starting point.

5.2 Sound Pressure Mapping

Representative plots of amplitude versus microphone position and phase versus microphone position for various tailpipe configurations are shown in Figures 5.7 to 5.10. Complete results are to be found in the Appendix D. Note that microphone position three always has a zero response. This is because the responses are relative to microphone 3.

At low frequencies (i.e. 500 Hz to 1100 Hz), the system is characterized as a lumped element; that is, amplitude and phase response is approximately equal at all locations. Figures 5.7 and 5.8 show the recorded amplitude and phase responses of the outer tube of the mid-hole and top-and-bottom-hole co-axial tube

tailpipe systems, respectively, at low frequencies. These responses are similar for all microphones indicating that the outer chamber is responding as a lumped element (and for single hole configurations at 600 Hz as a Helmholtz resonator). At frequencies approaching 2000 Hz, the relative responses start to change dramatically, and at 2200 Hz becomes quite erratic.

Figure 5.9 shows the mapped outer tube pressure response of the mid-hole tailpipe system at 2.6 kHz. The phase response shows that the acoustic pressures at the extremes of the tube are each approximately 50 to 75 degrees lagging the centre location. This suggests two quarter-wave resonances. Except for minor variations in the amplitude response, this figure is typical for all frequencies above approximately 1.9 kHz with this configuration. (See Appendix D.)

For the top-and-bottom-hole system at frequencies from approximately 1.9 kHz to 2.2 kHz, Figure 5.9 is also typical. However, at approximately 2.5 to 2.6 kHz, the mapped pressure response of the outer tube of the top-and-bottom-hole system is shown in Figure 5.10. The phase response appears to be similar to quarter-wave resonances, however, with the upper section of the outer tailpipe (microphones 1 and 2) lagging the centre microphone by more than 100 degrees, and also lagging the lower section (microphones 4 and 5) by up to 150 degrees. Therefore, this phase relationship between the opposite ends of the outer tube suggests a half-wavelength resonance. (See Appendix D.)

Figure 5.7a: Amplitude Response of the outer tube of the mid-hole tailpipe at 700 Hz. Typical of response at frequencies below 1.1 kHz.

Figure 5.7b: Phase Response of the outer tube of the mid-hole tailpipe at 700 Hz. Typical of response at frequencies below 1.1 kHz.

Figure 5.8a: Amplitude response of the outer tube of the top-and-bottom-hole tailpipe at 700 Hz. Typical for all frequencies below 1.1 kHz.

Figure 5.8b: Phase response of the outer tube of the top-and-bottom-hole tailpipe at 700 Hz. Typical for all frequencies below 1.1 kHz.

Figure 5.9a: Amplitude response of the outer tube of the mid-hole tailpipe at 2.6 kHz. Typical for frequencies from 1.9 kHz to 2.7 kHz.

Figure 5.9b: Phase response of the outer tube of the mid-hole tailpipe at 2.6 kHz. Typical for frequencies from 1.9 kHz to 2.7 kHz.

Figure 5.10a: Amplitude response of the outer tube of the top-and-bottom-hole tailpipe at 2.6 kHz. Unique to this configuration and frequency.

Figure 5.10b: Phase response of the outer tube of the top-and-bottom-hole tailpipe at 2.6 kHz. Unique to this configuration and frequency.

5.3 Interpretation of Results

The insertion loss measurements have presented the performance of the tailpipe configurations under various thermal conditions and no flow, while the sound pressure mapping has suggested some of the acoustic mechanisms which may be responsible for that performance. The results have indicated several significant points which are presented below.

- 1) Co-axial tube tailpipe muffler systems clearly have significantly improved insertion loss over simple tailpipe systems.
- 2) Acoustical performance of cavity muffler tailpipe systems is dependent on radial temperature gradients, specifically radially decreasing gradients.
- 3) The number of coupling holes between the inner and outer co-axial tubes and their axial locations appear to influence the performance.
- 4) The additional insertion loss provided by the co-axial tailpipes, especially the top and bottom hole tailpipe, appears to be related to wave phenomena in the outer tube. These probably include half wavelength resonances of the outer co-axial tube.

VI Conclusion

The design and construction of an apparatus for the investigation of the effects of radial temperature gradients on the acoustic performance of simple exhaust systems has been presented. In addition, a preliminary analysis of the acoustic performance of a particular acoustic device has been evaluated using the apparatus.

The apparatus comprised an acoustic simulation of a single cylinder engine in the closed port/valve condition, a series of exhaust systems including a typical cavity-tailpipe system, and a cavity and co-axial tube tailpipe system with variable coupling conditions between the two tubes, an acoustic termination into anechoic half-space, and an acoustic measurement system. The exhaust systems shared an axially-mounted heating device, introducing a radial temperature gradient within the exhaust path.

Measurements of the insertion loss of the various exhaust systems, with both uniform room temperature, and radially decreasing temperatures revealed that the co-axial tube with coupling holes at approximately $1/4$ and $3/4$ of the length along the tailpipe, possessed the most promising acoustic performance of the systems tested. The co-axial tube with a single longitudinally centralized coupling hole also performed better than the classic cavity and simple tailpipe exhaust system.

Mapping of pressure amplitudes and phases in the outer co-axial tube, under uniform temperature conditions, suggest that the additional insertion loss of the co-axial tailpipes is related to wave phenomena in the tailpipe (for example, half-wave resonances).

While the understanding of the phenomena occurring under the radially decreasing temperature condition is left to further work, the internal response of the two-hole co-axial tube tailpipe at a uniform temperature indicates that it may be effective as a tailpipe resonance silencer by coupling the inner and outer co-axial tubes in anti-phase. As radially decreasing temperature gradients are characteristic of hot gas flows, in particular in co-axial exhaust elements, further experimental evaluation of simple exhaust elements under these thermal conditions is required. Detailed experimental data would allow for the development of a model for these elements for use in predicting the performance of muffler systems in which radial temperature gradients occur.

APPENDIX A: CONSTRUCTION OF THE ANECHOIC HALF-SPACE WITH TAILPIPE TERMINATION

Space for this apparatus was available on the National Research Council (NRC) premises in a large room referred to as "the ballroom" because of its immense size. So as not to interfere with other acoustical apparatus in the "ballroom," the only available space was against a wall under a steel staircase. The tallest height under the stairs which would accommodate the box shape of the chamber was approximately 2 m (6.5'), under the mid-span landing. To accommodate an approximately cubic box under the stairs, a change to the support structure of the staircase would be required.

In addition to moving the structural support, the railing of the landing was made removable, and a swing-down platform extending sideways from the landing was incorporated. This provided a suitable place from which to tend to much of the apparatus mounted on top of the chamber. A further modification was made to the stairway in the form of a permanent platform directly over the centre of the chamber, but approximately 1 m (3') higher than the landing. This provided a structure from which to support the heavy elements of the apparatus, namely the driver and the exhaust system.

Chamber Size

The general dimensions of the chamber were determined primarily from the space available under the stairs. The maximum height was thus 2 m (6.5'). For efficient use of space, and overall symmetry of the chamber, it was practical to make the three dimensions approximately equal. However, it was important to

design the interior dimensions such that they were all slightly unequal. This would prevent the formation of standing waves in all three directions at the same frequency. The minimum interior dimensions were determined by the space required by the measurement schemes used in the existing real engine apparatus. The required space is defined by a parallelepiped with a 1 metre diagonal from the centre of the top face to a lower corner.

With the overall dimensions determined, and the minimum interior space established, the remainder was available for the application of absorptive material and structural considerations. The resulting depth of absorptive material was then 0.5 m (20"). This results in a lower limit to the anechoic effect of approximately 170 Hz. This limit is more than adequate for the project.

Construction

The construction of the chamber comprises 2" x 4" wood stud framing for the floor and walls, with an edge suspended ceiling. (In order that the apparatus may be gained access from the top, a second floor was built over the ceiling on an edge supported 2" x 6" frame.) All panels comprise a layer of 3/4" plywood, followed by 1/2" Gypsum Drywall, and 1/4" finishing plywood. This construction has an estimated STC rating of 36, effectively isolating the inside of the chamber acoustically from the outside under the expected conditions.

The chamber is mounted on a grade level poured concrete floor. Acoustically induced floor vibrations are expected to be minimal, however, the potential for road traffic-induced vibration exists, which could affect the alignment of the muffler system. As a precautionary measure, some degree of isolation is required. As the lower limiting frequency of interest is 500 Hz, the isolation should be effective above 500 Hz. For non-critical applications, an isolation efficiency of 90 percent is adequate, and this requires that the natural frequency of

the isolation system be less than $1/5$ of the effective isolation frequency. A natural frequency of 100 Hz, requires that the isolators have a static deflection of approximately 0.02 millimetres. Given the unevenness of the poured floor, it was desirable to have a large number of isolators, so that the isolation is more evenly distributed. For a large number of isolators to have a small static deflection, they must be very resilient or small. However, a static deflection in excess of the design, which may occur where some isolators do not touch the floor, simply ensures a more conservative isolation.

Employing a conservative estimate of the chamber weight, including hardware at 900 kg (2000 lbs), the isolators chosen are 60 durometer neoprene, in the shape of 5 cm (2") squares, and a thickness of 9.5 mm (3/8"). The number of these isolators is approximately 200. They were evenly distributed on the floor at the site of the chamber, and it was then constructed on top of the isolators. Plate A.1 shows the isolators adhered to the bottom of the chamber, with some obviously not touching the floor. Also, the layered construction of the panels is evident.

Access Door

A door is required for access to apparatus within the chamber. The door must have an acoustical seal. In order that an average height person gain access to the interior, the door should be as large as possible, i.e. approximately 1.8 m (6') tall and 0.75 m (2.5') wide. The operation of the door must not interfere with the absorptive lining. For practicality, the door width should match the size of the interior open space width. The height of the door should provide access to the full height of the open space. In addition, it was foreseen that a removable floor lining will greatly improve access to the working space within the chamber. Therefore, the door should extend the full height of the chamber, minus framing and jambs.

The sealing of the door is accomplished via two layers of resilient weather stripping. As the entire perimeter must be completely sealed, the weather-stripping must be resilient enough that no undue force is required to compress (in parallel) the sum of two door-lengths and widths of weather stripping when closing the door.

Ducting for Future Gas Flow

The future provision for gas flow required the inclusion of exhaust ducts from the space. The placement of the ductwork and the shape of the ducts were determined by considering the shape of absorptive wedges in existing chambers. The ducts were thus triangular in cross-section to reflect incident acoustic energy into the absorptive material and reduce standing waves. One such duct extended the length of the chamber on each side. The inlets were large in area compared to the area of the ducts themselves to allow a transition from slow flow velocity within the chamber to higher flow velocity in the duct, to minimize aero-acoustic noise in the chamber. The ducts were lined with an absorptive material specifically design for high speed gas flows in ducts, a resin faced fibreglass ductliner. In addition, the ducts were fitted with partial baffles, perpendicular to flow, to eliminate line of sight.[4,5] The size of the ducts were chosen to be as wide as the absorptive material depth on the walls, with a height sufficient to transport a reasonable airflow. For a single cylinder engine of 100 cc, at 7000 rpm,, the gas flow is approximately 25 cfm. This is the approximate operating condition of the engine to be simulated by this project. It is conceivable that a multi-cylinder engine with larger cylinders would be modelled. Thus a large engine model may comprise up to 8 cylinders displacing 500 cc each operating at 6000 rpm on a 4 stroke cycle and would exhaust approximately 425 cfm. In order that no special techniques be required for the installation of the duct and liner (e.g. special adhesives and edge sealing), the flow velocities should be kept to less than 600 fpm (according to the product literature on the ductliner). The duct flow

velocity can be kept below 600 fpm with 0.066 m^2 (0.71 ft^2) in cross sectional area. Larger area reduces the velocity further, and is therefore acceptable. For ease of construction, an area of 0.047 m^2 (0.5 ft^2) for each duct was used, giving a total of 0.093 m^2 (1 ft^2). Thus, as the duct width was approximately 0.33 m (13") the duct height was 0.28 m (11"). Plates A.2 and A.3 show the internal construction of the ducting.

Conduit for Internal Cabling

The ductwork also provided a suitable conduit for the cabling that would be required for the internal microphone, and any other equipment that may require communication with the exterior. A draw-string is provided in each duct for threading cable. Plate A.4 shows the heater supply cable in the duct/conduit.

Mounts for Apparatus

Mounts for the microphone and other sensing equipment are provided by aluminum rods, 6 mm ($1/4$ ") in diameter, secured to the framework of the walls. The rods pass through the absorbent material, and extend into the space at least 5 cm. The locations of these rods (10 in total) allows for the mounting of the free-field microphone at any of the eight corners of the open space as well as against the ceiling at the edge of the open space, and midway along its length. In addition, these rods have holes drilled near the end, for possible future attachment of cables in orthogonal directions, potentially allowing the placement of a microphone at any point within the open space. Plate A.5 shows the secure mounting of these rods, prior to the installation of the sound absorbing material.

Sound Absorbing Material

The sound absorbent material is glass fibre. Loose fibreglass is the material used in the existing anechoic chamber packaged in wedge shapes with cheesecloth. Fabrication of a wedge shape is difficult and only provides benefit at the lower frequencies. Therefore, batts of fibreglass are used in the present work.

Another project of the NRC was being dismantled and contained a large amount of fibreglass batts, 1.2 m x 0.4 m x 0.1 m (48" x 16" x 4"). These were then available for use in this project. The dimensions were easily accommodated in the design in that one layer of batts on-end would fit between the studs of the framework (built at 0.4 m (16") on centres) and be flush with the studs, and then the remainder of the 0.5 m (20") depth was accomplished by stacking the 0.4 m (16") wide batts flat from the floor to the ceiling.

The fibreglass batts are held in place by two mechanisms, the first of which serves several purposes. In the existing anechoic chamber, cheesecloth holds the fibreglass in shape and place. However, given the traffic anticipated into and out of this chamber, a less flimsy material than cheesecloth is used. Upon consultation with one of the loudspeaker manufacturers, an acoustically transparent fabric used as a speaker grille-cloth, LaCoste Doubleknit, a polyester knit fabric, was selected. This choice was confirmed by the experience at NRC, where the material has been used as an acoustically transparent visual screen for blind subjective testing of loudspeakers.

In addition to providing a sturdy shield for the fibreglass, the fabric is very stretchy. This allows the fabric to be drawn tightly over the fibreglass and hold it in place. To destroy the smooth flat surface resulting from stretching the cloth tightly over the fibreglass, a pattern of locations on the wall was used as anchor points for nylon fishing line. These lines were passed through the fibreglass batts, through the supporting fabric, and drawn taught, which in turn drew the fabric in

towards the wall. This has the effect of producing an undulating surface. Thus, any reflections off the material would be scattered in the otherwise anechoic sound field. The nylon lines also served to support the stack of fibreglass batts, and keep them from settling.

The fibreglass was installed on the two side walls and the rear wall of the chamber in this way. The fibreglass on the floor required the same degree of absorbent material, but had to be removable to permit access to the chamber. Also, since there is only half as much absorption in the floor to ceiling direction (because the ceiling is bare) the attenuation of standing waves in this direction will be reduced. For this reason, it was decided that the backing of the absorbent floor inserts should form a triangular prism. Thus, acoustic energy not absorbed through the fibreglass would not be reflected directly back at the ceiling, rather into more fibreglass. Also, this provides an empty storage space below the backing for future small support equipment, for example, the motor of a revolving microphone support.

In that such a prismatic structure would have to be approximately 1.8 m (6') long, it was made in two parts, so that only the front section need be removed for access to the centre of the chamber. The rear section could also be removed for freer access to the chamber interior. These structures were designed to fold when lifted to reduce the awkwardness of moving them through the doorway, and to reduce the structural integrity required. That is, to maintain the triangular cross-section while lifting, a cross member would be required (forming the base of the triangle). Within the chamber, those horizontal forces are supported by the ductwork.

The fibreglass is held in place on the prismatic insert by a sheet of the double-knit fabric, stretched over the fibreglass, and stapled to the back of the support. Plate A.6 shows the rear prismic insert in place.

Finally, the installation of absorbent material on the door side of the chamber was achieved in two stages. Firstly, the door was covered with a single 10 cm (4") layer of fibreglass batt. These were stapled to the door and then covered with the double-knit fabric. Secondly, a stack of fibreglass batts, approximately 1 m tall by 1 m wide and 0.4 m (16") deep, was wrapped in the double-knit fabric and sewn closed. Three lengths of fishing line were fed from the bottom to the top of the stack, one at each end and another in the centre. They were secured at the bottom to the fabric and at the top they were available as lifting handles. The fishing line was passed through three screw-eyes in the ceiling of the chamber and used to suspend the entire stack. Thus, the stack may be lifted into the space immediately in front of the door within the chamber.

This final absorbent treatment, although not drawn into an undulating shape as with the walls, is a limp, highly absorbent panel, providing substantial absorption.

In this way, approximately one half of a cubic metre of open space, anechoic on five of six sides, and significantly isolated from external disturbance is obtained.

Plate A.1:

Plate A.2:

Plate A.3:

Plate A.4:

Plate A.5:

Plate A.6:

APPENDIX B: DESIGN CALCULATIONS

Thermo-acoustic Calculations

A calculation worksheet was developed using MathCAD 2.5 to calculate the lumped response of an exhaust system comprising a transition, cavity muffler, and simple tailpipe at room temperature as well as at an elevated temperature. The heated condition of this model allows for a different temperature in each element, thus a uniform temperature for each exhaust element. The axial temperature variation from one element to the next was adopted from measurements by Brammer [4]. To a certain extent, a similar axial gradient was expected in this experimental apparatus, due to convection. This calculation was intended to steer the design to dimensions which would not have lumped resonances near the frequencies of interest (> 500 Hz and < 5000 Hz). This worksheet and plots of the results at cold and elevated temperatures are presented on the following pages.

Following the calculation worksheet are the results of the QuickBASIC program HEATPIPE.BAS (see Appendix C). The results show the temperatures of the surfaces of the filament, the outer boundary of the exhaust pipe (or cavity muffler), and the air inside the exhaust element, as well as the energy per metre which must be generated by the filament to achieve those temperatures. The desired condition is approximately 100 degrees Celsius difference between the air temperature and the outer surface temperature of the tailpipe (this would be a reasonable temperature gradient). This occurs with a power dissipation of approximately 1010 Watts/metre in an axial rod heater. The program was then iterated again until the power dissipation was approximately 1000 Watts/metre for the cavity muffler, and the resulting temperatures were noted.

The temperatures obtained from the results of the above calculations were then used in another MathCAD worksheet. This worksheet takes the temperature at the filament, in the exhaust element, and at the surface, and calculates the fundamental and 1st harmonic axial acoustic resonances associated with those temperature ranges in each axial element. These resonance ranges are then plotted to verify that resonances of the transition do not coincide with the those of the cavity or tailpipe, and more importantly, that the resonances of the tailpipe and cavity do not coincide.

Finally, the calculations for the design of the 500 Hz 4-Pole High-Pass Electrical Filter. These were performed using a MathCAD 2.5 worksheet.

Lumped Response MathCAD 2.5 Worksheet:

Define all units, based on MKSKelvin

$$m \equiv 1L \quad kg \equiv 1M \quad s \equiv 1T \quad K \equiv 1Q$$

$$in \equiv 0.0254 \cdot m$$

$$rad \equiv 1$$

$$i \equiv \sqrt{-1}$$

$$cycle \equiv 2 \cdot \pi \cdot rad$$

$$c(T) := 331.6 \cdot \frac{m}{s} \sqrt{\frac{T}{273 \cdot K}}$$

$$cold := 0$$

$$hot := 1$$

$$case := cold$$

Set Variables

$$lcham := 4 \cdot in \quad ltp := 3 \cdot in \quad ltran := 1.375 \cdot in$$

$$rcham := \frac{lcham}{2} \quad rcham = 2 \cdot in$$

$$rtp := .375 \cdot in \quad rtran := .375 \cdot in$$

$$To := (273 + case \cdot 50) \cdot K \quad Ttp := (273 + case \cdot 100) \cdot K$$

$$Tcham := (273 + case \cdot 200) \cdot K \quad Ttran := (273 + case \cdot 400) \cdot K$$

$$To = 273 \cdot K \quad Ttp = 273 \cdot K \quad Tcham = 273 \cdot K \quad Ttran = 273 \cdot K$$

$$n := 0 \dots 40$$

$$\frac{n}{2} \quad \omega := f \cdot \frac{cycle}{rad} \quad k(c, \omega) := \frac{\omega}{c}$$

$$f_n := 100 \cdot \frac{(1.26)}{s}$$

$$\max(f) = 1.61894 \cdot 10^3 \cdot \frac{cycle}{s}$$

Define complex Impedance components

At the Tailpipe exit:

$$R_{o_n} := \frac{c(T_o)}{\pi \cdot r_{tp}^2} \cdot (k(c(T_o), \omega))^2 \cdot \frac{r_{tp}^2}{n^4}$$

$$X_{o_n} := \frac{c(T_o) \cdot (0.6)}{\pi \cdot r_{tp}^2} \cdot k(c(T_o), \omega) \cdot r_{tp}$$

In the Tailpipe:

$$R_{tp} := \frac{0}{m \cdot s} \quad X_{tp} := \frac{\omega \cdot l_{tp}}{\pi \cdot r_{tp}^2}$$

In the Chamber

$$R_{cham} := \frac{0}{m \cdot s} \quad X_{cham_n} := \frac{-(c(T_{cham}))^2}{\omega \cdot l_{cham} \cdot \pi \cdot r_{cham}^2}$$

In the Transition:

$$R_{tran} := \frac{0}{m \cdot s} \quad X_{tran} := \frac{\omega \cdot l_{tran}}{\pi \cdot r_{tran}^2}$$

Complex Impedances:

$$Z_o := R_o + X_o \cdot i \quad Z_{tp} := R_{tp} + X_{tp} \cdot i$$

$$Z_{cham} := R_{cham} + X_{cham} \cdot i \quad Z_{tran} := R_{tran} + X_{tran} \cdot i$$

$$\Omega_1 := Z_o + Z_{tp}$$

$$\Omega_2_n := \frac{\Omega_1_n \cdot Z_{cham_n}}{\Omega_1_n + Z_{cham_n}}$$

$$\Omega_3 := \Omega_2 + Z_{tran}$$

case = 0 (0=cold, 1=hot)

114

ltp = 3.in

rham = 2.in

lham = 4.in

frequency @ $\text{Im}(\Omega_3)=0$, system is in resonance
case = 1 (0=cold, 1=hot)

ltp = 3.in

rham = 2.in

lham = 4.in

frequency @ $\text{Im}(\Omega_3)=0$, system is in resonance

Calculation Results for the tube using HEATPIPE.BAS:

With the following parameters:

Filament temp.: 800 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .009525 m
 Pipe Outside Radius: .0111125 m

The inside surface temperature of the pipe is: 483.8108 K
 The average temperature of the air inside the pipe is: 562.8581 K
 $Q_{\text{convective1}} = 23.65384$ $Q_{\text{radiative1}} = 286.9325$ (W/m)
 $Q_{\text{convective2}} = 197.8193$ $Q_{\text{radiative2}} = 112.7132$ (W/m)
 Q1 must equal Q2 at steady state.
 The required heat input per metre is: 310.5594 W/m

With the following parameters:

Filament temp.: 900 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .009525 m
 Pipe Outside Radius: .0111125 m

The inside surface temperature of the pipe is: 554.692 K
 The average temperature of the air inside the pipe is: 641.019 K
 $Q_{\text{convective1}} = 25.83219$ $Q_{\text{radiative1}} = 454.0257$ (W/m)
 $Q_{\text{convective2}} = 272.0946$ $Q_{\text{radiative2}} = 207.538$ (W/m)
 Q1 must equal Q2 at steady state.
 The required heat input per metre is: 479.7452 W/m

With the following parameters:

Filament temp.: 1000 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .009525 m
 Pipe Outside Radius: .011125 m

The inside surface temperature of the pipe is: 630.1633 K

The average temperature of the air inside the pipe is: 722.6225 K

$Q_{\text{convective1}} = 27.66715$ $Q_{\text{radiative1}} = 681.1682$ (W/m)

$Q_{\text{convective2}} = 351.1914$ $Q_{\text{radiative2}} = 357.4441$ (W/m)

Q1 must equal Q2 at steady state.

The required heat input per metre is: 708.7354 W/m

With the following parameters:

Filament temp.: 1100 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .009525 m
 Pipe Outside Radius: .011125 m

The inside surface temperature of the pipe is: 707.4955 K

The average temperature of the air inside the pipe is: 805.6216 K

$Q_{\text{convective1}} = 29.36291$ $Q_{\text{radiative1}} = 981.3898$ (W/m)

$Q_{\text{convective2}} = 432.254$ $Q_{\text{radiative2}} = 578.3942$ (W/m)

Q1 must equal Q2 at steady state.

The required heat input per metre is: 1010.701 W/m

Calculation results for the CAVITY

With the following parameters:

Filament temp.: 800 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .0508 m
 Pipe Outside Radius: .0523875 m

The inside surface temperature of the pipe is: 357.1246 K
 The average temperature of the air inside the pipe is: 383.1761 K
 $Q_{\text{convective1}} = 41.57631$ $Q_{\text{radiative1}} = 365.2344$ (W/m)
 $Q_{\text{convective2}} = 306.8233$ $Q_{\text{radiative2}} = 99.65323$ (W/m)
 Q1 must equal Q2 at steady state.
 The required heat input per metre is: 406.6436 W/m

With the following parameters:

Filament temp.: 900 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .0508 m
 Pipe Outside Radius: .0523875 m

The inside surface temperature of the pipe is: 388.6745 K
 The average temperature of the air inside the pipe is: 418.7525 K
 $Q_{\text{convective1}} = 48.00227$ $Q_{\text{radiative1}} = 588.0372$ (W/m)
 $Q_{\text{convective2}} = 462.6486$ $Q_{\text{radiative2}} = 173.1003$ (W/m)
 Q1 must equal Q2 at steady state.
 The required heat input per metre is: 635.8942 W/m

With the following parameters:

Filament temp.: 1000 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .0508 m
 Pipe Outside Radius: .0523875 m

The inside surface temperature of the pipe is: 428.093 K

The average temperature of the air inside the pipe is: 461.7346 K

$Q_{\text{convective1}} = 53.68955$ $Q_{\text{radiative1}} = 897.3741$ (W/m)

$Q_{\text{convective2}} = 657.3425$ $Q_{\text{radiative2}} = 293.7106$ (W/m)

Q1 must equal Q2 at steady state.

The required heat input per metre is: 951.0583 W/m

With the following parameters:

Filament temp.: 1100 K
 Ambient temp.: 295 K
 Room Wall temp.: 293 K
 Filament emissivity: .85
 Brass emissivity: .6
 Brass Thermal Conductivity: 110 W/mK (at 300K)
 Heater Radius: .003175 m
 Pipe Inside Radius: .0508 m
 Pipe Outside Radius: .0523875 m

The inside surface temperature of the pipe is: 474.3519 K

The average temperature of the air inside the pipe is: 511.1547 K

$Q_{\text{convective1}} = 58.73466$ $Q_{\text{radiative1}} = 1312.493$ (W/m)

$Q_{\text{convective2}} = 885.8335$ $Q_{\text{radiative2}} = 484.7108$ (W/m)

Q1 must equal Q2 at steady state.

The required heat input per metre is: 1370.886 W/m

Resonant frequency range calculations of exhaust elements with radial temperature gradients using a MathCAD 2.5 worksheet

Define all units, based on MKS Kelvin

$$m \equiv 1L \quad K \equiv 1Q \quad s \equiv 1T \quad kg \equiv 1M \quad in \equiv .0254 \cdot m \quad deg \equiv \frac{\pi}{180}$$

$$j \equiv \sqrt{-1} \quad Hz \equiv s^{-1} \quad kHz \equiv 1000 \cdot Hz$$

$$c(T) := 331.6 \cdot \sqrt{\frac{T}{273 \cdot K}} \quad \omega(\lambda, c) := \frac{c}{\lambda} \quad v \equiv \frac{m}{s}$$

$$l := 1 \dots 2 \quad n := 1 \dots 3$$

In the EXPANSION CHAMBER, at increasing radial positions
{1=filament, 2=air, 3=wall}

Temperature

Sound Speed

$$T_{1,1} := 1015 \cdot K$$

$$c \left[\begin{matrix} T \\ 1,1 \end{matrix} \right] = 639.4 \cdot \frac{m}{s}$$

$$T_{1,2} := 470 \cdot K$$

$$c \left[\begin{matrix} T \\ 1,2 \end{matrix} \right] = 435.1 \cdot \frac{m}{s}$$

$$T_{1,3} := 436 \cdot K$$

$$c \left[\begin{matrix} T \\ 1,3 \end{matrix} \right] = 419.1 \cdot \frac{m}{s}$$

In the SIMPLE PIPE, at increasing radial positions
{1=filament, 2=air, 3=wall}

Temperature

Sound Speed

$$T_{2,1} := 1100 \cdot K$$

$$c \left[\begin{matrix} T \\ 2,1 \end{matrix} \right] = 665.6 \cdot \frac{m}{s}$$

$$T_{2,2} := 806 \cdot K$$

$$c \left[\begin{matrix} T \\ 2,2 \end{matrix} \right] = 569.8 \cdot \frac{m}{s}$$

$$T_{2,3} := 707 \cdot K$$

$$c \left[\begin{matrix} T \\ 2,3 \end{matrix} \right] = 533.6 \cdot \frac{m}{s}$$

In the TRANSITION: (Assume temp. is same as in simple pipe.)

$$l_{\text{tran}} := 1.375 \cdot \text{in} \quad \text{so...} \quad \lambda_1 := 4 \cdot l_{\text{tran}} \quad f_n := \omega \left[\lambda_1, c \left[T_{2,n} \right] \right]$$

1st harmonic

$$\lambda_4 := \frac{\lambda_1}{2} \quad f_{9+n} := \omega \left[\lambda_4, c \left[T_{2,n} \right] \right]$$

In the CHAMBER:

$$l_{\text{cham}} := 4 \cdot \text{in} \quad \text{so...} \quad \lambda_2 := 2 \cdot l_{\text{cham}} \quad f_{3+n} := \omega \left[\lambda_2, c \left[T_{1,n} \right] \right]$$

1st Harmonic

$$\lambda_5 := \frac{\lambda_2}{2} \quad f_{12+n} := \omega \left[\lambda_5, c \left[T_{1,n} \right] \right]$$

In the TAILPIPE:

$$l_{\text{pipe}} := 3.3 \cdot \text{in} \quad \text{so...} \quad \lambda_3 := 2 \cdot l_{\text{pipe}} \quad f_{6+n} := \omega \left[\lambda_3, c \left[T_{2,n} \right] \right]$$

1st Harmonic

$$\lambda_6 := \frac{\lambda_3}{2} \quad f_{15+n} := \omega \left[\lambda_6, c \left[T_{2,n} \right] \right]$$

N.B. The wavelength associated with the doubly open-ended pipe assumes the pipe is acoustically large.

$$\lambda_{\text{sys}} := (\text{ltran} + \text{lcham} + \text{lpipe}) \cdot 4$$

$$c_{\text{max}} := c(\text{max}(T)) \quad c_{\text{min}} := c(293 \cdot \text{K})$$

$$a_{1,1} := \omega \left[\frac{\lambda_{\text{sys}}}{c_{\text{min}}} \right] \quad a_{2,1} := \omega \left[\frac{\lambda_{\text{sys}}}{c_{\text{max}}} \right]$$

$$a_{1,2} := \omega \left[\frac{\lambda_{\text{sys}}}{3 \cdot c_{\text{min}}} \right] \quad a_{2,2} := \omega \left[\frac{\lambda_{\text{sys}}}{3 \cdot c_{\text{max}}} \right]$$

$$a_{1,3} := \omega \left[\frac{\lambda_{\text{sys}}}{5 \cdot c_{\text{min}}} \right] \quad a_{2,3} := \omega \left[\frac{\lambda_{\text{sys}}}{5 \cdot c_{\text{max}}} \right]$$

Frequency ranges between maximum & minimum frequencies of standing wave from end to end of the entire system, fundamental & 1st two harmonics

$f_{a1} := f_n$	$f_{b1} := f_{n+3}$	$f_{c1} := f_{n+6}$	fundamentals
$f_{a2} := f_n$	$f_{b2} := f_{n+9}$	$f_{c2} := f_{n+15}$	1st harmonics

Individual Resonant Frequencies of transition, muffler, and tailpipe in the range of temperatures expected.

Subscript of 'f': 1,2,3 fundamental distribution in transition
 4,5,6 fundamental distribution in chamber
 7,8,9 fundamental distribution in tailpipe
 10,11,12 1st Harmonic distribution in transition
 13,14,15 1st Harmonic distribution in chamber
 16,17,18 1st Harmonic distribution in tailpipe

Symbols: plusses transition
 exes chamber
 diamonds tailpipe

MathCAD 2.5 calculations for the insertion losses of a muffler and simple tailpipe as well as a muffler and middle-hole co-axial tailpipe system as compared to a simple tube:

$k = \omega/c$ Plane Wave wavenumber

Sound speed (in/s), radius (in), length (in)

$c := 1143$
 $\omega := 504$ $r := .375$ $l_{ref} := 10$
 $c_{air} := 11048$

$l_{tr} := 1.375$ $l_{cav} := 4$ $l_{tp} := 3.3$ $l_{cx} := \frac{l_{tp}}{2}$
 $r_{cav} := 2$
 $dh := .25$ $S_{cx} := \pi \cdot \left[.5625^2 - .4375^2 \right]$ $t := .0625$

$l_a := l_{tp} \cdot \frac{1}{5}$
 $l_b := l_{tp} \cdot \frac{4}{5}$

$Y := \frac{c}{\pi \cdot r^2}$ Characteristic impedance of tube

$Y_{cav} := \frac{c}{\pi \cdot r_{cav}^2}$ Characteristic impedance of cavity

$i := 1 \dots 50$ Array index

$f_i := 504 + 64 \cdot i$ Frequency

$\omega_i := f_i \cdot 2 \cdot \pi$ Radial frequency (radians per second)

$k_i := \frac{\omega_i}{c}$ Plane Wavenumber

$Z_{r_i} := Y \cdot \left[\frac{\omega_i^2}{c^2} \cdot r^2 \right] + j \cdot Y \cdot .85 \cdot \frac{\omega_i}{c} \cdot r$ Radiation Impedance

$$M_{src}(i) := \begin{bmatrix} 1 & \omega \cdot 10^{-20} \\ 0 & 1 \end{bmatrix} \quad \begin{array}{l} \text{Assumed Infinite} \\ \text{Source Impedance} \\ \text{Transfer Matrix} \end{array}$$

$$M_l(i) := \begin{bmatrix} \cos\left[\frac{\omega}{c} \cdot l_{ref}\right] & j \cdot Y \cdot \sin\left[\frac{\omega}{c} \cdot l_{ref}\right] \\ j \cdot \frac{1}{Y} \cdot \sin\left[\frac{\omega}{c} \cdot l_{ref}\right] & \cos\left[\frac{\omega}{c} \cdot l_{ref}\right] \end{bmatrix} \quad \begin{array}{l} \text{Tube Transfer} \\ \text{Matrix} \end{array}$$

$$M_r(i) := \begin{bmatrix} Z_r & \\ 1 & i \\ 0 & 1 \end{bmatrix} \quad \begin{array}{l} \text{Simplified for } kr < .5 \\ \text{Radiation Impedance} \\ \text{Transfer Matrix} \end{array}$$

Assume $Z_{src} \gg Z_{rad}$, then insertion loss is dependent only on velocity ratio, VR:

$$VR_{xp}_i := \left| \frac{(M_{src}(i) \cdot M_l(i) \cdot M_r(i))_{2,2}}{2,2} \right|$$

and insertion loss of exhaust pipe compared to no elements is:

$$IL_{xp}_i := 20 \cdot \log[VR_{xp}_i]$$

Transfer Matrices:

Transition:

$$M_{tr}(i) := \begin{bmatrix} \cos[k \cdot l_{tr}] & j \cdot Y \cdot \sin[k \cdot l_{tr}] \\ j \cdot \sin[k \cdot l_{tr}] & \cos[k \cdot l_{tr}] \\ Y & i \end{bmatrix}$$

Cavity:

$$M_{cav}(i) := \begin{bmatrix} \cos[k \cdot l_{cav}] & j \cdot Y_{cav} \cdot \sin[k \cdot l_{cav}] \\ j \cdot \sin[k \cdot l_{cav}] & \cos[k \cdot l_{cav}] \\ Y_{cav} & i \end{bmatrix}$$

Co-ax Upstream

$$M_{cxup}(i) := \begin{bmatrix} \cos[k \cdot l_{cx}] & j \cdot Y \cdot \sin[k \cdot l_{cx}] \\ j \cdot \sin[k \cdot l_{cx}] & \cos[k \cdot l_{cx}] \\ Y & i \end{bmatrix}$$

Co-ax Downstream -same as upstream

Impedance of Co-ax at hole

$$Z_{cx} := j \cdot \omega \cdot \frac{t + .85 \cdot dh}{\pi \cdot c} + \frac{\omega^2}{\pi \cdot c} - j \cdot \frac{c}{2 \cdot \tan[k \cdot l_{cx}]}$$

Co-ax Annulus:

$$M_{cx}(i) := \begin{bmatrix} 1 & 0 \\ 1 & Z_{cx} \\ Z_{cx} & 1 \\ i & \end{bmatrix}$$

Simple Tailpipe:

$$M_{tp}(i) := \begin{bmatrix} \cos[k \cdot 2 \cdot l_{cx}] & j \cdot Y \cdot \sin[k \cdot 2 \cdot l_{cx}] \\ j \cdot \sin[k \cdot 2 \cdot l_{cx}] & \cos[k \cdot 2 \cdot l_{cx}] \\ Y & i \end{bmatrix}$$

Assume $Z_{src} \gg Z_{rad}$, then insertion loss is dependent only on velocity ratio, VR:

$$VR_{cx_i} := \frac{(M_{src}(i) \cdot M_{tr}(i) \cdot M_{cav}(i) \cdot M_{cxup}(i) \cdot M_{cx}(i) \cdot M_{cxup}(i) \cdot M_r(i))}{2,2}$$

and insertion loss of tube compared to no elements is:

$$IL_{cx_i} := 20 \cdot \log[VR_{cx_i}]$$

The velocity ratio for the cavity muffler and simple tailpipe:

$$VR_{cp_i} := \frac{(M_{src}(i) \cdot M_{tr}(i) \cdot M_{cav}(i) \cdot M_{tp}(i) \cdot M_r(i))}{2,2}$$

$$IL_{cp_i} := 20 \cdot \log[VR_{cp_i}]$$

$$M_{cxa}(i) := \begin{bmatrix} \cos[k \cdot la] & j \cdot Y \cdot \sin[k \cdot la] \\ j \cdot \sin[k \cdot la] & \cos[k \cdot la] \\ Y & \end{bmatrix}_i$$

$$M_{cxb}(i) := \begin{bmatrix} \cos[k \cdot lb] & j \cdot Y \cdot \sin[k \cdot lb] \\ j \cdot \sin[k \cdot lb] & \cos[k \cdot lb] \\ Y & \end{bmatrix}_i$$

$$VR_{cxab_i} := \frac{(M_{src}(i) \cdot M_{tr}(i) \cdot M_{cav}(i) \cdot M_{cxa}(i) \cdot M_{cx}(i) \cdot M_{cxb}(i) \cdot M_r(i))}{2,2}$$

and insertion loss of tube compared to no elements is:

$$IL_{cxab_i} := 20 \cdot \log[VR_{cxab_i}]$$

Insertion Loss of Muffler and simple tailpipe
compared to the simple tube exhaust

Insertion Loss of Muffler and mid-hole co-ax
tailpipe compared to the simple tube exhaust

Insertion Loss of Muffler and mid-hole co-ax
tailpipe compared to the simple tube exhaust

MathCAD 2.5 calculations for the response of the 500 Hz High-Pass Electrical Filter:

$j := \sqrt{-1}$ Complex unit vector

$k := 0 \dots 70$ Index for regular variables

$i := 0 \dots 71$ Index for variables including $f=\infty$

$\mu := 10^{-6}$

$f_k := (1 + k)^2$ Angular frequency, Hz (up to $f=5041$ Hz)

$f_{71} := \left[\frac{\infty}{2 \cdot \pi} \right]^{.1}$ "Infinite" angular frequency in Hz

$\omega_i := 2 \cdot \pi \cdot f_i$ Angular frequency in radians/second

$R := 250$ Value of resistance, Ω

$C := 11.5 \cdot \mu$ Value of Capacitance, F ($\mu=1$ E-6)

Transfer Function of Filter

$$TF_i := \frac{\left[\frac{\omega_i \cdot C}{i} \right]^8 + \left[\frac{\omega_i \cdot C}{i \cdot R} \right]^4 - 15 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^6}{R} - j \cdot \left[10 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^7}{R} - 7 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^5}{R^3} \right]}{\frac{1}{R^5} + 19 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^2}{R^6} + 87 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^4}{R} + 70 \cdot \frac{\left[\frac{\omega_i \cdot C}{i} \right]^6}{R^2} + \left[\frac{\omega_i \cdot C}{i} \right]^8}$$

Amplitude Response of the Filter

Phase response of the Transfer Function

Nyquist Plot of Transfer Function. $0,0 @ f \rightarrow 0, 1,0 @ f \rightarrow \infty$

Real Component of Transfer Function

Imaginary Component of Transfer Function

APPENDIX C: COMPUTER PROGRAMS: DATA ACQUISITION, PROCESSING, AND CALCULATIONS

Probe Microphone Data

The following flow chart outlines the scheme for acquiring the probe tube data. The source code for the two computer programs written in QuickBASIC and compiled follow. The first reads the digital values produced by the three-channel A/D converter (type Data Translation DT-2828) into the computer's memory and then immediately to disk storage in a compressed binary format. The second program reads the saved data, separates the three channels of data, and then saves the data in three separate ASCII format data files. These data files then contain the digitized time history of each channel. These files are then processed via a standard FFT routine [21,22], also written in QuickBASIC (not included here for brevity), to produce the real and imaginary components of the spectrum. (Figure C.2 shows a sample screen image during this processing of the time signal and the magnitude of the FFT.) These two components are saved as ASCII data files. The FFT files are then imported into a spreadsheet (Lotus 1-2-3) where the absolute phase angle and amplitudes are calculated. The amplitudes and phase angles relative to the data for microphone number three (the probe located at mid-pipe) are then plotted. These figures are found in the body of the thesis and in Appendix D.

Thermal/Acoustic Calculations

The program source code following the data acquisition programs is a QuickBASIC program called HEATPIPE.BAS which calculates the temperatures of the pipe surface and air within the pipe, as well as the thermal energy required to achieve those temperatures at steady state. These calculations are based on estimated thermal emissivities [18] of the metal surfaces and convection coefficients for the air-surface interfaces [19]. The calculations are iterated based on an assumed filament temperature until steady state is reached. The various heating requirements are then analyzed for the heat energy input required to achieve a 100 degree Celsius difference between the internal air temperature and the outer surface of the pipe. The program is used again to calculate the temperatures in the 100 mm diameter expansion chamber muffler with the same heat energy.

Figure C.1: Information flow chart for probe tube measurements.

```

' Program AD3C.BAS

' Double-Buffered 3-Channel Analogue to Digital Conversion Program
'   Using Low-Level Call Routines of ATLAB v1.11

' Written by Alexander P. C. K. S. Lorimer, July, 1990

```

```

' Declare which ATLAB BASIC files are to be included at compile-time

```

```

' $INCLUDE: 'C:\qb45\incl\atlab\ATLDEFS.BI'
' $INCLUDE: 'C:\qb45\incl\atlab\ATLERRS.BI'
' $INCLUDE: 'C:\qb45\incl\atlab\ATLEXSUB.BI'

```

```

DEFINT A-Z

```

```

DECLARE SUB plot (chA() AS INTEGER, chB() AS INTEGER, chC() AS INTEGER, bufflen AS INTEGER)

```

```

' Declare COMMON variables between the program and ATLAB modules

```

```

COMMON /SPACE0/ buffnum() AS INTEGER, ScanCount AS INTEGER, channel() AS INTEGER
COMMON /SPACE0/ gain() AS INTEGER, DataFileHeader AS AIFileHeader
COMMON /SPACE0/ configuration AS AIConfiguration

```

```

COMMON /SPACE1/ buff1(), buff2(), buffx(), buffy()

```

```

COMMON SHARED zoomx AS INTEGER, zoomy AS INTEGER

```

```

' Set Buffer size to 8 kB

```

```

CONST bufsize = 8192
CONST errcon = 1

```

```

' Dimension Buffers and other variables. Note that 4 (four) buffers are defined allowing
'   up to two errors in the buffers declaration call below.

```

```

DIM buff1(0 TO 8191), buff2(0 TO 8191), buffx(0 TO 8191), buffy(0 TO 8191)
DIM chA(0 TO 8191), chB(0 TO 8191), chC(0 TO 8191)
DIM buffnum(2) AS INTEGER
DIM channel(0 TO 15) AS INTEGER
DIM gain(0 TO 15) AS INTEGER

```

' BEGIN PROGRAM

' Initialize routines and do setup housekeeping.

' Initialize subroutines

reterr = AI.Initialize

' Turn on error monitor

reterr = AI.Set.Error.Control.Word(errcon)

Select board 1

unit = 1

reterr = AI.Select.Board(unit)

' Reset device

reterr = AI.Reset

' Display the current configuration

CALL DUMP CONFIGURATION

' Open the Data file. Prompts for filename of data.

PRINT "Enter the name of the save file <*.DAT>";

INPUT filename\$

filename\$ = filename\$ + ".DAT" + CHR\$(0)

IF reterr <> 0 THEN PRINT "Error"; reterr; ". Terminating"; reterr = AI.Terminate: STOP

reterr = AI.File.Open(SADD(filename\$), VARSEG(filename\$), AI.Write, DataFileHeader)

IF reterr <> 0 THEN PRINT "Error"; reterr; ". Terminating"; reterr = AI.Terminate: STOP

' Set up A/D parameters

ScanCount = 3

PRINT "Channel A is 0, B is 2 and C is 3"

channel(0) = 0: channel(1) = 2: channel(2) = 3

gain(0) = 1: gain(1) = 1: gain(2) = 1

reterr = AI.Setup.Adc(InternalTrigger + InternalClock, ScanCount, channel(0), gain(0))

' Set Frequency

PRINT INPUT "Enter the desired composite acquisition rate in Hertz<60 kHz>"; rate\$

IF rate\$ = "" THEN

rate! = 60000

ELSE rate! = VAL(rate\$)

END IF

reterr = AI.Set.Frequency(rate!)

' Enter the File Annotation here

PRINT INPUT "Enter any comments"; DataFileHeader.AcqComment

- **Declare, and link buffers**

- **1st Buffer Declaration. Two tries; either Buff1 or Buffx**

```
bufflen = bufsize
```

```
PRINT "Buffer length 1 ="; bufflen; " elements"
```

```
reterr = Al.Declare.Buffer(buffnum(0), buff1(0), bufflen)
```

```
IF reterr = 0 THEN
```

```
    reterr = Al.Link.Buffer(buffnum(0))
```

```
    buffdesc1$ = "1"
```

```
    PRINT "Buffer 1 is buff"; buffdesc1$
```

```
ELSE
```

```
    PRINT "Error #"; reterr; " in declaring buffer 1 as buff1"
```

```
    reterr = Al.Declare.Buffer(buffnum(0), buffx(0), bufflen)
```

```
    IF reterr = 0 THEN
```

```
        reterr = Al.Link.Buffer(buffnum(0))
```

```
        buffdesc1$ = "X"
```

```
        PRINT "Buffer 1 is buff"; buffdesc1$
```

```
    ELSE
```

```
        PRINT "Error #"; reterr; " in declaring buffer 1 as buffx"
```

```
        STOP
```

```
    END IF
```

```
END IF
```

- **2nd Buffer Declaration. Two tries; either Buff2 or Buffy**

```
PRINT "Buffer length 2 ="; bufflen; " elements"
```

```
reterr = Al.Declare.Buffer(buffnum(1), buff2(0), bufflen)
```

```
IF reterr = 0 THEN
```

```
    reterr = Al.Link.Buffer(buffnum(1))
```

```
    buffdesc2$ = "2"
```

```
    PRINT "Buffer 2 is buff"; buffdesc2$
```

```
ELSE
```

```
    PRINT "Error #"; reterr; " in declaring buffer 2 as buff2"
```

```
    reterr = Al.Declare.Buffer(buffnum(1), buffy(0), bufflen)
```

```
    IF reterr = 0 THEN
```

```
        reterr = Al.Link.Buffer(buffnum(1))
```

```
        buffdesc2$ = "Y"
```

```
        PRINT "Buffer 2 is buff"; buffdesc2$
```

```
    ELSE
```

```
        PRINT "Error #"; reterr; " in declaring buffer 2 as buffy"
```

```
        STOP
```

```
    END IF
```

```
END IF
```

- **Describe which buffers have been declared**

```
buffdesc$ = buffdesc1$ + buffdesc2$
```

- **Rewind file**

```
reterr = Al.File.Rewind(DataFileHeader)
```

```
IF reterr <> 0 THEN PRINT "Error"; reterr; " Terminating"; reterr = Al.Terminate STOP
```

```
CALL DUMP.FILE.HEADER(DataFileHeader)
```


- **Begin Data Acquisition**

```
PRINT "BEGINNING ACQUISITION"
```

```
reterr = AI.Continuous.Adc 'Initiates A/D on buffer #0 (1st on TransferList)
```

- **Wait for the release of buffer#0, write it to file, then
return it to the transfer list (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File.Write(releasedbuffer, DataFileHeader)
reterr = AI.Return.Buffer(releasedbuffer)
```

- **Wait for the release of buffer#1, write it to file, then
return it to the transfer list (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File.Write(releasedbuffer, DataFileHeader)
reterr = AI.Return.Buffer(releasedbuffer)
```

- **Wait for the release of buffer#0, write it to file, then
return it to the transfer list (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File.Write(releasedbuffer, DataFileHeader)
reterr = AI.Return.Buffer(releasedbuffer)
```

- **Wait for the release of buffer#1, write it to file, then
return it to the transfer list as the LAST buffer (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File.Write(releasedbuffer, DataFileHeader)
reterr = AI.Return.As Last(releasedbuffer)
```

- **Wait for the release of buffer#0, write it to file, then
return it to the transfer list (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File Write(releasedbuffer, DataFileHeader)
reterr = AI.Return.Buffer(releasedbuffer)
```

- **Wait for the release of buffer#1, write it to file, then, because it's the
last, no need to return it to the transfer list (this is saved data)**

```
IF AI.Release.Buffer(unit, releasedbuffer) = 0 THEN PRINT "buffer "; releasedbuffer; " is done"
reterr = AI.File Write(releasedbuffer, DataFileHeader)
```

```
CALL DUMP FILE.HEADER(DataFileHeader)
```

- **Close file. This actually writes to disk.**

```
reterr = AI.File.Close(DataFileHeader)
```

- **Terminate ATLAS and pause. This ends use of the ATLAS routines**

```
reterr = Al.Terminate
SLEEP
PRINT : PRINT : PRINT
```

- **Print out some data from last two buffers to ensure data has been obtained.**

```
FOR i = 0 TO buffsize \ 1000
    PRINT buff1(i), buffx(i), buff2(i), buffy(i)
NEXT i
SLEEP 3
```

- **Separate channels**

The order of channel storage in the buffers is 0, 2, 3, but when separating channels from the last two bufferloads, the starting value is a channel 3 value. If the first pair of buffers "timeouts" then the first two bufferloads (stored on file) are invalid, but the following display will be in the correct order

Therefore, the order for separation of the three channels for the last two buffers is C B A.

This results because 2×8192 is not a multiple of 3, and neither is 4×8192 . 6×8192 is (obviously) a multiple of three, and therefore, the data stored on the file (provided no timeout occurs) is in the channel order of A B C.

- **Separate channels based on which buffers were used.**

```
SELECT CASE buffdesc$
CASE "12"
    FOR i = 0 TO buffsize - 3 STEP 3
        chC(i \ 3) = buff1(i)
        chA(i \ 3) = buff1(i + 1)
        chB(i \ 3) = buff1(i + 2)
        chC(i \ 3 + 1 + (buffsize - 3) \ 3) = buff2(i + 1)
        chA(i \ 3 + 1 + (buffsize - 3) \ 3) = buff2(i + 2)
        chB(i \ 3 + 1 + (buffsize - 3) \ 3) = buff2(i + 3)
    NEXT i
    chC(2730) = buff1(8190): chA(2730) = buff1(8191): chB(2730) = buff2(0)
    FOR i = 0 TO 15
        PRINT chA(i + 2725), chB(i + 2725), chC(i + 2725)
    NEXT i
    SLEEP
```

```
CASE "1Y"
    FOR i = 0 TO buffsize - 3 STEP 3
        chC(i \ 3) = buff1(i)
        chA(i \ 3) = buff1(i + 1)
        chB(i \ 3) = buff1(i + 2)
        chC(i \ 3 + 1 + (buffsize - 3) \ 3) = buffy(i + 1)
        chA(i \ 3 + 1 + (buffsize - 3) \ 3) = buffy(i + 2)
        chB(i \ 3 + 1 + (buffsize - 3) \ 3) = buffy(i + 3)
    NEXT i
```

```

chC(2730) = buff1(8190) chA(2730) = buff1(8191): chB(2730) = buffy(0)
FOR i = 0 TO 15
    PRINT chA(i + 2725), chB(i + 2725), chC(i + 2725)
NEXT i
SLEEP

CASE "X2"
    FOR i = 0 TO bufsize - 3 STEP 3
        chC(i \ 3) = buffx(i)
        chA(i \ 3) = buffx(i + 1)
        chB(i \ 3) = buffx(i + 2)
        chC(i \ 3 + 1 + (bufsize - 3) \ 3) = buff2(i + 1)
        chA(i \ 3 + 1 + (bufsize - 3) \ 3) = buff2(i + 2)
        chB(i \ 3 + 1 + (bufsize - 3) \ 3) = buff2(i + 3)
    NEXT i
    chC(2730) = buffx(8190) chA(2730) = buffx(8191): chB(2730) = buff2(0)
    FOR i = 0 TO 15
        PRINT chA(i + 2725), chB(i + 2725), chC(i + 2725)
    NEXT i
    SLEEP

CASE "XY"
    FOR i = 0 TO bufsize - 3 STEP 3
        chC(i \ 3) = buffx(i)
        chA(i \ 3) = buffx(i + 1)
        chB(i \ 3) = buffx(i + 2)
        chC(i \ 3 + 1 + (bufsize - 3) \ 3) = buffy(i + 1)
        chA(i \ 3 + 1 + (bufsize - 3) \ 3) = buffy(i + 2)
        chB(i \ 3 + 1 + (bufsize - 3) \ 3) = buffy(i + 3)
    NEXT i
    chC(2730) = buffx(8190) chA(2730) = buffx(8191): chB(2730) = buffy(0)
    FOR i = 0 TO 15
        PRINT chA(i + 2725), chB(i + 2725), chC(i + 2725)
    NEXT i
    SLEEP

END SELECT

PRINT buffdesc$
SLEEP 3
zoomx = 1 zocmy = 1

Call procedure to plot the data on the screen

picture
    CALL plot(chA(), chB(), chC(), (bufflen \ 3))
SLEEP 1

```

· Prompt to view the data at a different Zoom.

```
LOCATE 24, 1: PRINT "<Z> for zoom, <RETURN> for quit".
INPUT z$
IF UCASE$(z$) = "Z" THEN
    CLS 0
    LOCATE 2, 10: PRINT "Current zoomX is ", zoomx
    LOCATE 3, 10: PRINT "Current zoomY is "; zoomy
    PRINT : PRINT "Enter new zoomX and zoomY": INP(1) zoomx, zoomy
    GOTO picture
END IF
```

END

· -----End of Program-----

· Procedure Module to plot data on the screen

```
DEFNG J
DEFSNG A-I, K-Z
```

```
SUB plot (chA() AS INTEGER, chB() AS INTEGER, chC() AS INTEGER, endel AS INTEGER)
```

· Setup screen co-ordinates

```
CLS 0
SCREEN 3
LOCATE 23, 5: PRINT "WAIT"
```

· Find the average DC offset

```
aver1 = 0: aver2 = 0: aver3 = 0
FOR J2 = 0 TO endel - 1
    aver1 = chA(J2) + aver1
    aver2 = aver2 + chB(J2)
    aver3 = aver3 + chC(J2)
NEXT J2
aver1 = aver1 / endel
aver2 = aver2 / endel
aver3 = aver3 / endel
```

· Set numerical boundaries for the channel A data plot

```
VIEW (10, 2)-(710, 97), , 1
XB = 1: XE = 2 * endel: YB = -aver1: YE = aver1
XR = XE - XB: YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
```

· Draw the Channel A boundary lines on the screen

```
LINE (XB + XR \ 4, YB + 400)-(XB + XR \ 4, YE - 400)
LINE (XB + XR \ 2, YB + 400)-(XB + XR \ 2, YE - 400)
LINE (XB + 3 * XR \ 4, YB + 400)-(XB + 3 * XR \ 4, YE - 400)
LINE (XB, YR \ 2)-(XE, YR \ 2)
```

· **Draw the Channel A data on the screen**

```
PSET (zoomx * 1, zoomy * (chA(0) - aver1))
FOR J1 = 1 TO 2 * endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chA(J1) - aver1))
NEXT J1
```

· **Set numerical boundaries for the channel B data plot**

```
VIEW (10, 102)-(710, 197), , 1
YB = -aver2 YE = aver2
XR = XE - XB YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
```

Draw the Channel A boundary lines on the screen

```
LINE (XB + XR \ 4, YB + 400)-(XB + XR \ 4, YE - 400)
LINE (XB + XR \ 2, YB + 400)-(XB + XR \ 2, YE - 400)
LINE (XB + 3 * XR \ 4, YB + 400)-(XB + 3 * XR \ 4, YE - 400)
LINE (XB, YR \ 2)-(XE, YR \ 2)
```

· **Draw the Channel B data on the screen**

```
PSET (zoomx * 1, zoomy * (chB(0) - aver2))
FOR J1 = 1 TO 2 * endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chB(J1) - aver2))
NEXT J1
```

· **Set numerical boundaries for the channel C data plot**

```
VIEW (10, 202)-(710, 297), , 1
YB = -aver3 YE = aver3
XR = XE - XB YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
```

Draw the Channel C boundary lines on the screen

```
LINE (XB + XR \ 4, YB + 400)-(XB + XR \ 4, YE - 400)
LINE (XB + XR \ 2, YB + 400)-(XB + XR \ 2, YE - 400)
LINE (XB + 3 * XR \ 4, YB + 400)-(XB + 3 * XR \ 4, YE - 400)
LINE (XB, YR \ 2)-(XE, YR \ 2)
```

· **Draw the Channel C data on the screen**

```
PSET (zoomx * 1, zoomy * (chC(0) - aver3))
FOR J1 = 1 TO 2 * endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chC(J1) - aver3))
NEXT J1
```

END SUB

' Program BIN2ASC3.BAS

' Import Binary 3-Channel Digital Data, Segregate Channels, and Save as 3 ASCII Data Digital Files
' Using Low-Level Call Routines of ATLAB v1.11

' Written by Alexander P. C. K. S. Lorimer, July, 1990

' Declare which ATLAB BASIC files are to be included at compile-time

' \$INCLUDE: 'c:\qb45\incl\atlab\atldefs.bi'
' \$INCLUDE: 'c:\qb45\incl\atlab\atlerrs.bi'
' \$INCLUDE: 'c:\qb45\incl\atlab\atlsub.bi'

' \$DYNAMIC

DEFINT A-Z

' Declare COMMON variables between the program and ATLAB modules

COMMON /space0/ DataFileHeader AS AIFileHeader, Configuration AS AICConfiguration
COMMON /space0/ bufnum() AS INTEGER

COMMON /space1/ buff1(), buff2(), buffx()

COMMON SHARED zoomx AS INTEGER, zoomy AS INTEGER, fl\$, flag

DECLARE SUB fileget ()
DECLARE SUB PLOT (chA() AS INTEGER, chB() AS INTEGER, chC() AS INTEGER, endel AS INTEGER)
DECLARE SUB filesave (buff1() AS INTEGER, buff2() AS INTEGER, buff3() AS INTEGER, endel AS INT)

' Setup constants

CONST TRUE = 1
CONST false = 0

CONST NBUFFERS = 2 ' number of data buffers
CONST bufsize = 8192 ' buffer size in words

' Declare channel, gain, and buffer number arrays

DIM channels(16) AS INTEGER ' channel scan list
DIM gains(16) AS INTEGER ' gains for channels
DIM bufnum(2) AS INTEGER ' storage for returned buffer numbers

' Declare store area

DIM buff1(0 TO 8191)
DIM buff2(0 TO 8191)
DIM buffx(0 TO 8191)
DIM chA(0 TO 16383)
DIM chB(0 TO 16383)
DIM chC(0 TO 16383)

' BEGIN PROGRAM

beginning

flag = false

CLS 0

PRINT ""

PRINT "This program performs double-buffered file reading "

PRINT "from a binary data file created by ATLAB, displays"

PRINT "the signal, and then saves the data to disk in an "

PRINT "ASCII format data file (PRN). "

SLEEP 3

' Initialize the ATLAB subroutines.

retErr = Al.Initialize

' Declare and Link buffers to ATLAB

IF Al.Declare.Buffer(bufnum(0), buff1(0), buffsize) = 0 THEN

retErr = Al.Link.Buffer(bufnum(0))

ELSE

PRINT "Alternate buffer used for buffer 1 .."

retErr = Al.Declare.Buffer(bufnum(0), buffx(0), buffsize)

retErr = Al.Link.Buffer(bufnum(0))

END IF

IF Al.Declare.Buffer(bufnum(1), buff2(0), buffsize) = 0 THEN

retErr = Al.Link.Buffer(bufnum(1))

ELSE

PRINT "Alternate buffer used for buffer 2 .."

retErr = Al.Declare.Buffer(bufnum(1), buffx(0), buffsize)

retErr = Al.Link.Buffer(bufnum(1))

END IF

' Open the Binary Data File.

CALL fileget

filename\$ = fl\$ + CHR\$(0)

retErr = Al.File.Open(SADD(filename\$), VARSEG(filename\$), AlRead, DataFileHeader)

' Display data file header information

CALL DUMP.FILE.HEADER(DataFileHeader)

Calculate the sample rate of the data in the file in KHz.

rate! = 4000! / DataFileHeader.ClockPeriod

PRINT USING "Composite samples were recorded at #####.## KHz."; rate!

' Load the first two buffers.

FOR i = 0 TO NBUFFERS - 1

retErr = Al.File.Read(bufnum(i), DataFileHeader)

NEXT i

```

' Enter the first two buffers full into chA(), chB(), and chC()
' This enters data from 0 to buffersize-1 (8191)

FOR i = 0 TO buffersize - 3 STEP 3
    chA(i \ 3) = buff1(i)
    chB(i \ 3) = buff1(i + 1)
    chC(i \ 3) = buff1(i + 2)
    chA(i \ 3 + 2 + (buffersize - 3) \ 3) = buff2(i + 1)
    chB(i \ 3 + 2 + (buffersize - 3) \ 3) = buff2(i + 2)
    chC(i \ 3 + 2 + (buffersize - 3) \ 3) = buff2(i + 3)
NEXT i

    chA(2730) = buff1(8190): chB(2730) = buff1(8191): chC(2730) = buff2(0)
    chA(5461) = buff2(8191)

' Load the second two buffers.

FOR i = 0 TO NBUFFERS - 1
    reterr = AI.File.Read(bufnum(i), DataFileHeader)
NEXT i

FOR i = 0 TO buffersize - 3 STEP 3
    chA(i \ 3 + 5462) = buff1(i + 2)
    chB(i \ 3 + 5462) = buff1(i + 3)
    chC(i \ 3 + 5462) = buff1(i + 4)
    chA(i \ 3 + 5463 + (buffersize - 3) \ 3) = buff2(i)
    chB(i \ 3 + 5463 + (buffersize - 3) \ 3) = buff2(i + 1)
    chC(i \ 3 + 5463 + (buffersize - 3) \ 3) = buff2(i + 2)
NEXT i
    chB(5461) = buff1(0): chC(5461) = buff1(1)
    chA(10922) = buff2(8190): chB(10922) = buff2(8191)

' Load the last two buffers

FOR i = 0 TO NBUFFERS - 1
    reterr = AI.File.Read(bufnum(i), DataFileHeader)
NEXT i

' Close the data file.

reterr = AI.File.Close(DataFileHeader)

' Terminate ATLAB operations.

reterr = AI.Terminate
SLEEP 3

```


' **Separate channels**

' This enters data from 0 to buffersize-1 (8191)

FOR i = 0 TO buffersize - 3 STEP 3

 chA(i \ 3 + 10923) = buff1(i + 1)

 chB(i \ 3 + 10923) = buff1(i + 2)

 chC(i \ 3 + 10923) = buff1(i + 3)

 chA(i \ 3 + 10925 + (buffersize - 3) \ 3) = buff2(i + 2)

 chB(i \ 3 + 10925 + (buffersize - 3) \ 3) = buff2(i + 3)

 chC(i \ 3 + 10925 + (buffersize - 3) \ 3) = buff2(i + 4)

NEXT i

 chC(10922) = buff1(0)

 chA(13653) = buff2(8191): chB(13653) = buff2(0): chC(13653) = buff2(1)

' **Display the data**

zoomx = 1 zoomy = 1

picture:

CALL PLOT(chA(), chB(), chC(), ((buffersize - 1) \ 3) * 6)

LOCATE 24, 1 PRINT "<Z> for zoom, <RETURN> for quit";

INPUT z\$

IF UCASE\$(z\$) = "Z" THEN

 CLS

 LOCATE 2, 10: PRINT "Current zoomX is", zoomx

 LOCATE 3, 10: PRINT "Current zoomY is", zoomy

 PRINT : PRINT "Enter zoomX and zoomY"; : INPUT zoomx, zoomy

 GOTO picture

END IF

' **ASCII filesave routine**

endel = 2 * buffersize

CALL filesave(chA(), chB(), chC(), endel)

IF endel = -1 THEN GOTO beginning

END ' **Program Termination** -----

' Save-file Error Handling Routine

filesaveerror:

```

LOCATE 20, 15
SELECT CASE ERR
  CASE 52
    PRINT "Error #"; ERR; "Bad filename or number"
  CASE 53
    PRINT "File not found, saving ", fl$
    RESUME NEXT
  CASE 58
    PRINT "Error #"; ERR; "File", fl$, " already exists"
    LOCATE 21, 15: PRINT "OK to overwrite <Y/N>  N"
    LOCATE 21, 37: INPUT ; q$: q$ = UCASE$(q$)
    IF q$ = "Y" THEN RESUME NEXT
  CASE 61
    PRINT "Error #"; ERR; "Disk full. Include new path"
  CASE 64
    PRINT "Error #"; ERR; "Bad filename"
  CASE ELSE
    PRINT "Error #"; ERR
END SELECT

fls$ = ".PRN"
LOCATE 23, 1: INPUT "ENTER THE SAVE FILENAME <.PRN IS THE FILESPEC>". fl$
IF fl$ = "" THEN RESUME beginning
fl$ = UCASE$(fl$) + fls$
RESUME 0

```

' End of Routine -----

' Get-file Error Handling Routine

filegeterror

LOCATE 20, 15

SELECT CASE ERR

 CASE 52

 PRINT "Error #"; ERR, "Bad filename or number"

 CASE 53

 PRINT "File not found"

 CASE 58

 PRINT "Error #"; ERR, "File", fl\$, " exists"

 RESUME NEXT

 CASE 61

 PRINT "Error #"; ERR, "Disk full Include new path"

 CASE 64

 PRINT "Error #"; ERR, "Bad filename"

 CASE ELSE

 PRINT "Error #"; ERR

END SELECT

fls\$ = " DAT"

LOCATE 23, 1 INPUT "ENTER THE BINARY FILENAME <.DAT IS THE FILESPEC>"; fl\$

IF fl\$ = "" THEN

 PRINT "That's all folks"

 END Program Termination

END IF

fl\$ = UCASE\$(fl\$) + fls\$

RESUME 0

' End of Routine

' End of Program

' Procedure Module to Get a Binary Data File

' \$STATIC

SUB fileget

fls\$ = ".DAT"

CLS 0

FILES "*.DAT"

LOCATE 23, 1: INPUT "ENTER THE BINARY FILENAME < DAT IS THE FILESPEC>"; fls\$

fls\$ = UCASE\$(fls\$) + fls\$

ON ERROR GOTO filegeterror

NAME fls\$ AS fls\$ ' if it exists then error 58, if not then error 53

END SUB

' End of Procedure

' Procedure Module to Save ASCII Data File

DEFSNG A-Z

SUB filesave (buff1() AS INTEGER, buff2() AS INTEGER, buff3() AS INTEGER, endel AS INTEGER)

SHARED flag, fls\$

flA\$ = "P_A" flB\$ = "P_B" flC\$ = "P_C"

CLS 0

LOCATE 21, 1: PRINT "THE DATA SOURCE FILENAME WAS ", fls\$

LOCATE 23, 1: INPUT "ENTER THE SAVE FILENAME FOR CHs A,B & C< P_? IS THE FILESPEC>";

flENTERS\$

IF VAL(flENTERS\$) <> 0 THEN

endel = -1

GOTO restart

ELSEIF flENTERS\$ <> "" THEN

fls\$ = flENTERS\$ + "P_?"

END IF

fls\$ = UCASE\$(LEFT\$(fls\$, LEN(flENTERS\$) - 4) + flA\$)

ON ERROR GOTO filesaveerror

NAME fls\$ AS fls\$ ' if it exists then error 58, if not then error 53

OPEN fls\$ FOR OUTPUT AS #1

FOR j = 0 TO endel - 1

WRITE #1, buff1(j)

NEXT j

CLOSE #1

fls\$ = LEFT\$(fls\$, LEN(flENTERS\$) - 4) + flB\$

NAME fls\$ AS fls\$ ' if it exists then error 58, if not then error 53

OPEN fls\$ FOR OUTPUT AS #1

FOR j = 0 TO endel - 1

WRITE #1, buff2(j)

NEXT j

CLOSE #1

fls\$ = LEFT\$(fls\$, LEN(flENTERS\$) - 4) + flC\$

NAME fls\$ AS fls\$ ' if it exists then error 58, if not then error 53

```

OPEN #1$ FOR OUTPUT AS #1
FOR J = 0 TO endel - 1
    WRITE #1, buff3(j)
NEXT J
CLOSE #1

```

```

restart
END SUB

```

' End of Procedure

' Procedure Module for Displaying Three Channel Data

```

DEFBNG J
SUB PLOT (chA() AS INTEGER, chB() AS INTEGER, chC() AS INTEGER, endel AS INTEGER)

```

```

CLS 0
SCREEN 3
LOCATE 23, 5 PRINT "WAIT"
VIEW (10, 2)-(710, 97), 1
aver1 = 0 aver2 = 0 aver3 = 0

```

find average or DC offset of each buffer

```

FOR J2 = 0 TO endel
    aver1 = chA(J2) + aver1
    aver2 = chB(J2) + aver2
    aver3 = chC(J2) + aver3
NEXT J2
aver1 = aver1 / (J2) aver2 = aver2 / (J2) aver3 = aver3 / J2
XB = 1 XE = endel
YB = -aver1 YE = aver1
XR = XE - XB YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
LINE (XB + XR \ 4, YB)-(XB + XR \ 4, YE)
LINE (XB + XR \ 2, YB)-(XB + XR \ 2, YE)
LINE (XB + 3 * XR \ 4, YB)-(XB + 3 * XR \ 4, YE)
LINE (XB, YR \ 2)-(XE, YR \ 2)

```

' Draw Channel A Data

```

PSET (zoomx * 1, zoomy * (chA(0) - aver1))
FOR J1 = 1 TO endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chA(J1) - aver1))
NEXT J1

```

```

VIEW (10, 102)-(710, 197), 1
YB = -aver2 YE = aver2
XR = XE - XB YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
LINE (XB + XR \ 4, YB)-(XB + XR \ 4, YE)
LINE (XB + XR \ 2, YB)-(XB + XR \ 2, YE)
LINE (XB + 3 * XR \ 4, YB)-(XB + 3 * XR \ 4, YE)
LINE (XB, YR \ 2)-(XE, YR \ 2)

```

' Draw Channel B Data

```
PSET (zoomx * 1, zoomy * (chB(0) - aver2))
FOR J1 = 1 TO endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chB(J1) - aver2))
NEXT J1
```

```
VIEW (10, 202)-(710, 297), , 1
YB = -aver3: YE = aver3
XR = XE - XB: YR = YE - YB
WINDOW (XB, YE)-(XE, YB)
LINE (XB + XR \ 4, YB)-(XB + XR \ 4, YE)
LINE (XB + XR \ 2, YB)-(XB + XR \ 2, YE)
LINE (XB + 3 * XR \ 4, YB)-(XB + 3 * XR \ 4, YE)
LINE (XB, YR \ 2)-(XE, YR \ 2)
```

' Draw Channel C Data

```
PSET (zoomx * 1, zoomy * (chC(0) - aver3))
FOR J1 = 1 TO endel / zoomx
    LINE -(zoomx * (J1 + 1), zoomy * (chC(J1) - aver3))
NEXT J1
```

END SUB

' End of Procedure -----

Figure C.2 Graphic capture of the computer video screen during the analysis of the middle hole co-axial tailpipe at 1.4 kHz showing the time histories of the signals of each probe microphone and the magnitude of the Fast Fourier Transform.

' Program HEATPIPE

' Iterative Calculation of Heat Power Required per Metre of Length to Achieve the Filament
' Temperature Specified. Program is Repeated Until the Inside Surface Temperature is
' Approximately 100 Degrees Celsius Less Than the Inside Air Temperature at Equilibrium.

' Written by Alexander P. C. K. S. Lorimer, February 1990

DEFINT A-Z

' Set constants - All SI units

CONST kBrass! = 110, G! = 5.67E-08, e1! = 85, e2! = 6, Tsur! = 293, Tinf2! = 295, pi! = 3.14159

' Set semi-constants - radii are defined in inches, all others in SI

CONST h1! = 5, h2! = 15

r1! = 0.125, r2! = 0.375, r3! = 0.4375

' Convert radii in inches to metres

r1! = r1! * 0.0254, r2! = r2! * 0.0254, r3! = r3! * 0.0254

' Set Ts1 to a trial value

CLS

INPUT "Enter trial value of TS1, the filament temperature", Ts1!

' Hardcopy Output?

INPUT "Hardcopy output?", q\$

' Open Data File and Set Ts2new equal to Tinf2 as an equilibrium starting point.

OPEN "thermdat.asc" FOR OUTPUT AS #1
equilibriumstart:

Ts2new! = Tinf2!

VIEW PRINT 10 TO 15

' Start loop

DO

Ts2! = Ts2new!

' Solve equation # 1 for $q_1 = q_1/2\pi L$

$q_{1a}! = 2 * pi! * r1! * h1! * (Ts1! - (r1! / r2! * Ts1! + Ts2!) / (1 + r1! / r2!))$

$q_{1b}! = 2 * pi! * r1! * G! * (Ts1! ^ 4 - Ts2! ^ 4) / (1 / e1! + (1 - e2!) / e2! * (r1! / r2!))$

$q1! = q_{1a}! + q_{1b}!$

' Solve equation #2 for Ts#

$$Ts3! = Ts2! + q1! * \text{LOG}(r3! / r2!) / (2 * \pi! * k\text{Brass}!)$$

' Solve equation #3 for q2=q2/2piL

$$\begin{aligned} q2a! &= 2 * \pi! * r3! * h2! * (Ts3! - Tinf2!) \\ q2b! &= 2 * \pi! * r3! * e2! * G! * (Ts3! ^ 4 - Tsur! ^ 4) \\ q2! &= q2a! + q2b! \end{aligned}$$

' Solve equation #4 for Ts2 again

$$Ts2new! = Ts3! - q2! * \text{LOG}(r3! / r2!) / (2 * \pi! * k\text{Brass}!)$$

LOCATE 10

PRINT 'q1a! = ' q1a! PRINT 'q1b! = ' q1b! PRINT "q2a! = "; q2a! PRINT "q2b! = "; q2b!

LOOP UNTIL ABS(INT(Ts2! - Ts2new!)) = 0

' solve for Tinf1

$$Tinf1! = ((r1! / r2!) * Ts1! + Ts2!) / (1 + r1! / r2!)$$

VIEW PRINT PRINT "Tinf1 = "; Tinf1!

' Results

IF q\$ = " OR q\$ = "n" OR q\$ = "N" THEN GOTO screenoutput

Hardcopy output

LPRINT 'With the following parameters "

LPRINT

LPRINT " Filament temp ", Ts1!, " K"

LPRINT " Ambient temp ", Tinf2!, " K"

LPRINT " Room Wall temp ", Tsur!, " K"

LPRINT " Filament emissivity", e1!

LPRINT " Brass emissivity", e2!

LPRINT " Brass Thermal Conductivity ", kBrass!, " W/mK (at 300K)"

LPRINT " Heater Radius ", r1!, "m"

LPRINT " Pipe Inside Radius ", r2!, " m"

LPRINT " Pipe Outside Radius ", r3!, " m"

LPRINT

LPRINT "The inside surface temperature of the pipe is", Ts2!, " K"

LPRINT "The average temperature of the air inside the pipe is", Tinf1!, " K"

LPRINT "Qconvective1=", q1a!, "Qradiative1=", q1b!, "(W/m)"

LPRINT "Qconvective2=", q2a!, "Qradiative2=", q2b!, "(W/m)"

LPRINT "Q1 must equal Q2 at steady state"

LPRINT "The required heat input per metre is " (q1! + q2!) / 2, " W/m"

LPRINT

LPRINT

Print Results on VDT

screenoutput:

```
VIEW PRINT.CLS
PRINT "With the following parameters "
PRINT " Filament temp .", Ts1!, " K"
PRINT " Ambient temp .", Tinf2!, " K"
PRINT " Room Wall temp.", Tsurf!, " K"
PRINT " Filament emissivity", e1!
PRINT " Brass emissivity:", e2!
PRINT " Brass Thermal Conductivity", kBrass!, " W/mK ( at 300K)"
PRINT " Heater Radius:", r1!, "m"
PRINT " Pipe Inside Radius:", r2!, " m"
PRINT " Pipe Outside Radius:", r3!, " m" PRINT
PRINT "The inside surface temperature of the pipe is ", Ts2!, " K"
PRINT "The average temperature of the air inside the pipe is ", Tinf1!, " K"
PRINT "Qconvective1=", q1a!, "Qradiative1=", q1b!, "(W/m)"
PRINT "Qconvective2=" q2a!, "Qradiative2=" q2b!, "(W/m)"
PRINT "(Q1 must equal Q2 at steady state)"
PRINT "The required heat input per metre is . (q1! + q2!) / 2, " W/m"
```

fileoutput

```
PRINT #1, "With the following parameters " PRINT #1
PRINT #1, " Filament temp .", Ts1!, " K"
PRINT #1, " Ambient temp .", Tinf2!, " K"
PRINT #1, " Room Wall temp.", Tsurf!, " K"
PRINT #1, " Filament emissivity", e1!
PRINT #1, " Brass emissivity:", e2!
PRINT #1, " Brass Thermal Conductivity ", kBrass! " W/mK ( at 300K)"
PRINT #1, " Heater Radius:", r1!, "m"
PRINT #1, " Pipe Inside Radius:", r2!, " m"
PRINT #1, " Pipe Outside Radius:", r3!, " m" PRINT #1.
PRINT #1, "The inside surface temperature of the pipe is ", Ts2!, " K"
PRINT #1, "The average temperature of the air inside the pipe is ", Tinf1!, " K"
PRINT #1, "Qconvective1=", q1a!, "Qradiative1=", q1b!, "(W/m)"
PRINT #1, "Qconvective2=", q2a!, "Qradiative2=", q2b!, "(W/m)"
PRINT #1, "Q1 must equal Q2 at steady state"
PRINT #1, "The required heat input per metre is . (q1! + q2!) / 2 " W/m
PRINT #1, PRINT #1,
```

```
BEEP. BEEP
SLEEP 6
```

Repeat until Filament Temperature is 1200

```
Ts1! = Ts1! + 100
IF Ts1! < 1200 THEN GOTO equilibriumstart
CLOSE #1
```

```
END
```

APPENDIX D: EXHAUST SYSTEM PERFORMANCE DATA

"Free-field" Measurements

The sound pressures recorded by the microphone in the anechoic chamber are presented in Figure D.1 (at room temperature), D.2 (at 230°C), and D.3 (at 265°C). Each figure shows the performance of the muffler and various tailpipes (a) is the simple tailpipe, b) is the middle-hole co-axial tailpipe, c) is the top-hole co-axial tailpipe, and d) is the two-hole co-axial tailpipe) against that of the simple tube exhaust. (The insertion loss comparisons of each system referenced to the simple tube exhaust are found in Figures 5.3 through 5.5). Following Figure D.1 through D.3, the insertion losses of each co-axial system referenced to the muffler and simple tailpipe system are presented for the three temperature conditions in figures D.4 through D.6.

Probe-tube Measurements

Following the figures of the anechoic chamber measurements, the relative responses (amplitude and phase) of the probe tube microphones under the conditions of middle-hole (Figure D.7) and two-hole (Figure D.8) co-axial tailpipe at room temperature are presented for frequencies ranging from approximately 500 Hz up to 2700 Hz.

a) Muffler with simple tailpipe

b) Muffler with middle-hole co-axial tailpipe

Figure D.1: Sound pressures of exhaust systems recorded in the anechoic chamber at room temperature. *-continued...*

c) Muffler with top-hole co-axial tailpipe

d) Muffler with top-and-bottom-hole co-axial tailpipe

Figure D.1: Sound pressures of exhaust systems recorded in the anechoic chamber at room temperature. *-continued...*

a) Muffler with simple tailpipe

b) Muffler with middle-hole co-axial tailpipe

Figure D.2: Sound pressures of exhaust systems recorded in the anechoic chamber at 230°C. -continued...

c) Muffler with top-hole co-axial tailpipe

d) Muffler with top-and-bottom-hole co-axial tailpipe

Figure D.2: Sound pressures of exhaust systems recorded in the anechoic chamber at 230°C. -continued...

a) Muffler with simple tailpipe

b) Muffler with middle-hole co-axial tailpipe

Figure D.3: Sound pressures of exhaust systems recorded in the anechoic chamber at 265°C. *-continued...*

c) Muffler with top-hole co-axial tailpipe

d) Muffler with top-and-bottom-hole co-axial tailpipe

Figure D.3: Sound pressures of exhaust systems recorded in the anechoic chamber at 265°C. *-continued...*

Figure D.4: Insertion Losses of muffler with co-axial tailpipe systems referenced to the muffler with simple tailpipe system at room temperature.

Figure D.5: Insertion Losses of muffler with co-axial tailpipe systems referenced to the muffler with simple tailpipe system at 230°C.

Muffler & Two-hole Co-ax. IP, 265 C

Figure D.6: Insertion Losses of muffler with co-axial tailpipe systems referenced to the muffler with simple tailpipe system at 265°C.

a) Amplitude and Phase for Middle Hole at 600 Hz

b) Amplitude and Phase for Middle Hole at 700 Hz

c) Amplitude and Phase for Middle Hole at 1100 Hz

Figure D.7: Relative responses of probe tube microphones for various frequencies at room temperature for the middle-hole co-axial tailpipe. - *continued...*

d) Amplitude and Phase for Middle Hole at 1500 Hz

e) Amplitude and Phase for Middle Hole at 1600 Hz

f) Amplitude and Phase for Middle Hole at 1700 Hz

Figure D.7: Relative responses of probe tube microphones for various frequencies at room temperature for the middle-hole co-axial tailpipe. - *continued...*

Figure D.7: Relative responses of probe tube microphones for various frequencies at room temperature for the middle-hole co-axial tailpipe. - *continued...*

j) Amplitude and Phase for Middle Hole at 2600 Hz

k) Amplitude and Phase for Middle Hole at 2700 Hz

Figure D.7: Relative responses of probe tube microphones for various frequencies at room temperature for the middle-hole co-axial tailpipe.

a) Amplitude and Phase for Top and Bottom Hole at 534 Hz

b) Amplitude and Phase for Top and Bottom Hole at 700 Hz

c) Amplitude and Phase for Top and Bottom Hole at 805 Hz

Figure D.8: Relative responses of probe tube microphones at three frequencies at room temperature for the axial tailpipe. - *continued...*

d) Amplitude and Phase for Top and Bottom Hole at 900 Hz

e) Amplitude and Phase for Top and Bottom Hole at 1100 Hz

f) Amplitude and Phase for Top and Bottom Hole at 1400 Hz

Figure D.8: Relative responses of probe tube microphones for various frequencies at room temperature for the top-and-bottom-hole co-axial tailpipe. - *continued...*

g) Amplitude and Phase for Top and Bottom Hole at 1500 Hz

h) Amplitude and Phase for Top and Bottom Hole at 1600 Hz

i) Amplitude and Phase for Top and Bottom Hole at 1700 Hz

Figure D.8: Relative responses of probe tube microphones for various frequencies at room temperature for the top-and-bottom-hole coaxial tailpipe. - *continued...*

j) Amplitude and Phase for Top and Bottom Hole at 1900 Hz

k) Amplitude and Phase for Top and Bottom Hole at 2000 Hz

l) Amplitude and Phase for Top and Bottom Hole at 2100 Hz

Figure D.8: Relative responses of probe tube microphones for various frequencies at room temperature for the top-and-bottom-hole coaxial tailpipe. - *continued...*

m) Amplitude and Phase for Top and Bottom Hole at 2200 Hz

n) Amplitude and Phase for Top and Bottom Hole at 2400 Hz

o) Amplitude and Phase for Top and Bottom Hole at 2500 Hz

Figure D.8: Relative responses of probe tube microphones for various frequencies at room temperature for the top-and-bottom-hole coaxial tailpipe. - *continued...*

p) Amplitude and Phase for Top and Bottom Hole at 2600 Hz

q) Amplitude and Phase for Top and Bottom Hole at 2700 Hz

Figure D.8: Relative responses of probe tube microphones for various frequencies at room temperature for the top-and-bottom-hole coaxial tailpipe.

APPENDIX E: APPARATUS: PERFORMANCE DATA AND SKETCHES

The following pages contain the assembly sketches of the anechoic chamber, as well as the transformer, transition, and exhaust system apparatus from the cavity muffler (or reference tube) to the flanged termination of the tailpipe. The full set of printouts from the anechoic chamber impulse response evaluation are shown. The circuit diagram of the microphone pre-amplifier is given, as is the circuit diagram of the electrical filter. The measured frequency response of the filter is also presented. Finally, the frequency response of the compression driver mounted in a plane-wave tube (as provided by the manufacturer) is included.

The time history of the impulse response shows that there are no significant reflections greater than at least 30 dB below the input energy. There is a low frequency wave evident (< 50 Hz) which may correspond to a chamber resonance due to the tailpipe termination opening in the ceiling which was not sealed during the procedure. Alternately, this may simply be low frequency noise because it is evident earlier in time than the impulse. The coherence function indicates that the impulse response was sufficiently coherent from 500 Hz to greater than 74.6 kHz, and the frequency response function indicates that there is sufficient energy in the input, at least at 1 kHz and greater.

The electrical filter response is seen to follow closely the predicted response (found in Appendix B).

Figure E.1: Anechoic Chamber Placement

Figure E.2: Anechoic Chamber Assembly

Figure E.3: Acoustic Transformer

Figure E.4: Acoustic Transition

Figure E.5a: Reference Tube Exhaust System

3

of/de

3

**PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT**

PRECISIONSM RESOLUTION TARGETS

PIONEERS IN METHYLENE BLUE TESTING SINCE 1974

1880 COUNTY ROAD 5, BURNVILLE, N.Y. 10814 USA
TEL: 512 426 7867 FAX: 512 426 7867 TLA: 0 00000000

Figure E.5b: Cavity Muffler Exhaust System

Figure E.5c: Reference Exhaust Pipe with Flanged Ends

Figure E.5d: 4 Inch Cavity Muffler Top Plate

Figure E.5e: 4 Inch Cavity Muffler Flanged Casing

Figure E.5f: 4 Inch Cavity Muffler Bottom Plate

Figure E.5g: 4 Inch Cavity Muffler Bottom Plate, View-AA

Figure E.5h: Outer Tube of Co-axial Tailpipe

Figure E.5i: Base Flange (Termination) and Inner Tailpipe

Figure E.5j: Split Outer Sleeve of Co-axial Tailpipe Inner Tube

W11 IMPULSE RESP ████████ INPUT MAIN Y: 53.24B
 F: 35.04B 404B X: -54.626ms
 K: 0.000ms + 31.3ms
 SETUP W11 4A: 512x

IMPULSE, REFLECTIONS OF CHAMBER

IMPULSE: 0
 W11 IMPULSE RESP MAG MAIN Y: 53.24B
 F: 35.04B 404B X: -54.626ms
 K: -59.753ms + ████████
 SETUP W11 4A: 512x

IMPULSE, REFLECTIONS OF CHAMBER

Figure E.7: Magnitude of impulse response of the anechoic chamber

Figure E.8: Coherence and frequency response of the signal measured in the anechoic chamber compared to the source electrical signal.

Figure E.9: Electrical circuit diagram of 500 Hz 4-pole High-Pass Filter.
 $R = 250 \, \Omega$ and $C = 11.5 \, \mu\text{F}$.

Amplitude Response of 4-Pole High-Pass Filter

Figure E.10: Frequency response of 500-Hz High Pass Electronic Filter.

Notes:

Microphone Pins 2 & 3 are shorted together

E is the output signal

Vcc & Vee are DC power supply

R1 = 5.1 kOhm C1 = 0.1 μ F

R2 = 82.5 kOhm C2 = 0.01 μ F

R3 = 390 Ohm C3 = 10 μ F

R4 = 10 kOhm

R5 = 628 Ohm

R6 = 523 Ohm

Figure E.11: Electronic circuit diagram for the microphone pre-amplifier.

The probe microphone response is presented in the following figures. The probe tube microphone and a Brüel and Kjær 1/2" precision microphone were connected to a Brüel and Kjær intensity probe calibrator. This apparatus is configured to expose two microphones to identical sound fields. The signal was a pseudo-random noise introduced into the calibrator. The first figure, E.12, presents the response of the probe tube microphone with a 13 cm (5") probe and no damping added compared to the 1/2" microphone. The second figure, E.13, presents the response of the probe tube microphone damped with 22 x 0.1 mm (0.004") diameter zinc wires in the tubes. The final figure, E.14, presents the difference between the damped and undamped responses, or the insertion loss of the damping wires.

The final figure, E.15, is taken from the product literature on the JBL-2445J compression driver. The plane-wave terminated tube response was used as a basis for comparison between other makes and models.

Figure E.12: Frequency response of undamped 13 cm probe tube compared to a 1/2 inch microphone.

Figure E.13: Frequency response of 13 cm probe tube damped with 22 x 0.1 mm diameter wires compared to a 1/2 inch microphone.

Figure E.14: Insertion loss of 22 x 0.1 mm diameter wires in 13 cm probe tube.

Frequency response and impedance modulus of Model 2445J coupled to a 49 mm (2 in) diameter terminated plane-wave tube with sensitivity referenced to a 25 mm (1 in) tube. This is the power response of the transducer, and is the frequency response that will be obtained on a true full range constant directivity horn design such as JBL's 2360 series of Constant Coverage Bi Radial Horns.

Figure E.15: Plane-wave tube frequency response of the JBL 2445J compression driver.

REFERENCES

- [1] M.L. Munjal, *Acoustics of Ducts and Mufflers*, John Wiley and Sons, Toronto, 1987, ch. 1-2
- [2] A.D Jones, "Modelling the Exhaust Noise Radiated from Reciprocating Internal Combustion Engines - A Literature Review", *Noise Control Engineering Journal*, **23** (1), pp. 12-31, 1984
- [3] C.O. Reinhart and J.E. Kleinhenz, "Analytical Techniques used in Design of Low Back Pressure Mufflers", *Proceedings of Noise-Con-81*, pp. 111-116, 1981
- [4] A.J. Brammer & S.E. Keith, conversations with reference to co-axial tailpipes for single-cylinder engines
- [5] L.L. Beranek (ed.), *Noise and Vibration Control*, Institute of Noise Control Engineering, Washington, D.C., 1988, ch. 9, 10, & 12
- [6] J.D. Irwin & E.R. Graf, *Industrial Noise and Vibration Control*, Prentice Hall, Englewood Cliffs, New Jersey, 1979, ch. 7 & 8
- [7] M.G. Prasad and M.J. Crocker, "Evaluation of four-pole parameters for a straight pipe with a mean flow and a temperature gradient", *Journal of the Acoustical Society of America*, **69**(4), 916-921, (1981)
- [8] M.G. Prasad and M.J. Crocker, "Acoustic Modelling of Automotive Exhaust Systems", *Proceedings of Noise-Con-81*, 93-98, 1981
- [9] J. Okda, "Performance of Reactive Mufflers and Calculations of Engine Exhaust Noise", *Proceedings of Internoise-75*, pp 655-662, 1975
- [10] L.L. Beranek, *Acoustics*, American Institute of Physics, New York, N.Y., 1986, Ch. 4 & 9
- [11] Lord Rayleigh, *The Theory of Sound*, 2nd Ed, MacMillan, London, 1894, Art. 307-314
- [12] H.J.J. Braddick, *Vibrations, Waves and Diffraction*, McGraw-Hill, Toronto, 1965, p 16
- [13] A.V. Sreenath & M.L. Munjal, "Evaluation of noise attenuation due to exhaust mufflers", *Journal Sound and Vibration*, **12**(1), 1-19 (1970)

REFERENCES

continued

- [14] A.J. Brammer, "Non-linear effects in the generations of sound by internal combustion engine exhausts," *Journal of the Acoustical Society of America*, **65**(5), 45 (1979)
- [15] E.F. Obert, *Internal Combustion Engines and Air Pollution*, Harper and Row, New York, 1973, Ch. 1
- [16] Visual inspection of anechoic chambers at the NRC
- [17] C.M. Harris, *Shock and Vibration Handbook*, McGraw-Hill, Toronto, 1988, Ch. 35, pp 15-17
- [18] R.C. Weast (ed), *CRC Handbook of Chemistry and Physics*, CRC Press, Boca Raton, Fla, 1981, pp. E-398-399
- [19] F.P. Incropera & D.P. DeWitt, *Introduction to Heat Transfer*, John Wiley and Sons, Toronto, 1985, ch. 3 & 6
- [20] H.L. Harrison & J.G. Bollinger, *Introduction to Automatic Controls*, Harper and Row, New York, N.Y., 1969, pp. 78-80
- [20] W.H. Press, B.P. Flannery, S.A. Teukolsky, W.T. Vetterling, *Numerical Recipes*, Cambridge University Press, New York, 1986, Chapter 12,
- [21] H. Herlufsen, "Dual Channel FFT Analysis (Part I)", *Technical Review*, Brüel and Kjær, Denmark, No. 1, 1984

END

24-1 2-92

FIN