Investigation into relative temperature measurement of pulsed constrained gas flow using passive acoustic means

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Abstract

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The requirement to measure the real time, dynamic temperature of exhaust system gases is becoming more and more important in the areas of aeronautics, automotive (cars, trucks, etc), marine and industrial/environmental applications, in particular on a cycle-by-cycle (CBC) basis. Monitoring exhaust gas temperatures of any power-plant can give important diagnostic information for the monitoring of fuel mixture, combustion efficiency etc. This 'diagnostic' information can be used to help 'dynamically' tune the combustion process or engine operation to assist in reducing exhaust emissions, optimize fuel flow rates and help improve overall efficiency of the system or engine.

In order to realize this capability, it is necessary to monitor the dynamic temperature of the exhaust flows with a temporal resolution in the millisecond range. The fastest response time achievable by currently available temperature sensors which are robust enough to survive in these very hostile environments is of the order of several hundred milliseconds or greater. To this end, conventional measurement systems such as resistive-temperature sensitive devices, bi-metallic solutions or even optical solutions are orders of magnitude too slow to be of any use. In order to achieve the temporal resolution required, a new measurement methodology has to be developed.

This work presents a novel temperature measurement sensor methodology, based on passive acoustic tone generators and specifically the labial flue pipe, that respond to the constrained pressure pulses generated by a pressure release mechanism in an exhaust or blow-down system. The work includes the development of a unique software suite that allows optimal tone generator design parameters be determined and identifies a unique interaction between the Ising efficiency number and the Reynolds number pertaining to the jet mechanism of the tone generator. Experimental results presented indicate a sensitivity to tone generator oscillation frequency, with pre-oscillation of the tone generators shown to further enhance this sensitivity.
Declaration

I hereby declare that, except where otherwise indicated, this thesis is entirely my own work and had not been submitted in whole or in part to any other university. Some of this research has been published elsewhere by the author and where the work of others is used it has, to the authors best knowledge, been fully referenced.

Signature:

_______________________________
Brian Joseph Moss
Dedication

To my Parents and Family
Acknowledgments

I would like to acknowledge the help and support of a number of people.

My supervisors, Prof. Elfed Lewis, Dr. Gabriel Leen and Dr. Andrew Niven for their patient academic guidance, and allowing me the freedom to independently pursue my research interests.

Paddy Kelly, Paddy O’Donnell, Joe Leen, Jim Ryan and Kort Bremer for their wisdom, expertise and guidance with theoretical, mechanical and construction elements of the various apparatus used in this research. To Michael Johnson for his impressive knowledge of LabVIEW, and tales of far off places.

Thanks also to Damien McCartney and Tracey Mullins of Analog Devices, Limerick for their help and support during this research.

Finally, a special thanks to my parents Pat and Mary, wife Eileen and son Oisin, for all their understanding, patience, support, and distraction.
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<th>Definition</th>
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<tbody>
<tr>
<td>$a_0$</td>
<td>Acoustic velocity in ambient or undisturbed gas</td>
</tr>
<tr>
<td>$a$</td>
<td>Instantaneous specific Acoustic velocity in the travelling wave</td>
</tr>
<tr>
<td>$T_0$</td>
<td>Undisturbed gas temperature</td>
</tr>
<tr>
<td>$T$</td>
<td>Instantaneous specific temperature in the travelling wave</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant ($R_{air} = 287 \text{ J} \cdot \text{Kg}^{-1} \cdot \text{K}^{-1}$)</td>
</tr>
<tr>
<td>$R_0$</td>
<td>Universal gas constant ($R_{0,air} = 8.314 \text{ J} \cdot \text{mol}^{-1} \cdot \text{K}^{-1}$)</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius of the neck in a Helmholtz resonator</td>
</tr>
<tr>
<td>$R_c$</td>
<td>Radius of the cavity in a Helmholtz resonator</td>
</tr>
<tr>
<td>$k_1$</td>
<td>Constant dependant on gas (see Blair, p171)</td>
</tr>
<tr>
<td>$k_2$</td>
<td>Constant dependant on gas (see Blair, p171)</td>
</tr>
<tr>
<td>$k_3$</td>
<td>Constant dependant on gas (see Blair, p171)</td>
</tr>
<tr>
<td>$d$</td>
<td>Linear distance measured in metres</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency</td>
</tr>
<tr>
<td>$y$</td>
<td>Attenuation</td>
</tr>
<tr>
<td>SPL</td>
<td>Sound Pressure Level</td>
</tr>
<tr>
<td>$W$</td>
<td>Velocity of the shock wave</td>
</tr>
<tr>
<td>$u_p$</td>
<td>Particle mass velocity in the travelling wave</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
</tr>
<tr>
<td>$S$</td>
<td>Cross-sectional Area</td>
</tr>
<tr>
<td>$L$</td>
<td>Length</td>
</tr>
<tr>
<td>$A$</td>
<td>Cross-sectional area of the cavity of the labial flue pipe</td>
</tr>
<tr>
<td>$B$</td>
<td>Cross-sectional area of the mouth of the labial flue pipe</td>
</tr>
<tr>
<td>$H$</td>
<td>Length of the mouth in a labial flue pipe</td>
</tr>
<tr>
<td>$w$</td>
<td>Width of the cavity in the labial flue pipe</td>
</tr>
<tr>
<td>$h$</td>
<td>Height of the cavity in the labial flue pipe</td>
</tr>
<tr>
<td>$I$</td>
<td>Ising number</td>
</tr>
<tr>
<td>AFR</td>
<td>Air-Fuel-Ratio</td>
</tr>
<tr>
<td>EGT</td>
<td>Exhaust Gas Temperature</td>
</tr>
<tr>
<td>k-Type</td>
<td>Thermocouple designation. Used for temperature measurement</td>
</tr>
<tr>
<td>CBC (cbc)</td>
<td>Cycle-by-Cycle</td>
</tr>
<tr>
<td>$K$</td>
<td>Kelvin. Temperature measurement unit</td>
</tr>
<tr>
<td>Bar (bar)</td>
<td>Unit of Pressure. 1 bar = 100000 Pa</td>
</tr>
<tr>
<td>Pa</td>
<td>Unit of pressure. 1 ISA atmosphere = 101325 Pa</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>ρ</td>
<td>Density.</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity, constant pressure</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Specific heat capacity, constant volume</td>
</tr>
<tr>
<td>γ</td>
<td>Specific heat capacity ratio, i.e. $\frac{c_p}{c_v}$</td>
</tr>
<tr>
<td>ISA</td>
<td>International Standard Atmosphere</td>
</tr>
<tr>
<td>OED</td>
<td>Oxford English Dictionary</td>
</tr>
<tr>
<td>TG</td>
<td>Tone Generator</td>
</tr>
<tr>
<td>RPM (rpm)</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>QUB</td>
<td>Queens University Belfast</td>
</tr>
<tr>
<td>SSR</td>
<td>Solid State Relay</td>
</tr>
<tr>
<td>TOF</td>
<td>Time of Flight</td>
</tr>
<tr>
<td>JND</td>
<td>Just Noticeable Difference</td>
</tr>
<tr>
<td>WST</td>
<td>Wind Sheet Thickness</td>
</tr>
<tr>
<td>Op-amps</td>
<td>Operational amplifiers</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>P</td>
<td>Driver Pressure or instantaneous specific pressure in travelling wave</td>
</tr>
<tr>
<td>$P_0$</td>
<td>Used to indicate ambient or undisturbed Pressure (Pa or bar)</td>
</tr>
<tr>
<td>vi</td>
<td>Virtual Instrument (LabVIEW)</td>
</tr>
<tr>
<td>IC</td>
<td>Internal Combustion</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>PC</td>
<td>Personal Computer</td>
</tr>
<tr>
<td>DSP</td>
<td>Digital Signal Processing</td>
</tr>
</tbody>
</table>
CHAPTER 1

1.1 Introduction and Motivation

Temperature monitoring of internal combustion engine exhaust gases can offer critical diagnostic information for the control of the combustion process in terms of optimizing the Air-Fuel-Ratio (AFR). Such information can be critical in maximizing economic return while minimizing environmental contaminants. Currently, exhaust gas temperature (EGT) monitoring is done through the use of thermocouple EGT sensors. The thermocouple is a device based on the seebeck effect whereby two dissimilar metals that are joined at one end, produce a non-linear voltage output in relation to temperature. Thermocouples come in various formats conforming to the ITS-90 [1] standard.

EGT monitoring provides the ability to optimize for stoichiometric or ideal burn rates with the consequential benefits of increasing fuel economy and reducing environmental contaminants. Optimum burn rates can also benefit engine lifetime by reducing stress due to over heating or spark-plug fouling due to the build-up of un-burnt carbon. Ideally such temperature measurements should be performed and processed on a cycle-by-cycle (CBC) basis for each combustion event, however, this is well beyond the capability of current technology where the fastest response time is of the order of hundreds of milliseconds to seconds.

With an economic and environmental justification for EGT measurement understood and with current technology being unable to match typical CBC rates, attention is turned to the use of acoustics as a means of measuring temperature. The justification for
using acoustics as a means to measure temperature is that each air-fuel mixture combustion event in an engine produces a finite duration pressure pulse that propagates into an exhaust or blow-down pipe. It was proposed that these pressure waves could be used, in conjunction with appropriate acoustic sensors, to determine the local temperature in the blow-down pipe, i.e. the exhaust gas temperature.

1.2 Overview

For the purposes of this research an apparatus incorporating pressure cylinder, exhaust or blow-down pipe and acoustic transducers was developed to induce and detect pressure waves at room or elevated temperatures.

Chapter 2 discusses the history of sound and its connection/relation to temperature. Chapter 3 introduces the best known temperature scales that have evolved over time and discusses briefly different temperature measurement devices including active acoustic pyrometric methodologies. Chapter 4 introduces the pressure wave and its propagation into a constrained conduit, and introduces the mechanism by which acoustic transducers would react to such a pressure wave. The key criteria required of an acoustic transducer in such an environment are also discussed in chapter 4. Chapter 5 introduces the Helmholtz resonator as a potential acoustic transducer in light of the criteria hitherto outlined. Chapter 6 introduces the labial flue pipe and describes a unique software suite to assist in its design. The efficiency of the labial flue pipe jet mechanism is discussed with experimental frequency spectrum plots presented showing the results of efficient or optimally blown pipes versus inefficient or over-blown pipes. Moreover, a novel correlation is made between the efficiency of the jet mechanism, the Reynolds number associated with the jet mechanism and the acoustic response from the labial flue pipe.
Chapter 7 introduces the hardware of the experimental apparatus, including the pressure cylinder, pressure release mechanism, pressure sensors and signal processing. Chapter 8 presents various experimental analysis methodologies and the results gathered. Chapter 9 presents a summary and conclusion of the findings from the previous chapters.

1.3 Contribution

This research involved in this thesis has resulted in several conference presentations and peer reviewed publications.


CHAPTER 2

2.1 What is sound?

The use of acoustics in temperature is not a new phenomenon and the relationship between temperature and acoustic velocity has been known for quite some time. However, this was not always the case and so a brief investigation of the history of sound and how it relates to temperature is warranted.

Rossing [2] states that the term sound describes two different things:

1. An auditory sensation in the ear
2. The disturbance in a medium, which can cause this sensation.

The Oxford English Dictionary [3] gives a definition of sound similar to Rossing as:

“vibrations that travel through the air or another medium and can be heard when they reach a person’s or animal’s ear”

Beranek [4] (1954, p.3) goes a stage further in the definition of sound by incorporating elasticity:

“a disturbance propagated through an elastic material causes an alteration in pressure or a displacement of the particles of the material which can be detected by a person or an instrument”.

Massey [5] (2007, p493) gives a definition of sound which incorporates pressure and density as:

“If the pressure at a point in a fluid is altered, the density is also altered – even if only slightly – and in consequence individual particles undergo small changes in position. To maintain a continuum, adjacent particles also change position and thus the new pressure is progressively, yet rapidly, transmitted through the rest of the fluid”
From these descriptions it is clear that sound is an event that causes pressure perturbations to propagate through a medium with that medium being of a substantial nature, i.e. not a vacuum.

But this was not always the understanding: Aristotle’s [6] (384-322 B.C.) view that air was necessary for the propagation of sound was often discounted because air did not appear to be affected by the propagation of sound. Lindsay [7] and Caleon et al [8], who references Blood [9] states that Pierre Gassendi; in the mid 17th century considered sound to be a thin stream of invisible particles emanating from the source and able to affect the ear. Blood [9] writes that this counter Aristotelian view was further promulgated by Otto von Guericke, known as the father of vacuum physics, who noted that sound travelled better in still air than in disturbed. Indeed Lindsay [7, 10] writes that Guericke further discounted the idea that air is needed for sound propagation following his “ringing bell in a vacuum jar” experiment. Von Guericke’s problem was that the bell was mechanically coupled to the jar with the sound being heard a result of coupling through the direct connection between the bell and the jar, a substantial medium. Athanasius Kircher [11] conducted a similar flawed experiment to Guericke’s, and like Guericke concluded that air was not necessary for the conduction of sound. However, once again the problem was insufficient decoupling between the bell and the jar, Lindsay [10] (1966, p. 635). It was not for some years after the experiments of Kircher and Guericke that in 1660 Robert Boyle contradicted the anti-Aristotelian view of sound propagation by showing that air is indeed necessary for sound transmission, i.e. his famous ‘Ticking clock in a vacuum’ experiment; Lindsay [7], West [12], Pierce [6], Caleon et al. [8]. Indeed, an extension of Boyle’s and for that matter Guericke’s and
Kircher’s experiments, is that sound propagates through any material that has substance, or as Massey (2007, p.493) states:

“This speed (of the pressure change) is determined by the relation between changes of pressure and changes of density, that is, by the elastic properties of the fluid”

Returning to Beranek’s ‘elastic’ medium, air is indeed elastic as experienced by every cyclist or driver, i.e. tyres. Tyres, typically filled with air under pressure, absorb a lot of the shocks of moving over the various perturbations that may be found on cycling or driving surfaces. The very fact that air can be pumped into and compressed in the tyre is evidence that it is elastic.
2.2 What of Acoustics and how does it relate to Sound?

The Oxford English dictionary [3] defines sound as:

“vibrations that travel through the air or another medium and can be heard when they reach a person’s or animal’s ear”.

Acoustics on the other hand is defined by Beranek[4] as:

“...intimately associated with sound waves or with the individual media, phenomena, apparatus, quantities, or units discussed in the science of sound waves.”

The Oxford English Dictionary defines acoustics in perhaps a slightly more readable format as:

“the branch of physics concerned with the properties of sound”.

Pierce [6] describes Marin Mersenne (1588-1648), whom Descartes described as “homo omnigenae sed indigestae eruditionis” (translated as “all kinds of confused, but a man of learning”), as the father of acoustics following his publication in 1627 of “Traites de l’harmonie universelle” otherwise known as “Harmonie Universelle” [13]. However, it is Joseph Sauveur [14] who is attributed with coining the phrase “Acoustique”, Lindsay (1945, p. xiv). Acoustics was coined by Sauveur from the Greek word “ακουστός” meaning “able to be heard”, as a general term for his research into the relationship between musical pitch, frequency and harmonics both with organ pipes and vibrating strings, Peirce [15] (1836 p. 78). Since then, the term ‘Acoustics’ has become readily associated with the science of sound and its propagation.
2.3 Isothermal or Adiabatic?

Following on from Boyle’s experiments in which it was confirmed that air was indeed necessary for sound transmission, investigations turned to trying to determine how sound propagated in different materials and specifically how fast. Peirce [6] and Rossing et al. [16] indicate that Jean Baptiste Biot made experiments regarding the speed of sound in solids, specifically 1km long iron pipes [17] and determined the acoustic velocity in the iron pipes to be approximately 10.5 times that in air, Peirce (1836, p 58). Colladon et al. [18] made experiments into the speed of sound in water, specifically in the waters of Lake Geneva. But with regards the speed of sound in air, people like Pierre Gassendi, Marin Mersenne, Sir Isaac Newton, Giovanni Borelli and Vincenzo Viviani amongst others made various experiments with differing levels of success. For instance, Gassendi determined the speed of sound in air as 478.43 m·s⁻¹ [19], which by today’s standard is a long way off. However, Gassendi did refute Aristotle’s view that the speed of sound was related to the pitch of the transmitted tone: Aristotle claimed that high frequencies travelled faster than low frequencies, despite the fact that he was unable to explain why a sound, specifically from a musical concert, was still coherent at varying distances. Marsenne determined the speed of sound to be 448.22 m·s⁻¹, Springer [16] and Lenihan [20], but it was the experiments of Borelli and Viviani that gave the most accurate speed of sound for the time at 349.8 m·s⁻¹, Miller [21].

Newton tried to develop a model for acoustic propagation by associating it with the simple harmonic motion of a pendulum, Lindsay [7] (1945, p xix). Lindsay [7] states that Newton’s harmonic approach yielded an acoustic velocity of approximately 288 m·s⁻¹ (945 ft/s). Newton’s theoretical results were somewhat divergent from the best
experimental results at the time, such as those obtained by William Derham [22], who from observation, determined that acoustic velocity was dependant upon temperature. Peirce [15] (1836, p.5) states that Derham calculated the acoustic velocity in air to be 348 m⋅s\(^{-1}\) (1142 ft/s). Blood [9] states that Newton tried to match his theoretical model to Derham’s experimentally derived result by claiming that water vapour was responsible for the discrepancy between the values, which to quote Lindsay [7] (1945, p.XIX) was a:

“…..rather specific and arbitrary assumption…”

In fact some commentators on the subject, such as Broad et al. [23] and Goodstein [24] have called Newton’s solution a scientific fraud and deceit. Lindsay (1945, pXX) says that:

“His (Newtons) explanation was so obviously ad hoc that it should have failed to carry conviction”

Blood [9] is equally critical in his view of Newtons solution by saying that:

“Newton's analysis was incredibly ad hoc.”

Such comments however, may have been unwarranted as there was one serious omission in all the theoretical analysis up to this time: no-one, other than Derham’s observational results, had accounted for temperature in any of the experiments. In fact Gassendi, despite refuting Aristotle’s claims on sound propagation versus frequency even discounted the effects of wind on sound propagation.

It was not until the experiments by G. Branconi that the relationship between acoustic velocity (Speed of Sound) and temperature was finally identified. Branconi measured the speed of sound (referenced to atmospheric conditions of 0°C and 1 atmosphere or
101325Pa) as being 332 m s\(^{-1}\), Lenihan [25]. However, it was not known how temperature played a part in acoustic velocity until in 1816 Laplace [26] suggested that the air through which the sound was propagating was not operating isothermally, i.e. one in which the temperature remains constant, but rather was acting adiabatically. This means that due to work being done by the sound wave, i.e. in the compression and/or expansion of the local environment, there is a change, albeit small, in local temperature, Lindsay [7] (1945 p xx) [10]. Laplace reasoned that these small temperature changes, due to the compressions and expansions of the gas, caused the elasticity of the gas to change, or as Beranek (1954, p.19) states:

“…whenever a portion of any gas is compressed rapidly, its temperature rises, and, conversely, when it is expanded rapidly, its temperature drops. At any point therefore in an alternating sound field, the temperature rises and falls relative to the ambient temperature. This variation occurs at the same frequency as that of the sound wave and is in phase with the sound pressure”.

Therefore the compression or expansion of the gas by the propagation of a sound wave causes the molecules of the gas to either get closer together or further apart. This causes the molecules to interact to a greater or lesser extent thereby causing a localised increase or decrease in temperature due to increased or decreased molecular interactions. There is no heat transfer outside the system, i.e. an adiabatic system, Lindsay [10] (1966 p. 637). This explains why the isothermal approach hitherto undertaken appeared to make sense; air, as a macro entity does not heat up when a sound passes through it. Laplace, determined that the elasticity of the gas is proportional to the ratio of the specific heats, \(c_p\) and \(c_v\) of the gas, or to quote Lindsay [7]:
“the compressions and rarefactions follow an adiabatic law in which the changes in temperature lead to a higher value of elasticity, namely, the product of the pressure by the ratio, $\gamma$ of the two specific heats of the air”.

The specific heat, $c$, of a substance is defined by Roshko et al. [27] as the “…heat needed to raise the temperature of a unit mass of the system by 1 degree”.

Laplace used the specific heats at constant pressure, $c_p$, and at constant volume, $c_v$, with the ratio being given by equation (2.1).

$$\gamma = \frac{c_p}{c_v} \quad (2.1)$$

The reader is referred to Roshko et al. [27], Massey [5] or Anderson [28] for derivations of these terms. Using specific heat capacity values for $c_p$ and $c_v$ that had been experimentally determined by DelaRoche and Berard [29], Laplace determined the speed of sound in air to be 345.9 m·s$^{-1}$, at 279K. This was not entirely in agreement with the best experimental value available at the time, 337.18 m·s$^{-1}$, Lindsay [7] (1945, p. xx). Lindsay indicates that the difference was due to inaccuracies in the Delaroche and Berard values for the specific heat capacity values however, it was never the less sufficiently accurate to warrant further work. Laplace returned to this theme with his treatise Mechanique Celeste [30] in which he used the more accurate value for the specific heat capacity values from the experiments of Clement and Desormes, Lindsay [7, 10] leading to an acoustic velocity of 332.9 m·s$^{-1}$.

Today, for air at international standard atmospheric (ISA conditions of 287K and 1013.25hPa), the specific heat capacity at constant volume, $c_v$, is measured at 734 J·kg$^{-1}$
1 K⁻¹ and at constant pressure, \( c_p \), it is measured as 1022 J·kg⁻¹·K⁻¹. The ratio of these numbers yields the specific heat capacity ratio, \( \gamma \), which for air is given in equation (2.2).

\[
\gamma_{air} = \frac{c_p}{c_v} = \frac{1022 \text{ J·kg}^{-1} \cdot \text{K}^{-1}}{734 \text{ J·kg}^{-1} \cdot \text{K}^{-1}} = 1.39
\]  

(2.2)

However, the values of the specific heat capacities, \( c_p \) and \( c_v \), are not linear with respect to temperature, Blair [31] (1999 p. 170). Blair [31] shows that the specific heat values for air are the sum of the individual specific heats of oxygen and nitrogen (ignoring trace constituent elements):

\[
C_{p,\text{air}} = C_{p,\text{O}_2} + C_{p,\text{N}_2} \quad \text{and} \quad C_{v,\text{air}} = C_{v,\text{O}_2} + C_{v,\text{N}_2}
\]

Blair’s [31] equations for the derivation of the specific heats for oxygen and nitrogen at constant pressure are given in equations (2.3) and (2.4).

\[
C_{p,\text{O}_2} = \frac{0.21 \cdot M_{\text{O}_2}}{M_{\text{air}}} \times \left( k_1^{\text{O}_2} + 2 \cdot k_2^{\text{O}_2} \cdot T + 3 \cdot k_3^{\text{O}_2} \cdot T^2 \right)
\]

\[
C_{p,\text{N}_2} = \frac{0.79 \cdot M_{\text{N}_2}}{M_{\text{air}}} \times \left( k_1^{\text{N}_2} + 2 \cdot k_2^{\text{N}_2} \cdot T + 3 \cdot k_3^{\text{N}_2} \cdot T^2 \right)
\]

(2.3)  

And at constant volume in equations (2.5) and (2.6).

\[
C_{v,\text{O}_2} = \frac{0.21 \cdot M_{\text{O}_2}}{M_{\text{air}}} \times \left( k_1^{\text{O}_2} + 2 \cdot k_2^{\text{O}_2} \cdot T + 3 \cdot k_3^{\text{O}_2} \cdot T^2 \right) - R_0
\]

\[
C_{v,\text{N}_2} = \frac{0.79 \cdot M_{\text{N}_2}}{M_{\text{air}}} \times \left( k_1^{\text{N}_2} + 2 \cdot k_2^{\text{N}_2} \cdot T + 3 \cdot k_3^{\text{N}_2} \cdot T^2 \right) - R_0
\]

(2.5)  

Where \( M \) is the molecular mass of the gas involved, \( R_0 \) is the universal gas constant and \( k_1, k_2, \) and \( k_3 \) are constants given in Table 1 for the respective gases, which Blair [31] (1999, p. 172) states are:

“reasonably accurate for a temperature range of 300K to 3000K”
Table 1 Properties of oxygen and Nitrogen

<table>
<thead>
<tr>
<th></th>
<th>$O_2$</th>
<th>$N_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_1$</td>
<td>2.9672E+04</td>
<td>2.7280E+04</td>
</tr>
<tr>
<td>$k_2$</td>
<td>2.6865E+00</td>
<td>3.1543E+00</td>
</tr>
<tr>
<td>$k_3$</td>
<td>-2.1194E-04</td>
<td>-3.3052E-04</td>
</tr>
</tbody>
</table>

Table 2 shows how the specific heat capacity ratio varies with temperature.

Table 2 Specific heat capacity ratio versus temperature

<table>
<thead>
<tr>
<th>$T_0$ (K)</th>
<th>$\gamma$</th>
</tr>
</thead>
<tbody>
<tr>
<td>273</td>
<td>1.4</td>
</tr>
<tr>
<td>293</td>
<td>1.39</td>
</tr>
<tr>
<td>373</td>
<td>1.39</td>
</tr>
<tr>
<td>500</td>
<td>1.37</td>
</tr>
<tr>
<td>1000</td>
<td>1.34</td>
</tr>
<tr>
<td>3000</td>
<td>1.3</td>
</tr>
</tbody>
</table>

Despite this temperature variation in the values of the specific heat capacities and their resulting ratio, Bannister [32] and Blair [31] both use a value of 1.4 for $\gamma$, which is the value that will be used for the remainder of this discussion.
2.4 The speed of Sound

In various texts on sound and acoustics such as Earnshaw [33], Bannister [32], Kinsler et al. [34], Massey[5], and Roshko et al. [27], it is proven that the velocity of a sound wave propagating into an undisturbed gas is given by equation (2.7).

\[ a_0 = \sqrt{\gamma R T_0} \]  \hspace{1cm} (2.7)

Where \(a_0\) is the local acoustic velocity and \(T_0\) is the local temperature in the undisturbed gas. \(R\) is the specific gas constant which is dependant on the composition of the gas and is found from the relationship using the universal gas constant \(R_0\), and the molecular weight of the gas \(M_{gas}\):

\[ R = \frac{R_0}{M_{gas}} \]

Where \(R_0\) is the product of Avogadro’s and Boltzman’s constants and has a value of 8.314 J·mol\(^{-1}\)·K\(^{-1}\). In the case of air, the molecular weight is the sum of the masses of the constituent parts and is given as:

\[ M_{air} = \sum(\%_{gas} M_{gas}) = 28.97 \text{ g} \]

Therefore the specific gas constant of air is determined to be 287 J·Kg\(^{-1}\)·K\(^{-1}\). Therefore the local acoustic velocity or the speed of sound in a given gas is simply proportional to the square root of the temperature of that gas, as shown in equation (2.8).

\[ a_0 \propto \sqrt{T_0} \]  \hspace{1cm} (2.8)

Using equation (2.7), where \(\gamma = 1.4\) and \(R = 287 \text{ J·Kg}^{-1}\cdot\text{K}^{-1}\),

Table 3 shows how the local acoustic velocity, \(a_0\), in air varies with temperature.
Table 3 Effect of temperature on speed of sound in air (constant $\gamma$)

<table>
<thead>
<tr>
<th>$T_0$ (K)</th>
<th>$a_0$ (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>273</td>
<td>331.2</td>
</tr>
<tr>
<td>293</td>
<td>343.1</td>
</tr>
<tr>
<td>323</td>
<td>360.3</td>
</tr>
<tr>
<td>373</td>
<td>387.1</td>
</tr>
<tr>
<td>500</td>
<td>443.4</td>
</tr>
</tbody>
</table>

2.5 Summary

In this chapter a brief history of the research into the determination of acoustic velocity has been given. It has been shown that an acoustic wave propagating in air does so adiabatically, i.e. no heat exchange occurs, and that acoustic velocity is proportional to the square root of the local temperature of the gas. With an acoustic velocity versus temperature relationship established, the following chapter presents various temperature measurement devices and the typical scales involved.
CHAPTER 3

3.1 Temperature Measurement Scales, Formats and Devices

The first temperature measurement device reported is attributed to Galileo Galelei, Michaelski [35]. This device consisted of a glass bulb with a long tube that contained a heated gas. The end of the tube was immersed into a liquid and as the gas cooled it sucked the liquid up. The liquid rose or fell as a function of ambient temperature. While interesting, this thermometer was not of much practical use other than it being a curiosity. In fact Michaelski [35] refers to this device as a thermoscope rather than a thermometer due to it’s apparent lack of graduations or reference to pressure. Since Galileo’s ‘thermoscope’, numerous attempts have be made at defining a scale such that the ‘thermometer’ could be made into a useful device. One such device that answered a specific need during the early days of the pottery trade in England, was that designed by Josiah Wedgewood in 1780 to assist in determining the temperature of kiln furnaces, Wedgewood [36]. This device consisted of two tapering channels on a flat plane (i.e. piece of wood or stone), see Figure 1, into which specially sized pieces of argillaceous, or clay based material that had been heated in the furnace would slot into. Since heating this clay material caused it to shrink, testing samples at different colours gave an indication of the temperature of the furnace.
Figure 1 Wedgewood Pyrometer

While Wedgewood’s solution and design was ingenious, it was specific to a particular application, and of little use in other areas. Enfield [37] (1820, book III, part II, p. 112) compares the Wedgewood scale to that of the Fahrenheit scale where Wedgewood’s design was in essence a variation of the Fahrenheit scale with each of the Wedgewood intervals or degrees equivalent to 130 Fahrenheit intervals or degrees, see Table 4.

Table 4 Wedgewood versus Fahrenheit scaling

<table>
<thead>
<tr>
<th>Visual Indication</th>
<th>Wedgewood Scale (°)</th>
<th>Fahrenheit Scale (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Extreme of the Wedgewood scale</td>
<td>240</td>
<td>32277</td>
</tr>
<tr>
<td>Cast Iron Melts</td>
<td>160</td>
<td>21877</td>
</tr>
<tr>
<td>Lowest Iron Welding Heat</td>
<td>90</td>
<td>12777</td>
</tr>
<tr>
<td>Fine gold melts</td>
<td>32</td>
<td>5237</td>
</tr>
<tr>
<td>Fine Silver melts</td>
<td>28</td>
<td>4717</td>
</tr>
<tr>
<td>Brass Melts</td>
<td>21</td>
<td>3807</td>
</tr>
<tr>
<td>Red heat visible in day light</td>
<td>0</td>
<td>1077</td>
</tr>
</tbody>
</table>

Interesting as Wedgewood’s solution and scaling was it is not in mainstream use today unlike the Fahrenheit scale which is one of three scales that have withstood the test of time and are in common use today. These scales are:
• **Fahrenheit** - named after Daniel Gabriel Fahrenheit (1686 – 1736). In the early 1700’s he was responsible for the mercury in glass style thermometer, Chang [38], Cajori [39].

Fahrenheit identified three reference calibration points for his thermometer:

- $0^\circ\text{F}$ - a mixture of ice, water and ammonium chloride
- $32^\circ\text{F}$ - a mixture of ice and water
- $96^\circ\text{F}$ - average human body temperature

Fahrenheit placed the freezing point and boiling points of water 180° apart since on his scale water froze at $32^\circ\text{F}$ and water boiled at $212^\circ\text{F}$. The Fahrenheit scale is still widely used in the United States of America.

• **Celsius** - named after Anders Celcius (1701 – 1744) and edited by Mårten Strömer (1707–1770) *et al.* who made the freezing point of water equal to $0^\circ\text{C}$ and the boiling point equal to $100^\circ\text{C}$, Cajori [39]. The Celsius scale is widely used today throughout Europe.

• **Kelvin** – William Thompson (1824 – 1907), (Lord Kelvin) laid the foundations for an absolute temperature scale, Cajori [39], with a point defined at which the entropy, or the degree of disorder of the system approaches zero. This zero entropy point is given the temperature value of 0K which is termed absolute zero, Chang [38] and which corresponds to -273.15°C. Table 5 shows comparison between the three typical scales in use today.
Table 5 Temperature scale equivalents

<table>
<thead>
<tr>
<th></th>
<th>Kelvin</th>
<th>Celcius (°C)</th>
<th>Fahrenheit (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absolute zero</td>
<td>0.0</td>
<td>-273.15</td>
<td>-459.67</td>
</tr>
<tr>
<td>H₂O Triple point</td>
<td>273.16</td>
<td>0.01</td>
<td>32.02</td>
</tr>
<tr>
<td>Nominal</td>
<td>298.15</td>
<td>25.00</td>
<td>77.00</td>
</tr>
<tr>
<td>H₂O boiling point</td>
<td>373.23</td>
<td>99.98</td>
<td>211.97</td>
</tr>
</tbody>
</table>

For more information on temperature scales and the current definitions, the reader is referred to the *International Practical Temperature Scale of 1968 (IPTS-68)* [40] and the updated version *International Temperature Scale of 1990 (ITS-90)* [1].
3.2 Temperature Measurement Methods

Since the time of Galileo there has been an almost endless variety of temperature measurement devices for just about every application conceivable. The range of temperature measurement devices is too extensive to list, however, essentially they may be divided into two main categories:

- Contact
- Non-contact.

Michaelski [35] breaks these categories up again into three further categories:

- Electrical
- Non-Electrical
- Optical

Figure 2 gives a brief overview of some different temperature sensors and where they might fit in Michaelski’s classification.

![Figure 2 Basic temperature sensor formats [35]]
Figure 3 shows the approximate ranges of several of these devices... the list of devices given in Figure 3 is by no means extensive and does not include devices such as crayons, paints, semiconductors (P-N junction), patches/labels, lacquers, crystals, pellets or pyrometric cones such as Harrison or Seger cones. Harrison and Seger cones are elongated three-sided pyramids of ceramic based material that are designed to ‘bend’ or ‘collapse’ over at a given temperature, Hamer [41]. The reader is referred to Michaelski [35] for more information on these various temperature measurement solutions.

![Diagram showing typical ranges of common temperature measurement devices](image)

**Figure 3** Typical ranges of common Temperature measurement devices

The ranges for the devices listed in Figure 3 are somewhat idealised, i.e. mercury-in-glass devices are typically under vacuum, and to measure temperatures above...
approximately 200°C they must be filled with a pressurised inert gas such as Nitrogen, Michaelski [35].

There is one very significant drawback to all the measurement methods discussed in this chapter: that is that they invariably have slow temporal responses: i.e. a mercury-in-glass thermometer can take several tens of seconds, even minutes to reach temperature and can have a full recovery time of several hours due to glass hysteresis. Although some EGT thermocouples have response times of the order of hundreds of milliseconds [42], thermocouples generally can have a temporal response time of the order of seconds, and require a temperature reference, sometimes referred to as ‘cold junction’ compensation. Similar slow response times are usually observed in bimetallic strip sensors. Another issue with several measurement solutions is that of ‘secondary’ effects whereby the measurement device must extract heat from the medium being measured in order to measure the temperature, e.g. bi-metallic strip or mercury-in-glass. When it comes to the non-contact devices, e.g. optical pyrometers, unobstructed line-of-sight access to the target being measured is necessary, which can be problematic in the case of a gas. A gas at temperature, transiting in a conduit, will heat up the conduit at a rate dependant on the velocity of and the time that the gas is transiting in the conduit. The thermal efficiency of the material the conduit is constructed of, i.e. steel or plastic, will play a part in determining the heating effect. Therefore any temperature measurement made remotely against the exterior of the conduit will not necessarily measure the instantaneous temperature of the gas but rather the heating effect that the gas has as it transits the conduit. This is further complicated if the conduit is thermally lagged or insulated.
For other types of sensors, the response to temperature is irreversible, i.e. some colour changers or indicator pyrometric cones such as Seger or Harrison cones. The reader is referred to the work of Michaelski [35] for more detailed information on these and other temperature sensors.
3.3 Acoustics in Temperature Measurement

The use of acoustics in temperature measurement is not new and is commonly referred to as Acoustic Pyrometry. Kleppe [43] (1989, p.289) attributes the first use of this term to Mayer [44]. While acoustics is the name commonly used for the science of sound, the pyrometer is defined by the OED [3] as:

“an instrument for measuring high temperatures... in furnaces and kilns”

An example of such a device is the Optical Pyrometer. The optical pyrometer is used by the operators of large furnaces and by the likes of fire-departments to help identify hot-spots in a fire situation where close proximity to the heat source under measurement could be hazardous to the safety of the operator, the survivability of the sensor or both. As such, the optical pyrometer is a non-contact measurement device that intercepts and measures thermal radiation. A typical optical pyrometer is shown in Figure 4.

![Typical hand held optical (IR) pyrometer](image)

Acoustic pyrometry on the other hand typically involves the introduction of an acoustic signal through physical sensors in contact with the gas as shown in Figure 5. Livengood et al [45] and Kleppe [43] (1989, p. 291) describe the theory of the operation of such an apparatus. Figure 5 shows a typical apparatus whereby an acoustic signal is injected into
the gas that is detected by a suitable acoustic receiver a known distance, $d$, away. The time taken for this pulse to transit from the transmitter to the receiver is known as the ‘time of flight’ (TOF), of the signal. Knowing the distance travelled and the TOF involved, the acoustic velocity $a$, can be determined. By knowing certain properties of the gas in question, i.e. the specific heat capacity ratio, $\gamma$, and the specific gas constant, $R$, the local average temperature can be found by rearranging equation (2.7) as equation (3.1).

$$T_0 = \frac{a^2}{\gamma \cdot R} \quad (3.1)$$

Figure 5 Typical active acoustic pyrometric setup

The transmitted signal can be a time varying signal or more typically a digital pulse train of either a chirp or barker pattern. A chirp pattern is a consecutive series of 1’s and 0’s similar to a microprocessor clock train. Figure 6 shows a typical 8-bit chip pattern.
A barker pattern on the other hand is a sequence of 1’s and 0’s similar to the chirp pattern but with a phase change somewhere in the sequence. Figure 7 shows a typical 8-bit barker pattern. The phase change does not have to be centred in the middle of the bit train and is shown here for convenience only.

A digital transmission signal, such as is shown in Figure 6 or Figure 7 has benefits over a time varying analogue, i.e. a non-digital, transmitted signal in that signal recovery and analysis is facilitated through the use of digital signal processing techniques making the determination of TOF easier. Whether the system shown in Figure 5 is considered contact or non-contact is largely a matter of semantics. A more important distinction is whether the system is active or passive. For example, a mercury thermometer is inherently a passive device that reacts to the temperature of the medium in which it is immersed. Acoustic pyrometry, as shown in Figure 5, is an active system in which one part of the transducer reacts to a deliberate transmission introduced into the gas by another part of the transducer.

This system shown in Figure 5 is described by Kleppe [43, 46, 47] and Stones [48] and may be found in exhaust flues, chimneys and furnaces. Livengood [45] describes an end-gas temperature monitoring application in an internal combustion engine. The
resultant temperature measured by the apparatus shown in Figure 5 is an average temperature as opposed to a localised instantaneous result. The frequency of the transmitted acoustic pulse would ideally be in the ultrasonic range for two primary reasons:

- higher frequencies mean better temporal resolution in the determination of TOF
- higher frequencies mean the resultant tone is outside normal hearing range.

Kleppe [43] (1989, p.292) however, indicates that experimentation has shown that frequencies of between 500Hz and 2kHz to be optimal for large measurement systems such as chimney flues. This unfortunately means that the acoustic frequencies are well within the normal human and animal hearing range. One reason for such low frequencies is acoustic attenuation, see Kleppe [43] (1989, pp. 309..310). Acoustic attenuation is dependant on frequency, humidity, temperature, pressure as well as distance travelled. The reader is referred to ISO 9613-1 (1993), which gives details for determining the expected attenuation under given atmospheric conditions.

Table 6 shows data presented by Hsu et al. [49], showing the relationship between acoustic frequency and attenuation in air. The data is extrapolated down to 50kHz and up to 2MHz using the polynomial shown in equation (3.2) where \( f \) is the acoustic frequency and \( y \) is the resultant attenuation.

\[
y = -2 \times 10^{-12} \cdot f^4 - 5 \times 10^{-8} \cdot f^3 + 2 \times 10^{-4} \cdot f^2 - 2.07 \times 10^{-2} \cdot f + 4.2439 \tag{3.2}
\]

Where \( f \) is the frequency in Hertz and \( y \) is the resultant attenuation (dB\(\cdot\)m\(^{-1}\)).
Table 6 Ultrasonic attenuation in air

<table>
<thead>
<tr>
<th>Freq (kHz)</th>
<th>Attenuation (dB·m⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>3.7</td>
</tr>
<tr>
<td>120</td>
<td>5.0</td>
</tr>
<tr>
<td>500</td>
<td>45.0</td>
</tr>
<tr>
<td>1000</td>
<td>160.0</td>
</tr>
<tr>
<td>2000</td>
<td>331.0</td>
</tr>
</tbody>
</table>

Table 6 shows that even at moderately ultrasonic frequencies, e.g. 50kHz, the amplitude of the transmitted signal is attenuated by more than half for every metre of travel in air. Livengood et al. [45] and Gluckstein [50] describe an acoustic pyrometric system for the measurement of, what they describe as ‘end-gas’ temperature in an internal combustion engine for the investigation of engine knock. Livengood et al. [45] used an acoustic transmit frequency of 2MHz which was derived from the high-voltage excitation of a piezo-electric crystal. This frequency undoubtedly offers significant improvements in the temporal resolution of time of flight measurements over Kleppe’s rather low optimal frequency. In Kleppes defence however, the acoustic path distances involved are quite different, with acoustic attenuation being determined by distance travelled and acoustic signal transmission frequency as shown in Table 6. With regards active acoustic pyrometry Kleppe [43] (1989, p. 311) states that:

“One of the most fundamental design problems is selecting the source so that it will have enough energy to operate in the noisy environment... and overcome the attenuation resulting in an adequate signal-to-noise ratio for subsequent processing”

Active acoustic pyrometric systems can therefore require high levels of power output from the transmitter. The output power required is dependant on the length of the acoustic path, the acoustic properties of the gas and the acoustic frequency being transmitted. Green et al. [51] report that a spark gap transmitter in a furnace application
produced transmitter sound pressure levels (SPL) of up to 170dB. While spark gap sound sources are indeed capable of producing sufficiently high energy levels for use in most pyrometric applications, Kleppe [43] (1989, p312) questions their use on grounds of safety and reliability.

Active acoustic pyrometry is perfectly suited to situations where the gas flow is either a static flow, i.e. no flow, or a continuous flow such as might be the case in a boiler or furnace exhaust. Where the combustion process is non-continuous, as might be found in an internal combustion engine, and where ambient noise levels can be extreme, an active acoustic approach becomes problematic, Kleppe [43] (1989, p.312). According to Livengood et al. [45] and Gluckstein [50] significant modifications are required to the engine to incorporate, in their case ‘end gas’ temperature monitoring through active acoustic pyrometry. In addition the timing required for correct acoustic signal transmission is critical in that the signal must be transmitted at the correct point in the combustion cycle.

Therefore, where the gas flow consists of a series of finite duration pressure pulses, that are constrained in an exhaust or blow-down pipe, the hypothesis was formed that rather than induce a signal into the exhaust stream as per Livengood et al. [45] and Gluckstein [50], the pressure pulses themselves could be used as part of the sensor network. The pressure pulse or wave resulting from the actuation of an exhaust valve on an IC engine is a finite duration impulse generated by the disparate pressure regions in existence on either side of the valve. The resulting pressure wave is propagated or transmitted from the high pressure to the low pressure region at a velocity proportional to the pressure differential existing between the pressure regions at the moment of interface rupture.
The hypothesis was that the pressure pulses would interact with a transducer or transducers of some format during their transit. The results of this interaction would be the derivation of the local acoustic velocity $a_0$, and hence the local temperature $T_0$. Such a system is passive in operation, i.e. no external signal is induced into the gas. Figure 8 shows a block diagram of the apparatus that would be involved in such a passive acoustic pyrometric system.

![Figure 8 Passive Acoustic Pyrometry](image)

The pressure pulse or wave propagating in the exhaust or blow-down pipe interacts with detectors, which are positioned a known distance apart, to produce two temporally different responses. Figure 9 shows experimental acoustic responses following the passage of a pressure pulse or wave in the blow-down pipe from tone generators at positions $t_1$ and $t_2$ in an apparatus similar to that described in Figure 8. The temporal difference between each of the acoustic responses is clearly evident.
The temporal difference in the response times from each of the transducers determines the time-of-flight (TOF) of the pressure pulse. This TOF, in conjunction with the distance the pressure pulse has to travel is used to determine the wave velocity $W$. The velocity of the shock wave $W$, is not to be confused with acoustic velocity, $a_0$. They are two separate, yet interconnected properties of a pressure wave propagating in a pipe. The relationship between the acoustic velocity and pressure or shock wave velocity is discussed in Chapter 4.
CHAPTER 4

4.1 The Pressure wave

Bannister [32] (1958, p. 7) shows that for air, where the specific heat capacity ratio $\gamma$, is taken to be 1.4, the ratio of the instantaneous acoustic velocity, $a$, in the wave to the idealised undisturbed acoustic velocity, $a_0$, is given by equation (4.1).

$$\frac{a}{a_0} = \sqrt{\frac{T}{T_0}} = \left(\frac{P}{P_0}\right)^{\frac{\gamma-1}{2\gamma}}$$  \hspace{1cm} (4.1)

Where $a$, $T$ and $P$ are the local instantaneous acoustic wave velocity, temperature and pressure values respectively, at a specific point in the travelling wave. $a_0$, $T_0$ and $P_0$ are the acoustic velocity, temperature and pressure values of the undisturbed air, i.e. that area into which the acoustic wave is propagating. This means that for a given parcel of air ($\gamma = 1.4$) with undisturbed or local conditions as per Blackstock [52] (2000, p512), of $T_0 = 293\text{K}$, $P_0 = 101325\text{Pa}$ and $a_0 = 343\text{m}\cdot\text{s}^{-1}$, the instantaneous acoustic velocity in a pressure wave propagating into the undisturbed gas region resulting from a 2x pressure differential will be $378.7\text{ m}\cdot\text{s}^{-1}$. Equation (4.1) shows that the instantaneous acoustic velocity, $a$, at a given instant of time is dependant on the pressure gradient that exists at that time.

While equation (4.1) is interesting it is not the entire story with regards pressure waves in pipes. A simple pressure wave, Bannister [32] (1958, p.1) and Roshko et al. [27] (2001, p. 69), is a disturbance propagated in one direction in a fluid. Texts on the subject of pressure waves in pipes, such as those by Blair [31], Bannister [32], Anderson [28], and Kinsler [34] state that a pressure or shock-wave transiting down a pipe will consist of two distinct regions: a pressure wave and a following particle-mass.
Both these regions are illustrated in Figure 10 (B). In the diagram shown in Figure 10 (A), two distinct pressure regions exist: a high pressure or driver region and a low pressure or driven region. These regions are separated by an interface which could be an exhaust valve as found in an IC engine or a membrane or diaphragm as used in a shock tube. The shock tube is a tube in which a high pressure and a low pressure gas are separated by a diaphragm designed to rupture at a predetermined pressure differential, or by means of an external action, resulting in a pressure or shock wave propagating into the low pressure region, see Anderson [28] (2002, p. 262) or Roshko [27] (2001, p. 80).

![Figure 10 Finite Pressure wave regions](image)

The term ‘low-pressure’ does not necessarily mean vacuum or less than atmospheric, in fact it should simply be taken as meaning ‘lower’ pressure than the driver section, and can usually assumed to be atmospheric.

If at some instant in time the interface between the driver and driven pressure regions is ruptured, as in Figure 10 (B), the pressure gradient existing at the moment of rupture
will set up a pressure wave that travels from the high pressure region, \( P \), to the low pressure region, \( P_0 \). This pressure wave is known as the compression wave. Bannister [32], Blair [31], and Anderson [28] amongst others all give derivations for the velocities of the two distinct regions in the pressure wave due to the pressure gradient. However, the equation of interest for this discussion is that for the shock or pressure wave, \( W \). Bannister indicates that this pressure wave will propagate at a velocity determined by the conditions of the undisturbed gas immediately ahead of it, and from the work presented by Earnshaw [33], that the propagation velocity of any point on a pressure wave, \( W \), is equal to the sum of the instantaneous acoustic velocity, \( a \), and the particle-mass velocity, \( u_p \), in the wave, equation (4.2).

\[
W = a + u_p \quad (4.2)
\]

By rearranging equation (4.1) and substituting for the instantaneous acoustic velocity, \( a \), in equation (4.2) yields equation (4.3).

\[
W = a_0 \left( \frac{P}{P_0} \right)^{\frac{\gamma-1}{2\gamma}} + u_p \quad (4.3)
\]

Bannister [32] (1958, p.7), referring to Earnshaw [31], defines the particle velocity, \( u_p \), as given in equation (4.4).

\[
u_p = \frac{2}{\gamma-1} a_0 \left[ \left( \frac{P}{P_0} \right)^{\frac{\gamma-1}{2\gamma}} - 1 \right] \quad (4.4)
\]

It should be noted that throughout the remainder of this document a single convention for the different terms will be used. These terms adhere closely to those used by Anderson [28] (2003, p269) in that

- \( W = \) velocity of the shock wave,
- \( P = \) driver or high pressure
- \( P_0 = \) ambient or driven pressure
- \( a_0 = \) the local undisturbed acoustic wave velocity.

Combining equations (4.3) and (4.4) produces equation (4.5), which is the velocity of the pressure wave, in terms of local undisturbed acoustic velocity \( a_0 \), and the pressure ratio involved.

\[
W = a_0 \cdot \left( 6 \cdot \left( \frac{P}{P_0} \right)^{\left( \frac{\gamma - 1}{2\gamma} \right)} - \frac{2}{\gamma - 1} \right)
\]  

(4.5)

The local acoustic velocity, \( a_0 \), can now be found by rearranging equation (4.5) to give equation (4.6).

\[
a_0 = \frac{W}{6 \cdot \left( \frac{P}{P_0} \right)^{\left( \frac{\gamma - 1}{2\gamma} \right)} - \frac{2}{\gamma - 1}}
\]  

(4.6)

The result for the acoustic velocity from equation (4.6) can then be substituted into equation (2.7) to yield the local undisturbed temperature \( T_0 \).
4.2 Acoustic transducers

Figure 8 showed a concept using passive acoustic detectors for determining the local acoustic velocity, $a_0$, using equation (4.6), and hence the local temperature, $T_0$, by using equation (2.7) from the propagation of a pressure wave. From this, the hypothesis was formed that the discontinuous, finite duration pressure waves, generated by the action of an IC combustion-exhaust cycle could be used as an integral part of the measurement solution. To detect the passage of these finite duration pressure waves using passive acoustic means, a tone generator that can respond to the passage of the pressure wave and produce an acoustic response was considered. Two tone generator formats that satisfy this requirement are the Helmholtz resonator and the edge-tone generator. A third acoustic resonator, the reed pipe was discounted because of its inherent moving parts, which was considered a reliability risk. The Helmholtz resonator, which is a tone generator where an enclosed cavity, coupled to the outside air by an aperture causes an oscillation when a burst of air is blown across the aperture, Berg [53]. This oscillation can be detected using a conventional acoustic microphone. The edge-tone mechanism such as in a labial flue pipe is a device where a moving mass of air is directed in a fine jet onto a blade or edge which is coupled to a cavity and produces an acoustic tone that can, once again be detected using conventional microphone technology.

The optimal tone generator was considered to require certain criteria that would dictate which was the best type to use. The criteria decided upon were:

1. **Tone Generator**: Must be capable of generating a tone that can be easily detected with the passage of a finite pressure wave under the conditions to be expected and ideally should be of a passive nature.

2. **Form factor**: The device must be of a size to allow it to be usable.
3. **Efficient with good dynamic response.** It should have a high dynamic response and be sensitive enough to react to small pressure gradients. It should also neither over-blow nor saturate easily.

4. **Large output amplitude.** The output amplitude should be of sufficient amplitude to facilitate detection.

5. **High frequency.** As already indicated for the case with the active acoustic solution: temporal resolution is proportional to frequency with high frequency improving the resolution of periodic or TOF measurements. High frequency capability is also desirable to allow the resultant tone be outside the hearing range of humans (20Hz - 20kHz) and animals (2Hz up to approximately 250kHz), Physics Hypertextbook [54]. However, for this proof of concept investigation, the ideal frequency response of the tone generators was considered to be in the audio range to facilitate setup and optimisation.

With these criteria in mind, the Helmholtz Resonator is discussed in Chapter 5.
CHAPTER 5

5.1 Helmholtz Resonator


The operation of the Helmholtz resonator is described in numerous texts such as Blackstock [52], Kinsler et al. [34], Fletcher [56] and Raichel [57]. It was developed by Herman von Helmholtz (1821 – 1894) as a means of singling out particular frequencies from complex tones, Helmholtz [58]. The Helmholtz resonator consists of a rigidly enclosed cavity of volume $V$, which is coupled to the outside atmosphere through an opening or aperture of cross-sectional area $S$, and length $L$, see Figure 11.

![Figure 11 Typical Helmholtz Resonator](image-url)
The basic operation is that as a mass of air is blown across the neck opening, the air in the neck, which acts as a plug with a mass of \( \rho \cdot L \cdot S \), where \( \rho \) is the density of the gas \( (\rho_{\text{air}} = 1.2 \, \text{kg} \cdot \text{m}^{-3}) \), presses into the cavity compressing the air within. The pressure in the cavity is therefore increased slightly, even though the volume remains constant. The volume remains constant because the compression operation occurs sufficiently rapidly enough that the system remains essentially adiabatic, meaning there is no heat exchange. The pressure in the cavity, now at a higher pressure than it was at equilibrium acts as a spring, which ‘relaxes’ forcing the air plug back out the neck in opposition to its initial deflection. This ‘relaxation’ causes the pressure in the cavity to drop causing air to be pulled back into the neck. Providing the air being blown across the neck that started the operation remains, this “pushing-pulling” effect of the air in the neck causes a standing wave to be induced resulting in a tone being produced.

Fletcher [56] describes the fundamental equation, given in equation (5.1), that defines the resultant frequency generated from a Helmholtz resonator in oscillation.

\[
f = \frac{a_0}{2\pi} \sqrt{\frac{S}{L \cdot V}}
\]  

(5.1)

### 5.1.1 Effective Length

However, equation (5.1) does not take into account the acoustical end-correction of the neck length. The physical length of the neck, \( L \), and the perceived length, \( L' \), as seen by a pressure wave, are two different things. Therefore, a more correct version of the Helmholtz resonator frequency equation is given by equation (5.2).

\[
f = \frac{a_0}{2\pi} \sqrt{\frac{S}{L' \cdot V}}
\]  

(5.2)
The difference between \( L \) and \( L' \) is called the end-correction, where \( L \) is the physical length of the neck and \( L' \) is the acoustical length or Effective Length of the neck.

The effective length of an open pipe is understood by realising that a pressure wave exiting the pipe does not suddenly drop to zero gauge pressure at the exact end of the pipe, but instead takes some distance to dissipate. This extra distance, known as end-correction, has the effect of making the pipe appear longer, from an acoustical point of view, than its actual physical length. This total distance, actual length plus end-correction, is the acoustical Effective Length or Acoustical Length of the pipe, see Figure 12. Rayleigh [59] (1945, p. 203) credits Blaikley (1846 – 1936), with the experimental determination of the end-correction on an un-flanged pipe which he gave as \( 0.576 \cdot r \), where \( r \) is the radius of the pipe opening. Blackstock [52] and Kinsler [34] provide good descriptions and derivations for the effective length of a pipe.

\[
\hat{L}_{\text{UNFLANGED}} = L + 0.6 \cdot r \tag{5.3}
\]

The result of equation (5.3) closely matches Blaikleys experimental result. Kinsler also shows that the effective length of a flanged pipe is given by equation (5.4).
\[ L'_{\text{FLANGED}} = L + 0.85 \cdot r \]  

(5.4)

However, unlike the pipe shown in Figure 12, the Helmholtz resonator has two end-corrections to take account of, one open to atmosphere and another open to the cavity. Blackstock [52] (2000, p. 155) indicates this and refers the reader to Panton [60] who states that while it is usual to use a mixture of equations (5.3) and (5.4) in determining the effective length of a pipe, this can produce significant errors. Panton indicates that Ingard’s [61] proposal produces a more accurate result, whereby rather than treat the cavity end as a piston radiating into space, as Rayleigh does, it is better, and more accurate to model it as a piston in a tube of radius \( r \), i.e. the radius of the neck in Figure 11 radiating into another tube of radius \( R_c \), i.e. the radius of the cavity in Figure 11, with the constraint that:

\[ r < 0.4 \cdot R_c \]  

(5.5)

This means that equation (5.4) is rewritten as equation (5.6).

\[ L' = L + 0.85 \cdot r \cdot \left( 1 - 1.24 \frac{r}{R_c} \right) \]  

(5.6)

Panton [60] (1975, p. 1535) states that by using Ingard’s solution the resultant frequency error can be reduced to less than 2%.
5.2 Helmholtz Investigation

Regardless of the specifics of which end correction method to use, a more fundamental question was whether the Helmholtz resonator would be suitable for the operation required of it. Therefore, experimentation and investigation of the Helmholtz resonator was warranted resulting in the apparatus shown in Figure 13.

The container/resonator shown in Figure 13 had the following dimensions:

- Cavity volume ($V$) $\approx 125 \times 10^{-6} \text{m}^3$
- Neck Length ($L$) $\approx 9\text{mm}$
- Neck cross-sectional area ($S$) $\approx 58.1 \times 10^{-6} \text{m}^2$ (neck radius = 4.3mm).

V, L and S are shown in Figure 11.

Using these dimensions, and a combination of equations (5.2) to (5.6), the theoretical oscillation frequency to be expected was determined to be 339.45Hz. This is seen in the LabVIEW virtual instrument front panel in Figure 14 which uses Ingard’s equation (5.6) for end-correction.
Figure 14 Helmholtz theoretical

The resonant frequency was then determined experimentally, using the apparatus shown in Figure 13, in which a jet of air was blown across the open neck of the container. The resultant oscillation, or tone was captured using a microphone (Yoga EM-278) and a Tektronix THS-720 oscilloscope with appropriate LabVIEW based acquisition and analysis software. Analysis showed the resonant frequency of oscillation to be 337.84Hz. Figure 15 shows a screen capture plot of the LabVIEW analysis software with the measured frequency highlighted.

Figure 15 Helmholtz resonator experimental result
This experimental result is a very close match to the theoretically determined result shown in Figure 14. The discrepancy between the two results is 1.61Hz. At the theoretical frequency of interest, 339.45Hz, the difference is just over 12% of 1 semitone. This result indicates that the theoretical equations used are a valid means of analysing the Helmholtz resonator.

However, there were two key concerns regarding the use of a Helmholtz resonator:

- Frequency of operation versus physical dimensions
- Start up dynamics.

The first concern was the frequency of operation and the dimensions of the resonator required to produce usable frequencies. The experimental apparatus shown in Figure 13 produced the rather low frequency of approximately 338Hz. In order to obtain higher frequencies, as required by point 5 of the key points identified in section 4.2 Acoustic transducers, the physical dimensions of the resonator needed to be adjusted. Using equation (5.2), each of the dimensions of the resonator, \( S \), \( L \), and \( V \) were varied individually to theoretically determine the effects.

Table 7, Table 8 and Table 9 indicate the results of this analysis. In all cases, the non-varying parameters were arbitrarily set to default values as follows:

- Volume (V) = \( 10 \times 10^{-6} \) m\(^3\)
- Neck Length (L) = 8.5mm
- Neck opening cross-sectional area (S) = \( 60 \times 10^{-6} \) m\(^2\)

Table 7 shows theoretically how the Helmholtz oscillation frequency varies with respect to the neck opening area (S). The neck length, \( L \), and volume of the cavity, \( V \), were held constant at \( 8.5 \times 10^{-3} \) m and \( 10 \times 10^{-6} \) m\(^3\) respectively.
Table 7 Helmholtz resonator frequency versus neck opening

<table>
<thead>
<tr>
<th>Neck Radius (m)</th>
<th>S (x10^{-6}m^2)</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0001</td>
<td>0.0314</td>
<td>33</td>
</tr>
<tr>
<td>0.0010</td>
<td>3.14</td>
<td>318</td>
</tr>
<tr>
<td>0.0100</td>
<td>314.2</td>
<td>3206</td>
</tr>
<tr>
<td>0.0150</td>
<td>706.9</td>
<td>7754</td>
</tr>
<tr>
<td>0.0170</td>
<td>907.2</td>
<td>41297</td>
</tr>
</tbody>
</table>

Table 7 indicates that high frequencies are certainly theoretically possible but the dimensions of the neck required are quite large, up to 33.99 mm in diameter to achieve an oscillation frequency of just over 41kHz. This means that at these conditions the cross-sectional area of the neck, S would be larger than the cross-sectional area of the cavity at it widest point, which for the dimensions given, has a diameter of 26.73mm yielding a cross-sectional area of 561.2x10^{-6} m^2.

Table 8 shows the theoretical effect on the oscillation frequency due to varying neck length \( L \). The neck opening area, \( S \), and the cavity volume, \( V \), are held constant at 60x10^{-6} m^2 and 10x10^{-6} m^3 respectively.

Table 8 Helmholtz resonator frequency versus neck length

<table>
<thead>
<tr>
<th>L (mm)</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000.0</td>
<td>134</td>
</tr>
<tr>
<td>100.0</td>
<td>418</td>
</tr>
<tr>
<td>10.0</td>
<td>1211</td>
</tr>
<tr>
<td>1.0</td>
<td>2361</td>
</tr>
<tr>
<td>0.1</td>
<td>2784</td>
</tr>
<tr>
<td>0.01</td>
<td>2840</td>
</tr>
<tr>
<td>0.001</td>
<td>2846</td>
</tr>
</tbody>
</table>

Table 8 indicates that neck length has very little impact on frequency. Changing \( L \) by a factor of 1,000,000x produced only a factor of 21x change in resonant frequency. Also,
a neck length of 1m is not commensurate with a ‘small’ form factor as required by point 2 in section 4.2 Acoustic transducers. Equally, a neck length of 0.001x10\(^{-3}\) m, while small, is too small to be of practical use, and is meaningless given the default parameters in use. The resultant oscillation frequency was considered also too low at 2.85kHz, considering that high frequency capability was one of the criteria of the tone generator.

Table 9 shows theoretically how resonant oscillation frequency varies with varying cavity volume, \(V\). Neck length, \(L\), and neck opening area, \(S\), are set to 8.5mm and 60x10\(^{-6}\) m\(^2\) respectively.

**Table 9 Helmholtz resonator frequency versus cavity volume**

<table>
<thead>
<tr>
<th>(V) (m(^3))</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>4</td>
</tr>
<tr>
<td>0.01</td>
<td>39</td>
</tr>
<tr>
<td>0.0001</td>
<td>394</td>
</tr>
<tr>
<td>0.000001</td>
<td>4466</td>
</tr>
<tr>
<td>0.0000001</td>
<td>18506</td>
</tr>
</tbody>
</table>

Table 9 shows that high frequency oscillation is theoretically possible as the cavity volume is reduced. However the dimensions become almost unusable. Using the dimensions of neck length and cross-sectional opening area as given, the cavity volume required to produce a frequency of 18.5kHz is 100x10\(^{-9}\) m\(^3\), which if spherical as in Figure 13, requires a cavity diameter of 5.76mm. It is easy to see that such a cavity size is impractical compared with the neck diameter of 8.74mm.

The second concern that became obvious when using the apparatus shown in Figure 13 was the dynamic response of the resonator, i.e. the time taken to set up the standing
wave. While this may well be satisfactory in situations where there is a relatively constant air-flow over the neck, for the purposes of this research it was considered to be contrary to point 3 in the criteria given in section 4.2 Acoustic transducers which is that the tone generator be:

“Efficient with good dynamic response. It should have a high dynamic response and yet be sensitive enough to react to small pressure gradients”
5.3 Summary

The Helmholtz resonator, while an excellent tone generator, as evidenced by its original use by Helmholtz in isolating specific frequencies from complex tones, was considered not ideal for the task required for this research. From observation with the apparatus shown in Figure 13, the time taken for the resonator to reach full oscillation was considered too long compared to the duration of a typical exhaust pressure wave, Blair [31] (1999, p. 290). Allied with concern over oscillation startup is the resonant frequency range. Table 7, Table 8, and Table 9 show that the resonant frequency of the resonator is quite limited considering its physical parameters and is considered to be too low in the audio range (20Hz – 20kHz) to provide sufficient temporal resolution for TOF measurements. Attention is therefore turned in Chapter 6 to the Labial Flue Pipe.
CHAPTER 6

6.1 Labial Flue Pipe

Having discounted the Helmholtz resonator as a suitable tone generator for the required application, due primarily to its low resonant frequency, attention was turned to the Labial Flue Pipe. The basis for this consideration was that the pressure wave resulting from a valve opening or an interface rupturing is simply a moving mass of gas (air is assumed) that travels in relation to, and at a velocity proportional to, the pressure gradient in existence at the time of interface rupture, see Figure 10. Jeans [62] (1968, p127-142) states that this moving mass of air, if directed in a thin stream onto a blade or edge produces:

“Little Whirlwinds” or “Edge Tones”

These whirlwinds move along the blade or edge and repeat at a specific rate dependant on the velocity of the air stream and the distance the blade or edge is from the start of the free thin-stream emanating from the slit. Figure 16 illustrates a typical apparatus used to generate edge tones. Verge et al. [63] (1994, pp.1122-1123) and Jeans [62] (1968, Plate VII, p. 133) provide Schlieren, i.e. a photographic process used to show fluid flow, and smoke-trail images respectively showing the vortices that form as a result of edge tone generation.

Figure 16 Typical edge-tone apparatus
The labial flue pipe, illustrated in Figure 17, is an extension of the edge-tone apparatus, and consists of several critically linked components. These include:

- the slit or Wind Sheet Thickness, WST,
- the mouth
- the blade or edge of the labium
- the tube or cavity.

![Diagram of the edge tone apparatus with tube](image)

**Figure 17 Edge tone apparatus with tube**

The Wind Sheet Thickness (WST) is a small slit through which a jet of air under pressure exits. This air jet travels across the mouth and impinges on the labium edge or blade, which is the wedge shaped formation that forms the top cover of the tube or cavity. The action of the air jet from the WST hitting the blade or labium edge results in the creation of edge tones. If an edge tone is of similar frequency to the resonant frequency of the cavity or tube, then this particular frequency will sound ‘clear’, Jones [64], as an acoustic tone, or as Jeans [62] puts it, the tube will ‘speak promptly’. Fabre et al. [65] states:

“The interaction of the jet with the labium transfers energy towards acoustic standing waves in the pipe.”

Elder [66] quotes the work of Cremer [67], Ising [68], and Bechert [69] saying that the acoustic tone produced is the result of:
“...the coupling between the jet and the standing wave field of the resonator...”.

Rossing [2] defines the standing wave as:

“A wavelike pattern that results from the interaction of two or more waves... has regions of minimum and maximum amplitude called nodes and antinodes”

To an observer these nodes and antinodes appear to stand still for a given set of conditions. Berg et al.[53], describes the standing wave as the sum of superimposed waves that is:

“...progressing neither to the right nor to the left...”.

In the case of the open ended labial flue pipe, there are two sets of waves involved, those due to the edge tones resulting from the jet mechanism, and those due to the longitudinal vibrations which are induced in the column of air in the body of the pipe. The vibrational effects of the edge tones are generally considered insignificant compared with the vibrational effects of the column of air in the tube. Even considering the close coupling that exists between the jet mechanism and the column of air in the pipe tube, the tone of the labial flue pipe is predominated by the column of air in the pipe tube. Jeans [62] (1968 p. 133) states that:

“The energy of the latter vibrations (i.e. the vibrations of the air column in the pipe) is so much greater than that of the edge tones that the latter may almost be disregarded, and the vibrations of the whole structure treated simply as those of the air in the pipe”

However, Jeans [62] goes on to state that despite this energy difference, the edge tones nevertheless exert a:

“...certain slight influence on the tone of the pipe as a whole”.

52
But that the:

“The closeness of the coupling will usually draw the edge tone into agreement with the fundamental tone of the pipe...”. Jeans [62] (1968, p140).

This coupling or interaction between the edge tones and the tube produces the tone or ‘voice’ of the pipe. The pipe will only sound clear, i.e. not distorted, for one particular set of conditions. These conditions are discussed in section 6.3 Pipe Efficiency.

The other components of the labial flue pipe are:

1. The reservoir, which is used as a means of providing a continuous and steady flow of air to the WST. In essence the reservoir acts as a form of low pass filter to any perturbations in the incoming air flow.

2. The open area between the WST or slit and the labium is known as the Mouth and is normally open to atmosphere.

3. The languid, which forms the lower part of the WST, consists of a flat top the length of the WST slit and will sometimes have a shallow ramp-up from the reservoir (gray area in Figure 17), although from observation the inclusion or omission of this ramp has been seen to not noticeably affect the performance of the pipe. The languid also forms the start of the tube or cavity and it is from its front face that the length, $L$, of the tube is measured and hence the effective or acoustic length, $L'$, is determined.

The combination of the WST, Mouth and Labium are collectively known as the Jet-Mechanism of the flue pipe. Lord Rayleigh [59] (p. 376) said of the jet mechanism that it is of such importance:

“... as to demand all the consideration that we can give.”
Elder [66] says that it is of

“... historic interest as one of the” unsolved” problems of classical physics.”

However, it is not the focus of this work to undertake an investigation of this jet mechanism as a fundamental property, rather, the reader is directed to works of Beranek [4], Rayleigh [59, 70], Elder [66], Fletcher [56], Olson [71], Jeans [62], Verge et al [63], Außerlechner [72], Kuhnelt [73] and others who all provide good descriptions of the operation of the jet mechanism.
6.2 Labial Flue Pipe Design

Typical examples of labial flue pipes are seen in the tubular pipes arrayed in ‘sets’ or ‘ranks’ around a church or concert-hall musical organ. While tubular pipes are typical, square pipes are also possible. Fletcher [56] describes the effects of pipe construction and the materials used and discusses the merits or otherwise of tubular and square/rectangular pipes. For this research a square, rather than a tubular geometry was considered to be the most appropriate format for construction as it was considered to be:

- easier to manufacture
- more amenable to adjustment and re-configuration.

While there are many treatises on the sound production mechanism of labial flue pipes such as Außerlechner [72], Elder [66], Verge et al. [63], Ising [74], Coltman [75], and Fletcher et al. [56], to name just a few, they all lack the basic design methodologies required to manufacture such a pipe. There are designs available for tone generating pipes from authors such as Hopkin [76], Shepard [77], Liljencrants [78] or indeed from the informative Mechanical Musical Digest web forum which provides designs such as the Webb [79] “No Arithmetic Flute pipe” and the Kleinbauer [80] “Klein Whistle”. However, these are of fixed design with limited or no information available to allow variations of them be developed. No automated systems, or design tables were found in the public domain to assist in the design of such a tone generator. The labial flue pipe in essence is a rather simple device that can be manufactured relatively easily using basic materials. However, the end result can largely be a matter of trial and error. Typically, organ pipe manufacturers use age-old lookup tables that have been developed over time, and provide dimensions and suggestions for optimal materials for a given tonal or acoustic requirement and typically require many hours of pain-staking work fine-tuning the pipes. Different materials produce different timbres or tonal colours.
Fletcher [56] states that a tin rich tin-lead alloy produces a bright tone whereas a lead rich tin-lead alloy produces a more dull tone. Fletcher [56] also states that there is:

“...generally a clear audible distinction between the tone of metal pipes and of wooden pipes...”

Fletcher [56] (2008, p. 572) however, indicates that these tonal differences have more to do with geometry than anything else, with the walls of a wooden pipe having to be, by necessity, of greater thickness than the walls of a metal pipe to provide sufficient structural stiffness to negate adverse wall vibrations.

“...and so there can be no audible effect of wall material properties”.

For this reason Fletcher states that:

“No organ builder would contemplate making square pipes of thin metal...

No organ builder has been persuaded to use thin plywood for bass pipes.”

Organ or flue pipe manufacturing tables are closely guarded proprietary information. As such the design and development of a software suite was required to facilitate theoretical design and parametric alteration of the labial flue pipe prior to construction.

### 6.2.1 Labial pipe End Correction

When dealing with a resonant pipe there are two possible types, open and closed or stopped. The open pipe as shown in Figure 18 has a voice or speaking frequency versus length relationship as given in equation (6.1).

\[
L' = \frac{a_0}{2 \cdot f} \quad (6.1)
\]

Where \( L' \) is the effective or acoustic length of the pipe, \( a_0 \) is the local undisturbed speed of sound and \( f \) is the *speaking* frequency of interest.
The closed or stopped pipe as shown in Figure 19 has a speaking frequency versus length relationship of

\[ L = \frac{a_0}{4f} \]  \hspace{1cm} (6.2)

Again, where \( a_0 \) is the local speed of sound and \( f \) is the speaking frequency of interest. In Figure 19, \( L \) is the length of the pipe from languid to the stopper. There is no end-correction required in equation (6.2) because the tube end is closed.

Comparing equation (6.1) and equation (6.2) clearly shows that the open pipe has a frequency double that of the stopped or closed pipe, ideally assuming that \( L = L' \). This is reinforced by examining simulations of the pressure profile in both a closed and open pipe, using equations (6.1) and (6.2) for a first harmonic. See Figure 20, and Figure 21.
Figure 20 Open pipe pressure profile

Figure 20 shows a simulated pressure profile for the open pipe resulting from equation (6.1). Figure 21 shows the same simulation but for a closed pipe using equation (6.2). It is clear to see for the open pipe configuration in Figure 20, that the pressure at either end of the pipe is zero bar gauge. Whereas for the case with one end of the pipe closed, Figure 21, the pressure at the closed end is maximum. Since one of the key criteria for the tone generator is high frequency, the stopped pipe is of little value since its maximum oscillation frequency for a given set of conditions will be half that of the open pipe.
For a labial flue pipe, the ‘speaking’ voice or natural frequency of resonance is of critical importance, not only in terms of frequency, but also in terms of tonal purity. For the open pipe the speaking frequency is predominantly determined by the acoustic length of the pipe, \( L' \), as given by equation (6.1). However, with an open labial flue pipe as with the Helmholtz resonator, there are two openings to take account of: the tube end, and the mouth. The end-correction for a single opening pipe has already been presented in section 5.1.1 Effective Length. For a tubular labial flue pipe with two openings, Ising [74] provides end-correction equations for both the cavity, \( \Delta L_C \), and the mouth, \( \Delta L_M \), of a tubular labial flue pipe as equations (6.3) and (6.4).

\[
\Delta L_C = 0.34\sqrt{S} \tag{6.3}
\]

\[
\Delta L_M = 0.73\frac{S}{\sqrt{S_M}} \tag{6.4}
\]

\( S \) is the cross-sectional area of the pipe cavity and \( S_M \) is the cross-sectional area of the mouth opening. Ising’s equation (6.3) produces a similar result to Kinsler’s equation (5.3) for the end correction of the open tube. Ising shows that the overall effective or acoustic length of an open ended tubular labial flue pipe is as given in equation (6.5).
It should be obvious that a closed or stopped pipe will only have the end correction for the mouth, $\Delta L_M$, to be taken into account as $\Delta L_C$ ceases to exist. However, equations (6.3) and (6.4) relate to pipes of tubular construction, whereas it has already been determined that a square pipe offers benefits over and above a tubular one for the investigation at hand, see section 6.2 Labial Flue Pipe Design. Therefore, the equations for acoustic or effective length, $L'$, need to be adapted to take account of the change in geometry. Liljencrants [81] provides a solution, given in equation (6.6), for the effective or acoustic length, $L'$, of a pipe with a square or rectangular internal cross-section, i.e. A in Figure 22.

$$L' = L + 0.3H + 0.8 \frac{A}{\sqrt{B}}$$

(6.6)

The terms H, A, B and L are defined with reference to Figure 22. $L'$ is the effective or acoustic length of the pipe. A is the internal cross-sectional area of the pipe, i.e. the product of w x h, which is considered to be uniform throughout the length of the pipe. B is the area of the mouth or the open area between the outlet of the WST and the blade or labium edge. L is the length of the pipe from the face of the languid to the physical end of the pipe and H the width of the mouth from outlet of the WST to the blade.

Figure 22  Square Labial Flue Pipe with key design parameters indicated
6.3 Pipe Efficiency

It has already been shown that the pipe length, \( L \), is important in defining the tone or speaking frequency of the pipe, but equally important is the jet mechanism that produces the vortices which couple with the standing wave in the pipe, Elder [66], to produce the tone. Critical to this coupling are the design of the mouth and the Wind Sheet Thickness (WST) or jet. Together these components, i.e. the WST and the mouth, form what are collectively known as the jet-mechanism. There are four jet-mechanism parameters that are crucial to the ‘voicing’ of the pipe, and the correct manipulation of these parameters defines the purity, otherwise known as the efficiency of the produced tone. These are:

- the oscillation or tonal frequency required
- the thickness of the WST, or air jet from top to bottom, see Figure 17
- the mouth area, sometimes known as “mouth cut-up”
- the velocity of the air jet exiting the WST.

Ising [74] defines the efficiency, \( I \), of the jet mechanism in a labial flue pipe as given in equation (6.7).

\[
I = \frac{v \sqrt{WST}}{f \sqrt{H^3}}
\]  

(6.7)

Where \( v \) is the velocity of the air jet exiting the WST, \( H \) is the mouth width, \( f \) is the oscillation frequency and \( WST \) is the thickness of the air jet, see Figure 17. The efficiency or Ising number, \( I \), has an ideal range of between 2 and 3. A flue pipe with an Ising number of 2 is the most efficient and produces the purest of tone, i.e. a tone with the least number of harmonic overtones, if any. A pipe with an Ising number of 3
indicates an overblown pipe, i.e. one where there are overtones or harmonics present in the produced tone.

Rioux’s [82] (2001, p. 15) adapts Ising’s equation by substituting the air jet velocity $v$, in equation (6.7) with Bernoulli’s equation (6.8).

$$v = \sqrt{\frac{2(P - P_0)}{\rho}}$$  \hspace{1cm} (6.8)

Where $\rho$ is the density of the gas, air is assumed, $P$ is the driver pressure i.e. the high pressure air introduced into the foot in Figure 22, and $P_0$ is the pressure around the mouth which can usually be considered to be atmospheric. Combining equations (6.7) and (6.8) gives the Ising number, $I$, or the efficiency of the jet mechanism in terms of pressure difference, air density, WST and mouth width, $H$.

$$I = \sqrt{\frac{2 \cdot \Delta P / \rho \cdot \sqrt{\text{WST}}}{f^2 \cdot \sqrt{H^3}}}$$  \hspace{1cm} (6.9)

From equation (6.9) it can be seen that the Ising number, $I$, is proportional to the square root of the pressure ratio between the blowing pressure driving air into the WST and the pressure around the mouth, all else being equal. Over blowing a labial flue pipe causes the number and intensity of unwanted harmonics to increase. This increase in harmonic content, which is theoretically signified by an increase in the Ising number, results in an increase in the resonant frequency from the pipe. Bhargava [83] reported that at a given differential pressure, an experimental labial pipe produced a fundamental frequency of 490Hz. By increasing that differential pressure by a factor of 5, Bhargava found that the fundamental pitch was seen to rise to 505Hz. Using equation (6.10), where $n$ is the harmonic number, $f$ is the frequency of the fundamental or base tone, and $f_n$ is the
frequency of the semitone required this frequency shift from 490Hz to 505Hz is approximately 52% of a standard chromatic musical semitone increase in pitch.

\[ f'_n = f \times \left( \frac{12}{\sqrt[12]{2}} \right)^n \]  

(6.10)

Bhargava also reported that increasing the differential pressure to a factor of 8 resulted in the frequency more than doubling to 1052Hz. Theoretically, a doubling of the fundamental frequency will result in a frequency of 980Hz. The frequency reported by Bhargava due to an 8x increase in blowing pressure is, in fact marginally sharper, i.e. higher in pitch, than a 13\textsuperscript{th} chromatic note of the fundamental, which ideally, from equation (6.10), has a frequency of 1038Hz. This sharpening or increase in pitch of the fundamental tone is the primary effect of over-blowing the pipe with the amount of sharpness equivalent to the pressure differential. This effect is seen in Table 10 where a labial pipe with a fundamental frequency of 3.27kHz was subjected to an increasing differential pressure. In each case the first four harmonics (fundamental, 1\textsuperscript{st}, 2\textsuperscript{nd} and 3\textsuperscript{rd}) were monitored for frequency and magnitude. Also indicated in Table 10 is the theoretical Ising number for the particular condition measured.

<table>
<thead>
<tr>
<th>Harmonic #</th>
<th>Measured Frequency (Hz)</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fundamental</td>
<td>3270 3405 3420 3480</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1\textsuperscript{st}</td>
<td>5640 6800 6835 6955</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2\textsuperscript{nd}</td>
<td>9805 10200 10255 10435</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3\textsuperscript{rd}</td>
<td>13070 13600 13680 13910</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Harmonic #</th>
<th>Measured Magnitude</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fundamental</td>
<td>0.215 0.223 0.224 0.234</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1\textsuperscript{st}</td>
<td>0.010 0.039 0.095 0.105</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2\textsuperscript{nd}</td>
<td>0.010 0.015 0.021 0.075</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3\textsuperscript{rd}</td>
<td>0.004 0.010 0.002 0.009</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| Ising # | 1.9 | 2.7 | 3.4 | 3.9 |
Figure 23, Figure 24, Figure 25 and Figure 26 graphically show the frequency spectrums corresponding to the pressure differentials, and theoretical Ising numbers which are given in Table 10.
Figure 23, Figure 24, Figure 25 and Figure 26 clearly show the increase in the harmonic content due to the increase in differential pressure across the jet-mechanism.

The effect of the higher harmonics at the higher differential pressures, especially as their amplitude increases relative to the fundamental, will produce audible effects that are easily discernable and noticeably different to the response at the 0.1 Bar pressure differential. To emphasise this point, Figure 27 and Figure 28 show sections of the acoustic oscillation results for a pipe with differential pressures of 0.1 bar and 0.4 bar.
respectively. Clearly seen in the plots is the effect of the increased harmonic content, which corresponds to the FFT results, illustrated in Figure 23 and Figure 26.

Figure 27 Oscillation waveform using 0.1 Bar ΔP

Figure 28 Oscillation waveform using 0.4 Bar ΔP

Figure 27 shows an easily recognised and reasonably linear sinusoid. This is consistent with an Ising, or efficiency number of around 2, theoretically calculated at 1.9 in Table 10 for a differential pressure of 0.1 Bar. Increasing the differential pressure to 0.4 Bar produces the distorted oscillation that is seen in Figure 28 which is characteristic of an Ising or efficiency number of 3 or greater. The theoretical Ising number for this condition is given as 3.9 in Table 10.
The difference between the acoustic responses is clear to see in Figure 27 and Figure 28. As differential pressure increases the magnitude and effect of over blown artefacts become more dominant in the frequency and shaping of the oscillation tone. This indicates that great care must be taken in the design of the jet mechanism of the labial flue pipe to ensure that over-blow conditions are either minimised or eliminated altogether.
6.4 Labial Flue Pipe Design Software

Following on from the investigation of Ising [74], Rioux [82], and Liljencrants [81], a software suite was developed to facilitate the design of the labial flue pipe. The LabVIEW based software, the front panel of which is shown in Figure 29, is based around a combination of Liljencrants’ equation (6.6) and the Ising/Rioux equation (6.9). When the software is run, the results of the design are displayed in the lower ‘Output Conditions’ panel. For the design given in Figure 29, the pipe has a theoretical design frequency of 3232.05 Hz and an Ising efficiency number of 2.00341.

![Figure 29 Labial Flue Pipe Design Software Front Panel](image)

The design parameters for the labial flue pipe used for the experiment that generated the data given in Table 10 is shown in Figure 30. The theoretical design frequency of this pipe is 3220.51 Hz as seen in Figure 30. The experimental results using this pipe design given in Table 10 show a measured fundamental frequency of 3270 Hz. This difference in frequency between theory and practice is just less than 50 Hz. Using equation (6.10), this is a difference of 26.4% of a chromatic semitone. More precise
control of the differential pressure may have resulted in reduction of this frequency error. However, the software, and resultant design, operated well for the purposes at hand.

Figure 30 Theoretical pipe design parameters for Table 10 data
6.5 Ising versus Reynolds

It has been shown that the efficiency or spectral purity of the labial flue pipe can be characterised by a term known as the Ising number, $I$, and that ideally, a labial flue pipe should have an Ising number of between 2 and 3. Using the software illustrated in Figure 29 and Figure 30, there is theoretically no limit to the range that can be assigned to the parameters of the labial flue pipe. For instance, Table 11 shows the results for a theoretical labial flue pipe as shown in Figure 22, with the following physical parameters, that is subjected to different differential pressures.

- $W = h = 1\text{cm}$
- $L = 4\text{cm}$
- $H = 9\text{mm}$
- $WST = 0.9\text{mm}$

### Table 11 Theoretical oscillation and jet mechanism parameters versus $\Delta P$

<table>
<thead>
<tr>
<th>$\Delta P$ (Bar)</th>
<th>$I$</th>
<th>$Re$</th>
<th>Jet Velocity (m/s)</th>
<th>$F$ (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01</td>
<td>0.43</td>
<td>2430</td>
<td>40.60</td>
<td>3343.7</td>
</tr>
<tr>
<td>0.1</td>
<td>1.35</td>
<td>7683</td>
<td>128.20</td>
<td>3343.7</td>
</tr>
<tr>
<td>0.22</td>
<td>2.00</td>
<td>11382</td>
<td>191.17</td>
<td>3343.7</td>
</tr>
<tr>
<td>0.5</td>
<td>3.01</td>
<td>17181</td>
<td>286.70</td>
<td>3343.7</td>
</tr>
<tr>
<td>1.0</td>
<td>4.26</td>
<td>24297</td>
<td>405.50</td>
<td>3343.7</td>
</tr>
<tr>
<td>4.0</td>
<td>8.52</td>
<td>48594</td>
<td>810.98</td>
<td>3343.7</td>
</tr>
</tbody>
</table>

The data shown in Table 11 would indicate that given the particular geometry of the pipe, oscillation is theoretically possible even at extremely high pressure levels. However, experimentation shows that this does not happen and in fact at extremes of differential pressure, it has been seen that oscillation did not occur. As a result, the connection between the Ising number and the Reynolds number associated with the WST in the labial flue pipe was examined.
Reynolds, $Re$, numbers are used to determine whether a particular flow is laminar or turbulent. Laminar flow, which is characterized by a smooth, layered flow, occurs at low Reynolds numbers. Turbulent flow, which is characterized by a chaotic flow, occurs at high Reynolds numbers. In the case of a pipe or duct, the Reynolds number is given by equation (6.11).

$$Re = \frac{\rho \cdot v \cdot l}{\mu}$$

(6.11)

In equation (6.11), $\rho$ is the density of the liquid (air = 1.2 kg·m$^{-3}$), $v$ = is the velocity of the flow which is determined by the use of Bernoulli (equation (6.8)), $\mu$ is the dynamic viscosity of the liquid (air $\approx$ 1.82x10$^{-5}$ Pa·s), and $l$ is the characteristic length, which is the dimension that has the greatest significance on the pattern of the flow. In the case of the air jet in the labial flue pipe this length is the thickness of the jet or the WST thickness. Flows with characteristic Reynolds numbers up to approximately 3500 are generally considered laminar, while flow with characteristic Reynolds numbers above this are considered to be turbulent. Equation (6.12) is used to determine the characteristic Reynolds number for the air jet in the labial flue pipe.

$$Re = \frac{\sqrt{2 \cdot \rho \cdot \Delta P \cdot WST}}{\mu}$$

(6.12)

Figure 32 shows a plot of experimental results versus theoretical Ising or efficiency number and Reynolds number for a labial flue pipe. The experiment was performed by applying a controllable differential pressure to the tone generator with a given mouth and WST geometry. Figure 31 shows a diagram of the experimental apparatus used. The differential pressure was slowly increased from zero until oscillation was seen to start. The differential pressure across the mouth continued to be increased until oscillation stopped. Each curve in Figure 32 represents a different jet-mechanism geometry with valid oscillation ranges indicated by the solid lines. The dashed lines indicate that
although a differential pressure was in existence across the pipe, no oscillation was observed. During the experiment it was noted that for some pipe geometries and differential pressures, oscillation did not occur within the ideal Ising range of 2 to 3, the gray shaded region in Figure 32. Therefore, even though the software shown in Figure 30 might theoretically indicate valid oscillation and efficiency for a given geometry, the pipe in practice may choke or not oscillate under certain conditions.

Figure 31 Diagram of Ising-Reynolds experimental apparatus
Figure 32 Reynolds versus Ising for various pipe geometries and differential pressures
What is interesting to note from Figure 32 is that for a mouth width of 9mm, noted by the $H = 9\text{mm}$ indicator on the three left most curves, even though oscillation occurred as indicated by the markers on the three curves, all oscillation ceased before the optimum Ising range could be achieved despite continuing to increase the differential pressure as indicated by the dashed lines. Equally for a pipe with a mouth width, $H = 3\text{mm}$ and a $WST = 0.3\text{mm}$: right most curve in Figure 32, oscillation did not commence until the theoretical Ising number was outside the optimum range. The case where the mouth width, $H = 5\text{mm}$ and the WST = 0.3mm, oscillation is seen to start just below the optimum Ising value of 2 and continues oscillating through to an Ising value of just over 4 before the jet mechanism chokes, stopping further oscillation.

These novel and significant results indicate that while Ising is important in the design of the labial flue pipe it should not be used in isolation. Monitoring the Reynolds number of the WST jet flow is equally important. From the data presented in Figure 32, ideal Reynolds numbers appear to be in the range of 1500 to a maximum of approximately 7000, although at the upper end of this range the pipe will have an Ising or efficiency value of approximately 3 and will hence be over-blown, thereby producing unwanted harmonics as seen in the FFT plots of Figure 24 and Figure 25. What is significant about this inter-relationship between Ising and Reynolds is seen from the data presented in Table 11. Table 11 shows that for the particular flue pipe outlined, with a pressure differential of 0.22 Bar, an Ising number of 2 is theoretically possible. However, experimental results for this particular flue pipe configuration: $H=9\text{mm}$, $WST = 0.9\text{mm}$, the left most curve in Figure 32, indicates that oscillation has ceased well before an Ising number of 2 is achieved. This novel result indicates that monitoring of the Reynolds number is as important in determining optimum oscillation conditions as the
Ising or efficiency number. To achieve an Ising number of between 2 and 2.5, in order to minimise over-blown harmonic effects that start to be introduced into the oscillation spectrum as the efficiency number approaches 3, the optimum Reynolds range, for a minimum of harmonic distortion appears to be approximately 1500 to approximately 5500.
6.6 Tone Generators

The software shown in Figure 30 is capable of designing labial flue pipes of any theoretical frequency. Table 12 shows the geometries required for an ultrasonic frequency of 44.18kHz using a pressure ratio of 1.1 Bar.

Table 12 Theoretical frequency capability of the labial flue pipe through geometry variation using a pressure differential of 0.1 Bar

<table>
<thead>
<tr>
<th>Frequency (kHz)</th>
<th>h=W (m)</th>
<th>L (m)</th>
<th>H (m)</th>
<th>WST (m)</th>
<th>I</th>
<th>Re</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.18</td>
<td>0.0015</td>
<td>0.002</td>
<td>0.0008</td>
<td>0.0005</td>
<td>2.04</td>
<td>3015</td>
</tr>
</tbody>
</table>

Table 12 indicates that high frequency at reasonable geometries is possible using a labial flue pipe. This is in keeping with the criteria for the tone generators as given in section 4.2 Acoustic transducers. However, as was shown in Figure 32, such a theoretical design is not always optimal. It has previously been indicated that tone generators with frequencies in the audio range (20Hz to 20kHz) be utilised to facilitate setup and optimisation. In fact, even at the extremes of the audio range, such frequencies become difficult to use. Therefore, it was decided that frequencies in the range of 3kHz to 15kHz be used. As such four labial flue pipe tone generators were designed using the software shown in Figure 30, using pressure ratios determined from experimental observation at each of the TG1 and TG2 tone generator positions indicated in Figure 33, of approximately 1.1 Bar and 1.05 respectively.
Table 13 shows the design parameters of these four tone generators.

Table 13 Labial flue pipe parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>3kHz</th>
<th>6kHz</th>
<th>10kHz</th>
<th>15kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TG1</td>
<td>TG2</td>
<td>TG1</td>
<td>TG2</td>
</tr>
<tr>
<td>P/P₀</td>
<td>1.1</td>
<td>1.05</td>
<td>1.1</td>
<td>1.05</td>
</tr>
<tr>
<td>W=h (mm)</td>
<td>10</td>
<td>10</td>
<td>5.0</td>
<td>5.0</td>
</tr>
<tr>
<td>L (mm)</td>
<td>40</td>
<td>40</td>
<td>9.5</td>
<td>9.5</td>
</tr>
<tr>
<td>H (mm)</td>
<td>5</td>
<td>4</td>
<td>2.5</td>
<td>2.0</td>
</tr>
<tr>
<td>WST (mm)</td>
<td>0.3</td>
<td>0.3</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>I</td>
<td>1.9</td>
<td>1.9</td>
<td>2.1</td>
<td>2.2</td>
</tr>
<tr>
<td>Re</td>
<td>2588</td>
<td>1830</td>
<td>4269</td>
<td>3018</td>
</tr>
<tr>
<td>F (kHz)</td>
<td>3.22</td>
<td>3.2</td>
<td>6.27</td>
<td>6.17</td>
</tr>
</tbody>
</table>

Figure 34 shows the finished designs with the labium covers removed for clarity. Two of each tone generator are required.
Experimentation using these tone generators showed that at the indicated conditions there was good frequency agreement between theoretical and actual designs. See Table 14, for the 3kHz, 6kHz and 10kHz tone generators. However, the 15kHz design failed to oscillate, regardless of the pressure ratio applied. From this it would appear that the envelope limit for the particular geometry used was exceeded.

The 3kHz tone generators have discrepancies that are in the region of 10% or less of a chromatic semitone. The 6kHz tone generators have discrepancies of just under a chromatic semitone and the 10kHz tone generators have discrepancies of less than a half semitone. All these discrepancies could be reduced if desired by fine adjustment of the differential pressure, the physical length or perhaps by adjustment of the mouth width,
although this latter will have consequences for the efficiency of the pipe without also adjusting the WST. However, for the purposes of this investigation these frequencies and discrepancies are more than acceptable.

Table 14 Actual versus theoretical tone generator frequencies

<table>
<thead>
<tr>
<th>Frequency(Hz)</th>
<th>3kHz</th>
<th>6kHz</th>
<th>10kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tone Generator</strong></td>
<td><strong>TG1</strong></td>
<td><strong>TG2</strong></td>
<td><strong>TG1</strong></td>
</tr>
<tr>
<td><strong>F&lt;sub&gt;Theory&lt;/sub&gt; (kHz)</strong></td>
<td>3.22</td>
<td>3.20</td>
<td>6.3</td>
</tr>
<tr>
<td><strong>F&lt;sub&gt;Actual&lt;/sub&gt; (kHz)</strong></td>
<td>3.23</td>
<td>3.18</td>
<td>6.5</td>
</tr>
<tr>
<td><strong>Δ¢</strong></td>
<td>5.4</td>
<td>-10.9</td>
<td>54.1</td>
</tr>
</tbody>
</table>

Table 14 introduces the term cent, ¢, which is a musical term to indicate the error a given tonal pitch has in relation to a known, constant pitch. There are 100¢ or divisions in a standard chromatic semitone, therefore, for the 3kHz, TG1 result, the difference between the theoretical calculated frequency, 3.22kHz and the actual measured frequency, 3.23kHz is only 5.4¢ or 5.4% of a chromatic semitone.
6.7 Summary

In this chapter, the labial flue pipe as a tone generator has been examined. Its start up mechanism, and designed for oscillation frequency were observed to be faster and higher respectively than for the Helmholtz resonator. The frequency range of the labial flue pipe has also been shown to be quite large while maintaining usable physical geometries, although it was seen that at 15kHz the theoretical design did fail to produce a working tone generator. Despite this, the labial flue pipe was found to be easily adjustable both for pitch and for tonal efficiency or purity.

The tonal purity of the flue pipe, as proffered by Ising, was been investigated by examining the different theoretical Ising numbers resulting from varying differential pressures across a given tone generator geometry. While the Ising number is an excellent guide in determining the efficiency of the pipe, it has been shown that Ising should not be used in isolation, but should be used in conjunction with the Reynolds number resulting from the jet geometry. A novel and significant relationship has been identified linking the Ising number and the Reynolds number of the jet mechanism. From this, it has been shown that although a theoretical design may indicate that a pipe will oscillate with ideal theoretical efficiency, in practice it may not oscillate as expected, as shown in Figure 32.
CHAPTER 7

7.1 Hardware and Data Acquisition

With tone generator designs in place, see Table 13, the next consideration was the experimental apparatus and data capture mechanism. As shown in Figure 33, the apparatus consisted of a pressure vessel connected to a blow-down pipe through a valve or controllable pressure-release system. Integrated into the blow-down pipe were suitable attachment points for the tone generators or labial flue pipes as well as a suitable heating system to allow the air in the pipe be heated. In addition, pressure sensors are required to monitor atmospheric, $P_0$, driver pressure, $P$, as well as the transient pressures at the tone generator positions. Also required are temperature sensors to monitor the air temperature, both the ambient or nominal air temperature and the heated air within the blow-down pipe, mid-way between the tone generators. These temperature sensors were k-type thermocouples monitored by a calibrated comark 2001 temperature meter.
7.2 Pressure Vessel

The pressure vessel used was a steel cylinder with a total volume of 0.00335 m$^3$ as shown in Figure 35. The cylinder had an integrated 8 bar pressure safety release valve as well as attachments for pressure and temperature sensors as shown in Figure 36.

![Figure 35 Pressure Vessel Dimensions](image1)

The cylinder is pressurized using a laboratory air supply through a manually controlled pressure regulator.

![Figure 36 Pressure cylinder showing pressure connections, sensors and regulator](image2)
7.3 Pressure Release Mechanism

The pressure release mechanism that acts as the interface between the pressure cylinder and the blow-down pipe must offer reasonably high and controllable cyclic rates and be easy to control. Several options presented themselves as discussed in the following sections.

7.3.1 QUB

The ‘QUB SP’ or Queens University Belfast Single Pulse rig is an apparatus designed by Blair [31] and used by Long [84] and others for the purposes of investigating unsteady gas flows in exhaust systems. The QUB consists of a cast iron pressure vessel with a volume of 912cm$^3$ (0.000912m$^3$). The valve mechanism is a sliding plate with a 25mm orifice that is actuated by a pneumatic hammer. When the valve is actuated by the hammer, a pressure pulse or wave is induced into the blow-down pipe. The resultant pressure wave is a single pulse or wave that propagates into the blow down pipe every time the valve is actuated. Following a valve actuation, the valve requires resetting and the system re-pressurised. A diagram of what this might look like, with integrated tone generators is given in Figure 37.

Figure 37 Example of the QUB SP rig applied to tone generator blow-down pipe
At first glance the QUB apparatus appeared to satisfy the requirements for the investigation at hand, however, a QUB analogue was discounted for several reasons:

- No clear design schematics or drawings could be found to facilitate the design
- Long [84] describes a shortcoming of the QUB rig, whereby the o-rings on the valve required regular maintenance and that regardless of the lubrication used the valve leaked and actuation and pulse duration times were variable.
- It was and is a one shot device, which while suitable for Blair’s and Long’s experiments, was considered too limiting for the investigation at hand due to its one-shot operation.

It is this last point that was main factor in deciding not to use a QUB style pressure release system.

7.3.2 Engine head

The investigation requires the controlled release of pressurized gas. One such release mechanism that appeared to fit the requirement was an engine exhaust valve such as found on the Honda CLR125 engine. This engine head has two valves, inlet and exhaust that are operated by a cam and rocker arm mechanism. The apparatus, with such a valve mechanism would consist of the pressure cylinder described earlier, connected to the engine head which in turn would be connected to the blow-down pipe via its exhaust manifold. By rotating the cam, the valve can be made to actuate thereby inducing a pressure pulse into the blow-down pipe. Figure 38 shows the engine head without the CAM or rocker arm assemblies with the exhaust valve indicated. However, the actuation control of the cam and valve presented significant issues. Several options presented themselves such as the basic lever, impact or motor control.
7.3.2.1 Lever Control

In order to actuate the valve, the rocker arm needed to be manipulated by rotating the cam. The exhaust valve was maintained in a closed position by the action of two overlapping coil springs held in place by a washer that can be seen in Figure 38. It was proposed that the cam be rotated by the action of a lever affixed to it as shown diagrammatically in Figure 39.
Several experiments were carried out on the engine head using this configuration to determine what torque would be required to actuate the exhaust valve using a mass acting through a point on the lever a distance, $d$, from the centre of the cam. Ignoring the effects of air pressure, $F_a$, that acts on the face of the valve when the cylinder is pressurized, the basic force to be overcome by the lever and mass was that of the springs, $F_s$, that hold the valve closed: see Figure 39. It was found that a torque of 10.7 Nm was required to actuate the valve which is equivalent to a mass of 5.32kg acting at a distance of 20.5cm from the centre of the cam.

In an effort to reduce this torque one of the valve springs holding the valve closed was removed. With the spring force, $F_s$, reduced the torque required was reduced to 2.92Nm or the equivalent of a 1.45kg mass operating through a point 20.5cm from the centre of the cam. The forces involved are not excessive and indeed if the distance, $d$, is increased, the force required reduces. However, rapid sequencing and iteration repeatability of the mechanism was somewhat questionable. The apparatus was a one-shot device similar to the QUB in that the lever, and mass would need to be reset after every iteration. Even if the lever was hand manipulated, repeatability becomes challenging.

7.3.2.2 Impact Control
Since a lever action was unsatisfactory from a sequencing and repeatability point of view, the idea used by Blair, a hammer of some sort, to directly impact and hence actuate the valve was examined. Since the hammer would yield a force in line with the stem of the valve, the force required was easily determined by applying weight vertically in line with the end of the valve stem until such time as the valve actuated, see
Figure 40. It was found that a mass of 17.1kg was required on the altered valve, i.e. with one of the coil springs removed, to actuate the valve. This equates to a force of 167.7N.

This is quite an excessive force and it is likely that the stem of the valve may not survive repeated impacts. Allied with this excessive force was the question of speed of operation and repeatability. Unlike the QUB where the valve consists of a sliding plate with orifice, the engine valve requires a reciprocating, or up and down movement to initiate a finite pressure pulse. This reciprocating motion, without the action of the cam and rocker arm assembly is difficult to achieve, especially at the rates required to match or even approach the QUB performance which Blair [31] states is of the order of 8ms.

\[ \zeta = \frac{\tau}{d} = \frac{F_m}{F_v} \]  

7.3.2.3 Effects of Air pressure

It was obvious that the action of the cam and rocker arm assemblies provided a significant gain advantage in actuating the valve when compared to a directly applied force. Using the reduced torque value of 2.92Nm obtained earlier for the lever operation, the amplification factor, \( \zeta \), of the cam, rocker, and lever assembly can be determined from equation (7.1).
Where $F_m$ is the force due to the mass acting on the lever at distance $d = 20.5\text{cm}$ which produces a torque $\tau$ of 2.92Nm. $F_v$ is the force required to overcome the force of the spring holding the valve closed, $F_s$. Therefore the cam-rocker arm and lever assembly has an amplification factor of 0.085, which is the reverse gain in the system, i.e. looking from the valve out to the mass at the end of the lever. This was confirmed by looking at the masses involved, 1.45kg at the end of the lever versus 17.1kg applied directly to the head of the valve stem.

\[
\frac{1.45\text{kg}}{17.1\text{kg}} = 0.085 = \xi
\]  

(7.2)

However, the effect of the air pressure in the pressure vessel that acts on the face of the valve has so far been ignored. This air pressure will act as a force, $F_a$ complimenting the force exerted by the spring in maintaining the valve in a closed position. The valve face is 25mm in diameter giving a surface area of $490\times10^{-6}\text{m}^2$. The force, $F_a$ acting on the face of the valve is given by Ivanoff [85] (p252) as equation (7.3).

\[
F_a = P \cdot A = P \cdot \frac{\pi \cdot D^2}{4}
\]  

(7.3)

Where $P$ is the air pressure in Pascals and $D$ is the diameter of the area the pressurized air acts on. Table 15 details how the force exerted by the pressurized air on the face of the valve varies as pressure varies. This data is presented graphically in Figure 41.

### Table 15 Air pressure versus force exerted on the valve face

<table>
<thead>
<tr>
<th>Gauge Pressure (bar)</th>
<th>$F_a$ (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>49.1</td>
</tr>
<tr>
<td>2</td>
<td>98.2</td>
</tr>
<tr>
<td>3</td>
<td>147.3</td>
</tr>
<tr>
<td>4</td>
<td>196.3</td>
</tr>
<tr>
<td>5</td>
<td>245.4</td>
</tr>
</tbody>
</table>
Therefore, the 167.7 N already determined as being the force required to directly actuate the valve to overcome the stiffness of the spring now requires significant extra force to overcome the combination of the spring and the air pressure within the cylinder. From this the total torque, $\tau$ required for a given air pressure can be determined by manipulating equation (7.1) where the force due to air pressure, $F_a$, acting on the face of the valve must be added to the force due to the spring, $F_s$.

Figure 42 shows a plot of the result of this analysis for a lever length of 20.5cm with an extrapolation, dotted section, back to zero bar gauge cylinder pressure. This backwards extrapolation intercepts the torque axis at 2.92Nm which was the torque determined to be required to overcome the spring force alone ignoring the effect of cylinder air pressure. It should be obvious that changing the lever length will have a significant impact on the resultant force. However, the rig is still only a one shot device, and regardless of the actuation methodology employed, impact or lever, limitations remain.
7.3.2.4 Motor Control

It has been shown that the use of a lever mechanism to actuate the engine valve is practical, however, it has some limiting drawbacks such as repeatability and speed of operation. In essence, it was considered a one-shot device. Impact actuation of the valve stem was considered equally impractical. Therefore, if the engine head was to be suitable as a pressure release mechanism an alternate means of actuating the engine valve was required which can provide repeatable valve opening times and offer a multi-shot capability. With this in mind the next option considered is the use of a motor of some format to drive the cam and hence actuate the valve. Blair [31] (1999, p290) used a valve opening time of 8ms on his QUB which he says is equivalent to an engine operating at 3000 rpm, therefore using this as a datum, a motor applied directly to the cam must be able to deliver a torque of 4.63Nm at a cylinder air pressure of up to 2 bar gauge, see Figure 43, and at a speed up to 3000 rpm.
7.3.2.5 Servo Motor

The servo motor requires a finite spin up time with the available torque increasing as velocity increases. However, since the required rpm rate was not instantly achieved, the motor required a spin-up time to achieve correct RPM. The valve cannot be actuated until the required rpm has been achieved therefore, some form of clutch mechanism was required to engage the motor with the cam when optimum conditions have been achieved. There was then the question of event control. The motor will be required to be disengaged from the cam/valve once the required number of valve actuations has been achieved, be that a single one-shot event or multiple events, and any energy in the mechanism dissipated such that the system stops relatively instantly preventing extra pressure pulse releases. This required a braking scheme of some form that not only would de-clutch the motor but also arrest any inertia the cam and associated mechanics may have. Such a scheme was deemed impractical for the operation at hand due to the required machining/fabrication cost and time required.

Figure 43 Torque required at 2 Bar gauge cylinder pressure
7.3.2.6 Stepper Motor

The stepper motor, for the application at hand, at first glance offers some advantages over a servo motor. The stepper motor requires very little or no control circuitry, they have very precise positioning resolution and have high, low-speed torque. However, stepper motors do have limitations, such as, slippage at high rpm which is compounded by the fact that the available torque drops off dramatically as rpm increases. An example of this torque versus rpm relationship is given in Figure 44 for the Nanotec ST11018L8004 stepper motor. What was also evident from Figure 44 was that the power required, even at modest rpm, was quite high. Achieving 3000 RPM at a torque of 4.63Nm if not more was not readily feasible using this stepper motor.

![Figure 44 Torque vs RPM relationship](image)

This analysis indicates that while the engine head valve appeared to offer an ideal means of pressure pulse release, control of the valve was problematic. Under simple lever control it was similar to the QUB in that it was a one-shot device, but it has a disadvantage from the QUB in that control of the valve opening time was somewhat more arbitrary. Controlling the valve using a motor or hammer equally presents difficulties without seriously modifying the engine head.
7.3.3 Pneumatically actuated Ball valve

Having discounted the engine head as a viable pressure release mechanism, the option of using a pneumatically actuated ball valve was examined. The ball valve offers benefits as a pressure release solution in that it is self contained and easily controllable. The ball valve selected was the FlowTech PW F04 and is seen operationally in Figure 45. The ball valve is actuated by pneumatics which are controlled by switching a 230V AC control signal.

Figure 45 FlowTech ball valve in system

Figure 46 shows a captured acoustic response from the actuation of the ball valve shown in Figure 45, where the response time of the valve is deduced to be commensurate with the initiation of the acoustic waveform through to the start of the decay of the waveform, i.e. from when the valve opens to when it closes. In the case shown in Figure 46, this is approximately indicated by the red cursor lines or markers which are positioned at x-axis time values of approximately 25ms and 96ms respectively.
This actuation time, while not in Blair’s [31] 8ms range, was considered reasonable for the investigation at hand, especially given the controllability and flexibility such a solution offers. However, the results from the ball valve were less than satisfactory due to the geometry of the valve opening. From a fully closed position, the valve starts to open creating a crescent shaped orifice through which pressurized air starts to escape. This orifice continues to increase in size releasing more pressurized air until fully open. This process is then reversed from fully open to fully closed. Such a continual graduated pressure release during valve opening increases the time taken for the oscillation to reach fully amplitude. This was seen by comparing the operation of the ball valve with that of a solenoid valve for similar pressure differentials. The solenoid valve used was the M&M International D892DPV which is described in a later section.

Figure 47 shows 16ms of the leading portion of the waveform shown in Figure 46, resulting from the actuation of the ball valve. Figure 48 shows 16ms of the leading portion of the waveform resulting from the actuation of a solenoid valve.
Both the plots in Figure 47 and Figure 48 have the 60% threshold position indicated. It is clear to see that the solenoid valve offers a faster acoustic response time than does the ball valve. Table 16 shows a comparison between calculated temperature results from the use of both a ball valve and a solenoid valve pressure release system. Figure 49 shows the standard deviation of the temperature results resulting from the actuation of the ball valve and the solenoid valve. As with the data shown in Figure 47 and Figure 48, it is clear from the results in Table 16 and Figure 49 that the solenoid valve offers
dramatically better results for temperature results, average, peak-to-peak and standard deviation, than the ball valve for similar conditions.

Table 16 Comparison of average and pk-pk temperature results using ball and solenoid valves

<table>
<thead>
<tr>
<th>Valve Type</th>
<th>Temperature (K)</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
<th>60%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ball</td>
<td>Average</td>
<td>586.44</td>
<td>700.64</td>
<td>678.49</td>
<td>727.23</td>
<td>694.94</td>
</tr>
<tr>
<td></td>
<td>Pk–Pk</td>
<td>717.85</td>
<td>1151.13</td>
<td>990.04</td>
<td>1774.65</td>
<td>1750.01</td>
</tr>
<tr>
<td>Solenoid</td>
<td>Average</td>
<td>328.27</td>
<td>335.72</td>
<td>332.78</td>
<td>339.77</td>
<td>348.58</td>
</tr>
<tr>
<td></td>
<td>Pk–Pk</td>
<td>356.14</td>
<td>388.14</td>
<td>352.84</td>
<td>512.85</td>
<td>663.55</td>
</tr>
</tbody>
</table>

Figure 49 Temperature Standard Deviation: Ball valve versus Solenoid valve

The use of the ball valve, while it does minimise turbulence thereby offering improved waveform linearity, the relatively slow and incremental pressure release mechanism seen in Figure 47, resulting in a correspondingly slow acoustic response ramp up rate has a demonstrated detrimental effect on the determination of $\Delta t$ and subsequently on the calculated temperature results. Verge et al. [63] (1994, p.1120) indicates a similar
phenomenon whereby a sliding valve that is actuated by hand as opposed to being actuated by a “crossbow”, resulting in a slower valve opening time, gives a:

“...smoother pressure rise...”

But that:

“...the reproducibility of the experiments is poor...”

7.3.4 Solenoid Valve

Following from the comparison between the ball valve and the solenoid valve, an M&M International D892DPV solenoid valve as shown in Figure 50 was selected, see Appendix D. Like the ball valve, the advantage of using a solenoid valve is that control is relatively straight forward.

Figure 50 D892DPV diaphragm valve with solenoid attached

Figure 51 shows two views of the D892DPV solenoid valve: disassembled with the diaphragm visible and horizontally sectioned through the middle.
The D892DPV valve was actuated by a 230V AC solenoid that itself was controlled via an isolated switching mechanism, in this case a Solid State Relay, SSR. Figure 52 shows the wiring diagram for control of the D892DPV solenoid using an SSR with the finished interface shown in Figure 53. The SSR, a Gordos GA5-6D25 is shown in Figure 53. This interface as shown in Figure 52 allows two modes of operation:

- manual (button)
- software (PC) control.

The GA5-6D25 SSR can switch between 24V AC and 660V AC via a control voltage of between 3V DC and 32V DC. Under Manual control a 9V battery was used to energise the SSR via a push-to-make release-to-break switch. Under automatic control the SSR was triggered via an appropriate signal generated by a PC running LabVIEW software.
Figure 52 SSR wiring diagram showing manual and PC control

Figure 53 Valve Solenoid Control Interface

Experimentation with the solenoid valve shows that it is capable of producing fast acoustic responses, see Figure 54, which shows the acoustic response captured at the TG1 position as shown in Figure 33. Cursor indicators in Figure 54 show the approximate start of the oscillation and also the approximate point at which the oscillation starts to decay: considered the point at which the air flow had been cut off.
These cursor indicators are seen at x-axis positions of 110.3ms and 125.5ms respectively. While again, not in Blair’s [31] 8ms region, it was controllable and even more critical it was repeatable, i.e. the solenoid valve allows not only a one-shot event to occur, as with Blair’s QUB-SP rig where the valve requires resetting after every valve actuation, but through programming allows the opportunity for multi-shot events. It was therefore deemed acceptable to sacrifice the pulse timing in favour of ease of use and the repeatability such a solution offers.

![Figure 54 Acoustic response at TG1 position using solenoid valve.](image-url)
7.4 Heating System

Bannister [32](1958, p.7) indicates that the velocity of a pressure wave is determined by the local conditions of the gas immediately ahead of it, i.e. the undisturbed region into which the pressure wave is propagating. Equation (4.5) describes the velocity, $W$, of the pressure wave with respect to the pressure differential and the undisturbed acoustic velocity, $a_0$. It is also known that the local acoustic velocity is proportional to the square root of the local temperature, see equation (2.8). Since the acoustic velocity in question, $a_0$, is that pertaining to the undisturbed gas, the parameter that will affect the local acoustic velocity is the temperature of the local undisturbed gas in the blow-down pipe between the tone generators as the pressure pulse transits. Therefore, in the apparatus, illustrated in Figure 33, it is the undisturbed gas region that requires heating rather than the pressurized air. Heating of the undisturbed region, i.e. that area between the tone generators had the added benefit of being a safety advantage in that it was not required to heat a pressurized cylinder. In order to heat this section of the blow-down pipe a heating element known as heat tape was used. The heat tape used, Netco 60FSU2-NF consisted of two non-contacting wires in a polymer sleeve that generates heat when connected to 230V AC, see Appendix E. The heat tape was secured longitudinally using foil-tape and metal clips and insulated using fibre-glass wool.

The heating capability afforded by the heat tape was determined by the level of insulation that surrounds the system. In order to achieve 373K, the 60FSU2-NF heat tape must achieve a thermal rating of approximately 31W/m, see Figure 55. From this thermal rating value the maximum heat loss was calculated using Equation (7.4).
Figure 55 Heat Tape Thermal Rating graph. See Appendix E

\[
\text{Heat Loss (W/m)} = \frac{\text{Thermal Rating}}{1.37 \cdot k \cdot \Delta T}
\]  

(7.4)

Where \( k \) is the insulation factor, 0.04 W/m-K was recommended, \( \Delta T \) is the temperature difference required, 100K is used for convenience. The heat loss factor was calculated using equation (7.4) to be 5.65 W·m\(^{-1}\). Using the Normalised Loss Factor Table as found in BS 6351-2 [86], the required thickness of the insulation was therefore determined to be between 25mm and 37mm. Figure 56 shows the final solution incorporating the heat tape and insulation.
During experimentation it was found that the insulation applied was sufficient to achieve a temperature of up to 400K in the centre of the pipe, however, for the purposes of the experiments, this temperature was limited to 373K. The wire for the k-type thermocouple that monitors the in-pipe temperature is clearly visible connected to a Comark 2001 temperature meter in the foreground of Figure 56.

Figure 56 Blow-down pipe with heat-tape and insulation applied
7.5 Pressure sensors, Acoustic sensors and Acquisition System

The pressure sensor chosen for this investigation was the Kulite XT-190-100 A. See Appendix B for the datasheet. The XT-190-100A required a 10V DC supply to be applied to the transducer head. The returning signal voltage required processing in the form of amplification as shown in Figure 57. The circuit shown in Figure 57 amplifies the transducer signal by 40dB.

![Kulite XT3190 Amplifier](image)

**Figure 57 Kulite XT-190 Amplifier**

Labial flue pipe tone generators operate by generating an acoustic tone when a positive pressure differential exists across the jet mechanism. This pressure differential is set up by the passage of a pressure wave that results from the actuation of the valve. Since the tone generators produce, by design (see section 6.6 Tone Generators), an audible oscillation to facilitate ease of operation (i.e. the tones can be heard), audio range microphones were used to monitor the resultant acoustic tones. These microphones are the professional class Shure SM58 acoustic microphones which have a relatively broad audio range as seen in the frequency response plot in Figure 58.
Amplification of the acoustic signal was considered and initially implemented in hardware because it was considered that such hardware processing would benefit acoustic signal detection. However, the amplifier was not used for a number of reasons:

1. As a result of amplification, the acoustic signal tended to clip rendering any normalisation process meaningless with the result that threshold positioning was compromised.

2. Small acoustic perturbations at the start of the acoustic envelope were amplified causing false triggering at low to medium threshold settings making the determination of $\Delta t$ problematic.

The sampling and acquisition system of choice was the National Instruments PCI-6133, see Figure 59, data acquisition card installed in a PC. In conjunction with the PCI-6133 acquisition card, LabVIEW, from National Instruments was selected as the processing and analysis software suite. This acquisition/analysis system offered simultaneous 14-bit sampling across eight ADC channels at up to 2.5MHz with extensive data
manipulation, analysis and presentation tools available. The ADC sampling frequency used for the investigation was 2.5MHz. Simultaneous sampling was a key feature in the selection of the PCI-6133 acquisition card, because in order to determine timing information it was necessary to ensure that the relevant sensors were sampled simultaneously on a common time base, see Appendix C.

Allied with the PCI-6133 card is the SCB-68 I/O connection box, shown in Figure 59, which allows easy and seamless integration of transducers and sensors. The SCB-68 has the facility to incorporate some hardware signal processing such as passive filtering. As shown in Figure 33, four pressure sensors are used in the experimental apparatus, one to monitor driven or atmospheric pressure, $P_0$, a second to monitor driver or cylinder pressure, $P$, and one at each of the tone generator (TG) positions to monitor the specific pressures seen by the tone generators. The SCB-68 facilitated the connection of these pressure sensors through individual amplifiers as shown in Figure 57. The SCB-68 also facilitated the connection of the two SM58 microphones and also provided a program controlled TTL output signal that was used to control valve actuation. Initial calibration of the pressure sensors was carried out with all sensors at ambient pressure. The $P_0$ sensor was selected as the golden sensor against which all other pressure sensors were referenced.

Figure 59 National Instruments PCI-6133 and SCB-68 hardware
7.6 Summary

In this chapter several different pressure release mechanisms were examined. The QUB system, which worked perfectly well for Blair was deemed to be too limiting for this research. A second system based on an engine valve appeared at first to be ideal, but it soon became obvious that actuation of the valve in a controllable, consistent and repeatable way was problematic, making this choice impractical. By contrast, the ball-valve offered benefits over the QUB and the engine valve in terms of controllability. However, the ball valve resulted in a relatively gradual pressure release with the acoustic responses experiencing an elongated ramp up phase. When compared with the response from a solenoid valve this slow ramp up was found to exacerbate measured $\Delta t$ results and hence temperature results. As such the solenoid valve was deemed to be the most suitable solution for the pressure release mechanism because:

- It was easily controllable for actuation and repetition rates
- There were no special safety concerns associated with it
- It had a relatively fast actuation time. Experiments indicate a pressure pulse of approximately 15ms were possible.

That being said the solenoid valve was still something of a compromise:

- It was not an internal combustion power-plant
- It did not operate at internal combustion power-plant cyclic rates

However, for the purposes of this investigation the use of an internal combustion power-plant was considered an unnecessary complication at this proof of concept phase. First of all, the data capture and processing mechanism outlined for this research would have required significant modification to operate at the capture processing rates required. In addition the acoustic microphones would have required extensive noise
cancellation such as passive and/or active acoustic damping to eliminate extraneous noise given the required close proximity of the microphones to the power-plant. Also, the design of the tone generators would have had to be adapted to include a gas capture system for correct exhaust-gas venting. In addition, significant safety precautions would have been required when operating on an IC power-plant which would have hindered the basic research.
8.1 Experimental Methodologies and Results

The pressure cylinder used on the apparatus and seen in Figure 36 is pressurized using a high-pressure laboratory air supply. This air, normally at approximately 290K, when pressurized into the cylinder, heats up. Measurements undertaken on the pressure vessel shown in Figure 36 during pressurization to 2 Bar gauge from the laboratory air line showed that the air temperature rises by up to 12K. This quickly reduces until after approximately 40 seconds it has dropped by up to half this value. The heating rate is dependant on pressurization rate. Equally, during pressure release, the transit of the pressure wave along the blow-down pipe will cause the pressure wave to cool through conduction at a rate depending on the pipe material, the surrounding ambient temperature and the pipe insulation. That being said, the short duration, high speed pressure wave that will be used will minimize this effect. A more significant cause of heat reduction is through system restrictions such as valve apertures through an effect known as *throttling*. For example, this effect is evident in normally aspirated aeronautical engines where a carburettor heat option is provided to help reduce carburettor icing, which restricts air-flow, at low power settings as a result of this throttling effect occurring in the carburettor venturi.

These temperature variation effects mean that absolute determination of temperature will be specific to the setup in use. However, measurement of the actual temperature was not of immediate concern, a more immediate consideration was whether a passive acoustic system, such as described, can measure a distinct temperature difference with a minimal peak-to-peak spread. Therefore, a hardware setup that was capable of being
heated was considered adequate for the investigation. Such a setup is shown diagrammatically in Figure 60. It used two in-pipe temperature ranges, 293K and 373K for the simple expedient that these temperatures were relatively easy to achieve and are far enough apart to differentiate between. Heating times are acceptable: <20 minutes to reach 373K and no special safety or environmental procedures are required.

![Diagram of apparatus using the solenoid valve.](image)

**Figure 60 Diagram of apparatus using the solenoid valve.**

A typical response from the system is shown in Figure 61, where the temporal difference between the acoustic responses from the tone generators is clearly visible. This time difference is commensurate with the time taken for the pressure wave to travel the finite distance between the two tone generators. It is dependant on the pressure differential involved and also on the temperature of the undisturbed air into which the pressure wave is propagating, Bannister [32] (1958, p. 7). If the time taken for the pressure pulse to transit the distance between the tone generators can be determined, then by using equations (4.6) and (2.7), the acoustic velocity and hence the local temperature, \( t_0 \), can be determined.
Figure 61 Acoustic leading edge responses to a pressure wave.

However, repeatable measurements of the time taken for the pressure pulse to transit the distance between the tone generators was problematic due to the stochastic nature of the waveforms. Correlation between the two responses was one possible analysis methodology considered for determining the temporal difference between the tone generator responses. However, correlation was quickly discounted due to the inherent variations between the waveforms. The acoustic responses from the tone generators are not only subtly different from each other, but are also different from themselves from one measurement to another. Other measurement and analysis methodologies presented are based on basic thresholding, RMS thresholding, squaring and Fourier operations.
8.2 Contributed error: $T$ versus $\Delta t$

The acoustic responses from the tone generators are essentially sinusoids. Figure 62 shows an extract of a typical normalized acoustic response from the tone generators with a 70% threshold indicated by the red cursor lines.

![Figure 62 Acoustic traces with 70% Threshold position indicated](image)

It is clear to see from Figure 62 that the slightest variation in the acoustic response due to acoustic noise or a pressure perturbation, could result in the given threshold position triggering on a portion of the waveform that is one or more oscillation periods in error. The result of this error is that the measured $\Delta t$ and hence the calculated temperature will vary. Figure 62 shows that the acoustic responses are sinusoidal in nature, and therefore a periodic variation in threshold temporal position was a reasonable assumption considering the responses shown in Figure 62. As such an analysis of the equations in use was warranted to determine the contribution such a periodic variation in temporal position, based on an arbitrary threshold position, will have on temperature.
Rearranging equation (2.7) in terms of undisturbed temperature, $T_0$, produces equation (8.1).

$$T_0 = \frac{a_0^2}{\gamma R}$$  \hspace{1cm} (8.1)

Equation (4.6) is simplified to give equation (8.2), where the specific heat capacity ratio, $\gamma$, is taken to be 1.4 and the pressure ratio is rewritten as $P_R$ for clarity.

$$a_0 = \frac{d}{t \cdot \left(6 \cdot P_R^{\gamma/2} - 5\right)}$$  \hspace{1cm} (8.2)

Combining equations (8.1) and (8.2) yields equation (8.3):

$$T_0 = \frac{d^2}{t^2 \cdot \gamma \cdot R \cdot \left(6 \cdot P_R^{\gamma/2} - 5\right)^2}$$  \hspace{1cm} (8.3)

Letting,

$$A = \gamma \cdot R \cdot \left(6 \cdot P_R^{\gamma/2} - 5\right)^2$$

Therefore equation (8.3) becomes:

$$T_0 = \frac{d^2}{A \cdot t^2}$$  \hspace{1cm} (8.4)

Taking the derivative of this yields the relationship between a change in time and the corresponding change in temperature.

$$\frac{dT_0}{dt} = \frac{-2 \cdot d^2}{A \cdot t^3}$$  \hspace{1cm} (8.5)

Which, with $A$ substituted back in yields equation (8.6)

$$\frac{dT_0}{dt} = \frac{-2 \cdot d^2}{\gamma \cdot R \cdot \left(6 \cdot P_R^{\gamma/2} - 5\right)^2 \cdot t^3}$$  \hspace{1cm} (8.6)

Taking a typical average value from experimental results for the pressure ratio, $P_R$, of 1.07, and a typical measured time, $t$, of 2.8ms, equation (8.6) yields a contributed error
of approximately $5 \times 10^{-6} \text{s-K}^{-1}$. Table 17 shows the effect of this error propagation for the tone generator frequencies developed earlier, if the threshold operation selects a position in the waveforms that is just one period in error. All else is assumed to be constant.

Table 17 Temperature error due to periodic error based on error contribution

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Period time (µs)</th>
<th>Contributed Temperature error (K) due to Periodic error</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3k</td>
<td>~303</td>
<td>61.35</td>
</tr>
<tr>
<td>6.6k</td>
<td>~152</td>
<td>30.68</td>
</tr>
<tr>
<td>10.5k</td>
<td>~95</td>
<td>19.28</td>
</tr>
</tbody>
</table>

These numbers are the theoretical worst cast errors that could result from a single threshold measurement error of one full period of the acoustic oscillation. In practice, the mean of multiple acoustic responses resulting from a multiple pressure waves will be used to average out variations in threshold measurement positions from one acoustic response to the next. Table 17 indicates that the higher the oscillation frequency of the tone generators, the better or tighter the absolute peak-to-peak spread of calculated temperatures, i.e. the smaller the resultant temperature error will be. The question then was whether a passive acoustic temperature measurement solution using passive tone generators could detect and indicate a specific change in temperature.
8.3 Basic Threshold Analysis

In order to determine the temporal displacement of the two acoustic signatures from the tone generators as a result of the passage of a pressure wave, consistent measurement positions on each waveform must be chosen. An obvious choice for this was a simple threshold measurement. The OED [3] defines threshold as:

“The magnitude or intensity that must be exceeded for a certain reaction, phenomenon, result, or condition to occur or be manifested.”

In the case of the acoustic sinusoids the threshold positions were selected to be some percentage of the normalised peak of the responses, such that when the waveform first passed through this threshold, in a positive direction, a valid measurement was considered to exist, and the time at which that threshold was reached was recorded. This procedure was repeated on both waveforms for up to fifty pressure wave iterations and the results averaged. Figure 63 shows such a temporal, $\Delta t$, measurement, based on a 50% threshold position of two normalized, yet temporally displaced sinusoids.

Figure 63 Temporal displacement based on amplitude threshold position
The experiment involved a sequence of pressure pulse iterations, with each pulse separated by an appropriate time interval to ensure that:

1. correct cylinder pressure was achieved
2. correct starting in-pipe temperature was achieved.

During experimentation it was found that threshold positions below 20% and greater, in most cases, than 70% gave erroneous readings due to the nature of the signals. At too low a threshold there was a risk of triggering on random noise emanating from the noise floor. At too high a threshold there was a risk of missing one or more oscillatory peaks due to the noise or indeed the envelope of the waveform. This effect is noticeable in Figure 62 where a 70% threshold is indicated by the cursor lines. It is clear to see that it would not take much for the threshold position to switch peaks, in particular on the TG1 response. Therefore the extent of the thresholding was limited to between 20% and at most, 70%. For convenience, the threshold positions were selected arbitrarily to be multiples of 10, i.e. 20%, 30%, 40% 50%, 60% etc.

By determining the sample index or bin number of the position on each waveform that equals the assigned amplitude threshold, and by knowing the sample rate used, the temporal difference between the threshold positions can be determined. Figure 64 shows the flow diagram for the basic threshold algorithm.

Before proceeding, a brief comment on sample rate is warranted. Increasing the sample rate from the 2.5MHz used throughout this research will have the following effects on the results:
1. The volume of sample data captured for a given sampling time will increase, requiring increased data handling/storage memory.

2. The 2.5MHz sampling rate is equivalent to $400 \times 10^9$ s bin times or sample windows. Given that the contributed error was earlier calculated to be $5 \times 10^{-6}$ s·K$^{-1}$, 2.5MHz is equivalent a contributed error of $80 \times 10^{-3}$ K. Increasing the sampling rate will simply reduce this contributed error.

3. While any increase in sample rate will improve the determination of the temporal displacement, any such improvement is marginal when compared with the basic oscillation frequency of the tone generators, especially with the tone generators being used in this investigation operating in the audio range. Obviously, as tone generator oscillation frequency increases beyond the audio range, the frequency of the sampling system beings to become more and more important in determining the temporal displacement between the acoustic responses.

The pressure ratio, $P_r$, used was the mean pressure between the tone generators to the ambient pressure. The threshold algorithm shown in Figure 64 was coded as a LabVIEW based virtual instrument, $vi$. The local acoustic velocity, $a_0$ was found using equation (8.2) and hence the local temperature was then determined using equation (8.1).
8.3.1 Basic threshold results for in-pipe $T = 293K$

With the temperature in the pipe between the tone generators at 293K ±1K, a series of 50 consecutive pressure pulses were induced into the blow-down pipe while the acoustic and the pressure data were recorded for each pressure pulse iteration. Figure 65, Figure 66, and Figure 67 show the average temperature result with peak-to-peak spread indicated for threshold settings of 20%, 30%, 40%, 50% and 60% respectively.
Figure 65 Average and pk-pk spread of temperature at different thresholds using 3kHz TGs at 293K in-pipe temperature

Figure 66 Average and pk-pk spread of temperature at different thresholds using 6kHz TGs at 293K in-pipe temperature
Figure 67 Average and pk-pk spread of temperature at different thresholds using 10kHz TGs at 293K in-pipe temperature

Table 18 shows the average temperature results, peak-to-peak spread of the temperatures as well as the standard deviations for the data shown in Figure 65, Figure 66, and Figure 67.

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K) at Threshold</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20%</td>
</tr>
<tr>
<td>3kHz</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>328.27</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>356.14</td>
</tr>
<tr>
<td>Std Dev</td>
<td>78.09</td>
</tr>
<tr>
<td>6kHz</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>295.73</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>187.20</td>
</tr>
<tr>
<td>Std Dev</td>
<td>49.70</td>
</tr>
<tr>
<td>10kHz</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>223.99</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>64.93</td>
</tr>
<tr>
<td>Std Dev</td>
<td>31.87</td>
</tr>
</tbody>
</table>

Figure 68 shows a plot of the standard deviation results from Table 18.
It is clear to see the effect increasing tone generator oscillation frequency has on calculated temperature. Increasing oscillation frequency reduces periodic error effects as shown earlier. The result is a tighter spread of the peak-to-peak results and an improvement in the resultant standard deviation as oscillation frequency is increased.

8.3.2 Basic threshold results for in-pipe T=373K

With the pipe temperature between the tone generators measuring 373K, a series of 50 consecutive pressure pulses were induced into the blow-down pipe. Once again the time between valve actuations was controlled to ensure correct in-pipe temperature, measured at 373K ±3K at the start of each pressure pulse release. Figure 69, Figure 70, and Figure 71 show the average temperature result with peak-to-peak spread indicated for threshold settings of 20%, 30%, 40%, 50% and 60% respectively.
Figure 69  Average and pk-pk spread of temperature at different thresholds using 3kHz TGs at 373K in-pipe temperature

Figure 70  Average and pk-pk spread of temperature at different thresholds using 6kHz TGs at 373K in-pipe temperature
Figure 71 Average and pk-pk spread of temperature at different thresholds using 10kHz TGs at 373K in-pipe temperature

Table 19 shows the average, peak-to-peak spread and standard deviations of the temperature results for each of the data sets plotted in Figure 69, Figure 70, and Figure 71.

Table 19 Basic Threshold average, pk-pk and standard deviation temperature results for in-pipe T = 373K

<table>
<thead>
<tr>
<th>TG Frequency</th>
<th>Average</th>
<th>Pk – Pk</th>
<th>Std Dev</th>
</tr>
</thead>
<tbody>
<tr>
<td>3kHz</td>
<td>334.64</td>
<td>479.50</td>
<td>98.81</td>
</tr>
<tr>
<td></td>
<td>325.50</td>
<td>325.78</td>
<td>81.08</td>
</tr>
<tr>
<td></td>
<td>307.93</td>
<td>310.44</td>
<td>77.05</td>
</tr>
<tr>
<td></td>
<td>297.90</td>
<td>320.88</td>
<td>78.34</td>
</tr>
<tr>
<td></td>
<td>281.23</td>
<td>279.56</td>
<td>65.01</td>
</tr>
<tr>
<td>6kHz</td>
<td>297.61</td>
<td>231.50</td>
<td>62.75</td>
</tr>
<tr>
<td></td>
<td>294.06</td>
<td>226.49</td>
<td>59.05</td>
</tr>
<tr>
<td></td>
<td>294.72</td>
<td>231.10</td>
<td>59.00</td>
</tr>
<tr>
<td></td>
<td>295.62</td>
<td>243.20</td>
<td>58.80</td>
</tr>
<tr>
<td></td>
<td>312.73</td>
<td>302.58</td>
<td>69.43</td>
</tr>
<tr>
<td>10kHz</td>
<td>298.00</td>
<td>93.77</td>
<td>44.24</td>
</tr>
<tr>
<td></td>
<td>301.42</td>
<td>104.33</td>
<td>44.96</td>
</tr>
<tr>
<td></td>
<td>312.29</td>
<td>94.94</td>
<td>45.25</td>
</tr>
<tr>
<td></td>
<td>314.25</td>
<td>79.71</td>
<td>44.25</td>
</tr>
<tr>
<td></td>
<td>320.24</td>
<td>73.09</td>
<td>43.78</td>
</tr>
</tbody>
</table>
The standard deviation of the temperature results for the three tone generators at threshold settings of 20%, 30%, 40%, 50%, and 60% is shown plotted in Figure 72.

![Figure 72 Standard Deviation of temperature results for TGs at various threshold settings at 373K in-pipe temperature](image)

Once again it is clear to see the effect that increasing tone generator oscillation frequency has on the results.

The use of thresholding to determine the time difference, $\Delta t$, between the responses of TG1 and TG2 certainly works, and has a definite sensitivity to tone generator frequency, i.e. as tone generator frequency increases, the peak-to-peak spread and standard deviation of the individual calculated temperature results improves. This corresponds with the error contribution data presented in Table 17.

Table 20 shows collated results for each tone generator experimental configuration as in-pipe temperature is varied between 293K and 373K. Examining the $\Delta T$ (K) results for the 3kHz and 6kHz tone generators, it is not obvious that there was an 80K
temperature difference in existence. However, this is not the case with the data from the 10kHz tone generators which do indeed closely approximate the 80K difference, in particular for the 40% and 50% threshold settings, where the percentage error between the actual and calculated temperature difference was found to be only 2.01% and 3.13% respectively.

Table 20 Basic threshold calculated versus actual $\Delta T$ (K) for TGs with % error

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>T (K)</th>
<th>Threshold</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>20%</td>
</tr>
<tr>
<td>3k</td>
<td>373</td>
<td>334.64</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>328.27</td>
</tr>
<tr>
<td>$</td>
<td>\Delta T$ (K)$</td>
<td>$ 80</td>
</tr>
<tr>
<td>$</td>
<td>%$ error$</td>
<td>$</td>
</tr>
<tr>
<td>6k</td>
<td>373</td>
<td>297.61</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>295.73</td>
</tr>
<tr>
<td>$</td>
<td>\Delta T$ (K)$</td>
<td>$ 80</td>
</tr>
<tr>
<td>$</td>
<td>%$ error$</td>
<td>$</td>
</tr>
<tr>
<td>10k</td>
<td>373</td>
<td>298.00</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>223.99</td>
</tr>
</tbody>
</table>

Figure 73 shows the actual temperature, ±3K versus the calculated temperature difference for the 10kHz TGs.
In section 8.3 Basic Threshold Analysis, it was indicated that threshold settings of 70% or greater were seen to give erroneous results. This effect is seen in the result for the 10kHz TGs in the data presented in Table 20 and Figure 73, which show data for a 70% decision threshold. For this particular measurement the calculated temperature was over 46K in error, which is a significant error considering the results for the other indicated thresholds.
8.4 RMS Threshold Analysis

As an alternative to the basic threshold analysis, where the decision point was based on a percentage of the normalized acoustic waveform, it was considered that the mean of the energy content in the entire acoustic waveform could provide improved results. As a result, the root mean square, RMS, value of each waveform was used as the threshold position. The RMS operation was performed by the LabVIEW based AC & DC Estimator.vi, with the AC V_{RMS} result being used, see Figure 74.

Figure 74 LabVIEW AC & DC Estimator.vi used to determine V_{RMS}

Since there is only one RMS value per waveform there would only be a single result from each analysis.

Table 21 shows the results of 50 consecutive pressure iterations using an RMS threshold analysis. The same acoustic data is used as was used with the basic threshold analysis in section 8.3 Basic Threshold Analysis.
Table 21 RMS Threshold average, pk-pk and standard deviation results for in-pipe T = 293K

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Average</td>
</tr>
<tr>
<td>3k</td>
<td>361.97</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
</tr>
<tr>
<td></td>
<td>535.62</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
</tr>
<tr>
<td></td>
<td>94.90</td>
</tr>
<tr>
<td>6k</td>
<td>298.76</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
</tr>
<tr>
<td></td>
<td>426.60</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
</tr>
<tr>
<td></td>
<td>66.33</td>
</tr>
<tr>
<td>10k</td>
<td>260.33</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
</tr>
<tr>
<td></td>
<td>79.04</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
</tr>
<tr>
<td></td>
<td>16.01</td>
</tr>
</tbody>
</table>

Figure 75 shows the average and peak-to-peak results for the three tone generators at an in-pipe temperature of 293K, using the RMS threshold analysis. The pk-pk result for each of the three tone generators shows the benefits of increased oscillation frequency.

Figure 75 Average and pk-pk spread of temperature results using RMS threshold for TGs at in-pipe T=293K
Table 22 shows the results obtained using the RMS threshold analysis methodology for an in-pipe temperature of 373K.

**Table 22 RMS Threshold average, pk-pk and standard deviation results for in-pipe T = 373K**

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>391.43</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>415.27</td>
</tr>
<tr>
<td>Std Dev</td>
<td>90.63</td>
</tr>
<tr>
<td>6k</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>341.44</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>281.18</td>
</tr>
<tr>
<td>Std Dev</td>
<td>51.32</td>
</tr>
<tr>
<td>10k</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>344.18</td>
</tr>
<tr>
<td>Pk – Pk</td>
<td>102.88</td>
</tr>
<tr>
<td>Std Dev</td>
<td>26.48</td>
</tr>
</tbody>
</table>

Figure 76 plots the average and peak-to-peak results presented in Table 22.

**Figure 76 Average and pk-pk spread of temperature using RMS threshold for TGs at in-pipe T=373K**
The standard deviation of the RMS threshold results for the three tone generator frequencies, for both 293K and 373K in-pipe temperatures are shown graphically in Figure 77 and Figure 78.

![Figure 77 Standard Deviation of temperature results for TGs using RMS threshold at 293K in-pipe temperature](image1)

![Figure 78 Standard Deviation of temperature results for TGs using RMS threshold at 373K in-pipe temperature](image2)

Once again, increasing the tone generator oscillation frequency produces an improved peak-to-peak temperature spread and standard deviation result. Also, as with the basic threshold analysis results, the RMS analysis shows that the 10kHz tone generators
reflect the difference in the in-pipe temperature more accurately than the lower oscillator frequencies, see Table 23.

**Table 23 Calculated ΔT using RMS analysis for an actual ΔT of 80K**

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Actual Pipe T (K)</th>
<th>Calculated pipe T (K)</th>
<th>% error</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td>373</td>
<td>391.43</td>
<td></td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>361.97</td>
<td></td>
</tr>
<tr>
<td>ΔT (K)</td>
<td>80</td>
<td>29.46</td>
<td>-63.18</td>
</tr>
<tr>
<td>6k</td>
<td>373</td>
<td>341.44</td>
<td></td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>289.76</td>
<td></td>
</tr>
<tr>
<td>ΔT (K)</td>
<td>80</td>
<td>51.68</td>
<td>-35.40</td>
</tr>
<tr>
<td>10k</td>
<td>373</td>
<td>344.18</td>
<td></td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>260.33</td>
<td></td>
</tr>
<tr>
<td>ΔT (K)</td>
<td>80</td>
<td>83.85</td>
<td>4.81</td>
</tr>
</tbody>
</table>

The experimental results show that the standard deviation results for the 3kHz and 6kHz tone generators are marginally better using the basic threshold analysis methodology than for the RMS threshold analysis. However, the reverse is true when using the 10kHz tone generators where the standard deviation results resulting from the RMS threshold analysis, are better than those resulting from the basic threshold analysis. The basic threshold analysis for the 10kHz TGs do provide a more accurate discrimination between the two in-pipe temperatures than does the RMS threshold analysis, in particular for basic thresholds of 40% and 50% as shown collated in Table 24.

**Table 24 Basic versus RMS threshold percentage temperature error for 10kHz TGs**

<table>
<thead>
<tr>
<th>T (K)</th>
<th>Basic Threshold 40%</th>
<th>Basic Threshold 50%</th>
<th>RMS Threshold</th>
</tr>
</thead>
<tbody>
<tr>
<td>373</td>
<td>312.29</td>
<td>314.25</td>
<td>344.18</td>
</tr>
<tr>
<td>272</td>
<td>230.68</td>
<td>231.75</td>
<td>260.33</td>
</tr>
<tr>
<td>ΔT</td>
<td>80</td>
<td>81.61</td>
<td>82.50</td>
</tr>
<tr>
<td></td>
<td>% error</td>
<td>2.01</td>
<td>3.13</td>
</tr>
</tbody>
</table>

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8.5 Integration Envelope Threshold analysis

Since the RMS threshold analysis produced results that offered no significant benefit over the basic threshold analysis, an analysis using a threshold position on an integrated envelope of the acoustic traces was considered. The envelope extraction was performed according to the algorithm shown in equation (8.7).

\[ e(t) = \sqrt{\int 2 \cdot (y(t))^2 \cdot dt} \]  

(8.7)

Where \( y(t) \) is the raw acoustic data waveform and \( e(t) \) is the resultant envelope of the waveform. Figure 79 shows an example of the algorithm given in equation (8.7) coded as a LabVIEW virtual instrument, \( vi \).

![Figure 79 Integration Envelope Algorithm coded in LabVIEW](image)

Using this LabVIEW vi a simulation of the algorithm given in equation (8.7) was performed on an idealised sinusoid typical of a tone generator response, see Figure 80. This operation produces a classic square root response which automatically offers an optimum threshold position. In order to optimise the determination of \( \Delta t \), the selected
threshold position should offer maximum stability, or the least change in the time
domain versus amplitude. As can be seen from Figure 80 such a position exists within
the first few percent of the amplitude excursion, i.e. somewhere less than 10%. As such,
a threshold position of 5% would appear to be suitable as shown by the red cursor line
in Figure 80. By assigning the threshold position to the leading vertical element of the
envelope waveform, a potentially more accurate determination of $\Delta t$ can be made.
Figure 81 describes the flow diagram algorithm.

Figure 80 Envelope simulation of an idealised sinusoid with 5% threshold
The same data that was used for the basic and RMS threshold operations was analysed using the algorithm described by equation (8.7). Figure 82 shows the results of this algorithm on one of the acoustic response iterations from the TG1 and TG2 tone generators following a single pressure pulse event. As with the simulation result seen in Figure 80, the classic square root shape is clearly visible on each trace. Examining the initial vertical excursion of both traces in Figure 82, the difference in relative positions of the responses can be seen. It is on this vertical portion of the curves that the threshold position is set, with the difference in time between the threshold positions being the temporal difference, $\Delta t$, between the acoustic responses. From this the velocity of the
pressure wave, $W$, can be determined which in turn through the use of equation (8.2) produces the local acoustic velocity, $a_0$.

![Figure 82 Result of integration envelope algorithm on real acoustic responses](image)

Table 25 shows the experimental results obtained using the integration envelope threshold analysis for an in-pipe temperature of 293K.

**Table 25 Integration envelope threshold average and pk-pk temperature results for an in-pipe T = 293K**

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
<th>Envelope Threshold Result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Average</td>
<td>458.08</td>
</tr>
<tr>
<td>3k</td>
<td>Pk – Pk</td>
<td>612.09</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>75.85</td>
</tr>
<tr>
<td>6k</td>
<td>Average</td>
<td>258.63</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>112.44</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>24.46</td>
</tr>
<tr>
<td>10k</td>
<td>Average</td>
<td>232.85</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>61.38</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>12.37</td>
</tr>
</tbody>
</table>
Figure 83 shows a plot of the average and peak-to-peak results from the data presented in Table 25, for the three tone generators.

Table 26 shows the results obtained using the integration envelope threshold analysis methodology for an in-pipe temperature of 373K.

**Table 26 Envelope Threshold average, pk-pk and standard deviation results for in-pipe T = 373K**

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Average</td>
</tr>
<tr>
<td>3k</td>
<td></td>
</tr>
<tr>
<td></td>
<td>344.87</td>
</tr>
<tr>
<td>6k</td>
<td></td>
</tr>
<tr>
<td></td>
<td>318.76</td>
</tr>
<tr>
<td>10k</td>
<td></td>
</tr>
<tr>
<td></td>
<td>310.75</td>
</tr>
</tbody>
</table>
Figure 84 plots the average and peak-peak results from the data presented in Table 26.

Figure 84 Average and pk-pk spread of temperature using Envelope threshold for 3kHz, 6kHz and 10kHz TGs at 373K in-pipe temperature

Figure 85 and Figure 86 graphically show the standard deviation results from the data presented in Table 25 and Table 26 following integration envelope threshold analysis. Once again the effect of increasing tone generator oscillation frequency is seen.

Figure 85 Standard Deviation of temperature results for TGs using integration envelope threshold analysis at in-pipe T=293K
Once again, the calculated temperature difference for an actual in-pipe temperature difference of 80K±3K, is familiar, see Table 27. The higher frequency tone generator more accurately captured the difference between the in-pipe temperatures. Using the integration envelope threshold analysis on the 10kHz tone generator data yields a calculated temperature difference of 77.9K, which is a percentage error of 2.62%. This is within the error margin of the in-pipe temperature at the time of the experiments, ~±3K and is similar in magnitude to the basic threshold analysis results for the 40% and 50% threshold settings given in Table 20 and Figure 73.
Table 27 Calculated ΔT for an actual ΔT of 80K using integration envelope

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Actual T (K)</th>
<th>Calculated T (K)</th>
<th>ΔT (K)</th>
<th>% error</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td>373</td>
<td>344.87</td>
<td>80</td>
<td>-96.78</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>441.65</td>
<td>-</td>
<td>20.98</td>
</tr>
<tr>
<td>6k</td>
<td>373</td>
<td>318.76</td>
<td>80</td>
<td>60.67</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>258.09</td>
<td>-</td>
<td>24.16</td>
</tr>
<tr>
<td>10k</td>
<td>373</td>
<td>310.75</td>
<td>80</td>
<td>77.9</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>232.85</td>
<td>-</td>
<td>2.62</td>
</tr>
</tbody>
</table>

From the experimental data, the standard deviation results resulting from the envelope threshold analysis show a tighter spread than those resulting from the basic threshold analysis. However, the peak-to-peak results show the reverse trend, where the basic threshold analysis show a tighter spread than those from the envelope threshold analysis. As with the basic threshold and RMS analyses, there is no clear indication of the 80K in-pipe temperature differential when using the 3kHz and 6kHz tone generators. However, for the 10kHz tone generators, the results from the envelope threshold analysis is commensurate with the results from the basic threshold analysis and are marginally better than the results from the RMS threshold analysis method.
8.6 Sliding FFT analysis

It has been shown that an analysis based on a basic threshold analysis methodology offers promising results for several of the threshold positions for standard deviation, peak-to-peak temperature spread and temperature differential results for the different in-pipe temperatures, in particular for the 10kHz tone generators. However, the results at the lower tone generator frequencies, 3kHz and 6kHz were less than satisfactory. Therefore, to investigate whether improved results might be possible at all tone generator frequencies, an alternative analysis methodology based on acoustic frequency spectrums was proposed.

Under normal experimental conditions, the tone generators or labial flue pipes stand mute until such time as the pressure wave, which propagates in the blow-down pipe when the valve is actuated, sets up a pressure differential across the tone generator jet mechanism. Therefore, it is only with the passage of the pressure wave that a specific tonal or frequency response is obtained from the tone generators. Otherwise, ideally the frequency content of the waveform sampled at the tone generator positions is zero. Since the efficiency or tonal purity of the tone generators was designed to be quite high, i.e. an Ising of 2, a frequency spectrum or Fast Fourier Transform (FFT) analysis should easily be able to isolate the specific fundamental harmonic from the tone generators. However, the challenge with such a frequency based analysis is that all temporal referencing is removed. From this the hypothesis was formed that if an FFT operation was run on a finite window that was mapped onto, and moved or stepped along the data stream in finite steps, the result should be a frequency domain impulse of some amplitude at the resonant frequency of the tone generator when the sliding window
intercepts or overlaps the relevant acoustic response. Relative timing information is
maintained through control the window size and its step progression.

There is however, a trade off between the overall time taken to perform the FFT on
successive windows and the resolution of the temporal response required. This trade off
is dependant on:

- the position in time of the acoustic responses relative to the start of the sample
  window
- the size of the sliding window
- the magnitude of the step or slide the window takes with each successive shift.

The sooner the FFT algorithm intercepts the acoustic waveforms, the sooner a temporal
response can be found. Therefore ideally the acoustic responses should be as close as
possible to the start of the sample window, which could require cropping the acoustic
responses to minimise the amount of leading 0Hz waveform. The window on which the
FFT operates, and which is moved along the data stream should ideally be small to
optimize temporal resolution. Allied with the size of the window, the magnitude of the
shift or slide the window takes is important. The window could be moved by one
sample point each time which would result in the best possible temporal resolution
result, but the time taken to perform all the FFTs required could be prohibitive.

For this analysis the beginning and end of the FFT was truncated to remove any DC
artefacts and an FFT window size of 1000 samples was used which was shifted by 50
samples each time. These FFT settings were considered adequate to maintain a suitable
level of temporal information. Since each sample or bin has a temporal resolution of
400ns, shifting by 50 bins each time gives an overall temporal resolution of 20µs. However, while this methodology appears to offer benefits, the FFT window size is of concern, especially in connection with the frequency resolution offered by the FFT.

Ifeachor et al [87] (p116) and Brigham [88] (p170) indicate that the frequency resolution, \( f_R \) of an FFT is given by equation (8.8).

\[
f_R = \frac{f_s}{N} \tag{8.8}
\]

Where \( f_s \) is the sampling rate used and \( N \) is the FFT size. The experimental acoustic data used for the FFT analysis contains 70,000 samples, \( N \), at a sample rate, \( f_s \), of 2.5MHz. A single FFT, performed on this entire sample set produces a frequency resolution of 35.7Hz. While this frequency resolution is quite good, an FFT using these parameters would result in no temporal information being available. However, if the FFT is limited to operate on a window of just 1000 samples with the same sampling rate, the resultant frequency resolution, from equation (8.8), becomes 2500Hz. Obviously this latter result would appear to offer a greatly reduced frequency selectivity than if the full 70000 bin window was used. However, what has not been taken into account is the amount by which the FFT window is shifted each time. Given a 70k total sample size, a window consisting of 1000 bins that is consecutively shifted by 50 bins can be mapped 1381 times into the total sample window, i.e.

\[
\text{FFT size} - \frac{\text{(Window size-Step size)}}{\text{Step size}}
\]

This will give an overlap of 950 samples of the previous window with every shift. Therefore the FFT only sees 50 new samples every time and it is these new 50 samples that determine the selectivity of the system. Since both tone generators are designed to be highly efficient or pure in terms of tonal output, using a wide frequency resolution
does not pose as significant a problem as it appears at first glance, providing the magnitude of the window shift employed is small enough. This was verified during analysis where windows of 1000 and 10000 respectively were used during FFT run time on the data from the tone generators at an in-pipe temperature of 293K. The theoretical frequency resolutions were calculated at 2500Hz and 250Hz respectively. The results however, were identical in practice as seen in Table 28.

**Table 28 Effect of FFT window size on average and pk-pk temperature results**

<table>
<thead>
<tr>
<th>FFT Window Size</th>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Average</td>
</tr>
<tr>
<td>1000</td>
<td>3k</td>
<td>291.15</td>
</tr>
<tr>
<td></td>
<td>6k</td>
<td>295.27</td>
</tr>
<tr>
<td></td>
<td>10k</td>
<td>277.32</td>
</tr>
<tr>
<td>10000</td>
<td>3k</td>
<td>291.15</td>
</tr>
<tr>
<td></td>
<td>6k</td>
<td>295.27</td>
</tr>
<tr>
<td></td>
<td>10k</td>
<td>277.32</td>
</tr>
</tbody>
</table>

In all cases the FFT window was shifted by 50 samples or bin positions each time and the decision threshold, which indicates a valid frequency component, was set to 200. This meant that the FFT operation for that particular tone generator halted when a frequency component of magnitude 200 or more was recorded. Because of the similarity in the results shown in Table 28 for the different window sizes, it was decided to use a window size of 1000 for the FFT analysis. Window shift and threshold magnitudes were fixed at 50 and 200 respectively. The results from the sliding FFT analysis, using the same data set as was used with previous analysis methodologies, for an in-pipe temperature of 293K, are given in Table 29.
Table 29 Sliding FFT average, pk-pk and standard deviation temperature results for an in-pipe $T = 293K$

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
<th>Sliding FFT Result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Average</td>
<td>291.15</td>
</tr>
<tr>
<td>3k</td>
<td>Pk – Pk</td>
<td>284.57</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>50.72</td>
</tr>
<tr>
<td>6k</td>
<td>Average</td>
<td>295.27</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>141.12</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>30.19</td>
</tr>
<tr>
<td>10k</td>
<td>Average</td>
<td>264.86</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>62.61</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>13.50</td>
</tr>
</tbody>
</table>

Figure 87 shows a plot of the average and pk-pk results for each of the tone generators using an FFT threshold analysis, as presented in Table 29.

Figure 87 Average and pk-pk spread of temperature using Sliding FFT analysis for TGs at 293K in-pipe temperature

Table 30 shows the average, peak-peak and standard deviation of the experimental data used previously, for an in-pipe temperature of 373K, using the sliding FFT analysis.
Table 30  Basic threshold versus sliding FFT results for in-pipe T = 373K

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K)</th>
<th>Sliding FFT Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td>Average</td>
<td>377.70</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>361.52</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>69.49</td>
</tr>
<tr>
<td>6k</td>
<td>Average</td>
<td>340.42</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>246.08</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>53.49</td>
</tr>
<tr>
<td>10k</td>
<td>Average</td>
<td>333.44</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>70.28</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>22.10</td>
</tr>
</tbody>
</table>

Figure 88 graphically illustrates the sliding FFT average and peak-to-peak results from Table 30.

Figure 88 Average and pk-pk spread of temperature using Sliding FFT analysis for the TGs at 373K in-pipe temperature

Figure 89 and Figure 90 show the standard deviation trends as tone generator oscillation frequency increases from 3kHz to 10kHz for both 293K and 373K in-pipe temperatures.
Interestingly the calculated temperature differential, $\Delta T$ does not show the same trend as seen with previous analysis methodologies. In previous cases the results for the 3kHz and 6kHz tone generators were completely unsatisfactory at detecting an 80K change in temperature, but the result for the 10kHz tone generators was satisfactory, producing an average error of between 2.01% and 3.13% depending on the threshold chosen (basic threshold analysis). For the sliding FFT analysis, using the same data set as was used with the basic threshold operation, the results are reversed with the 3kHz tone
generators showing an average temperature difference more in keeping with the actual measured difference of approximately 80K. This result is shown in Table 31 where the 3kHz tone generator produced an error of 8.19% as compared with 36.4% and 14.28% errors for the 6kHz and 10kHz tone generators respectively.

Table 31 Calculated ΔT for a measured ΔT of 80K using a sliding FFT analysis

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Actual Pipe T (K)</th>
<th>Calculated pipe T (K)</th>
<th>ΔT (K)</th>
<th>% error</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td>373</td>
<td>377.70</td>
<td>80</td>
<td>8.19</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>291.15</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6k</td>
<td>373</td>
<td>340.42</td>
<td>80</td>
<td>43.58</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>295.27</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10k</td>
<td>373</td>
<td>333.44</td>
<td>80</td>
<td>14.28</td>
</tr>
<tr>
<td></td>
<td>293</td>
<td>264.86</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The sliding FFT analysis produces similar standard deviation and pk-pk spread results as for the basic threshold analysis, producing improving standard deviation and peak-to-peak results with increasing tone generator oscillation frequency. However, aside from the reversal in the 80K temperature difference discrimination, there was no obvious improvement from the sliding FFT experimental results over those results obtained from the basic threshold analysis methodology. This lack of improvement over the basic thresholding methodology coupled with the fact that the sliding FFT operation is significantly more computationally intensive than the basic threshold operation makes the FFT methodology less than attractive.
8.7 Squaring pre-threshold analysis

Examining the acoustic responses in more detail revealed a phase variation between consecutive pressure pulse iterations which is the predominant cause of the large variation in pk-pk results already detailed, especially for the lower frequency tone generators. This phase variation is seen in Figure 91, in which two consecutive acquisitions, using the 3kHz tone generators at an in-pipe temperature of 293K are comparatively overlapped, with the TG1 responses aligned. This phase variation results in inconsistencies in the determination of the time difference, $\Delta t$, between TG1 and TG2 responding to consecutive pressure waves. In the acoustic responses seen in Figure 91, for a given threshold position, one of the responses yielded a $\Delta t$ of 2.8324ms while the second yielded a $\Delta t$ of 2.7ms. These times resulted in calculated temperatures of 287.96K and 316.54K respectively, a difference of 28.58K. The pressure ratios for the plots displayed in Figure 91 were 1.0451 and 1.0456 respectively.
Performing an error contribution investigation for a pressure ratio variation of 0.01 which is twenty times larger than the actual pressure difference involved in the plots shown in Figure 91, with all else being equal yields a temperature variation of only approximately 4.6K. Therefore it can be said that the pressure variation involved in the plots seen in Figure 91 plays a minor part, if any in the calculated temperature difference between the two iterations.

The TG2 responses shown in Figure 91 appear to be approximately 90° out of phase. In actual fact, the exact phase difference is almost impossible to determine, because the individual tone generator responses are stochastic in nature and have no inherent phase relationship. Even when consecutive responses appear to be in phase they could in reality be a full period or greater in error. As such the hypothesis was formed that the phase difference could be due to startup variations in the tone generators whereby one of the tone generators starts out of phase relative to the other. Squaring of the acoustic data before analysis was an approach that was considered to overcome or minimise this issue.

The raw data that was used with previous analysis methodologies was therefore squared and reanalysed using the standard threshold algorithm given in Figure 64. Table 32 shows the resultant average, peak-to-peak and standard deviation of the temperature results for the three tone generator frequencies at different thresholds at an in-pipe temperature of 293K. Table 33 shows the average, peak-to-peak and standard deviation of the temperature results at an in-pipe temperature of 373K.
Table 32 Average and pk-pk and std dev temperature results using squared data at 293K in-pipe temperature

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K) result at threshold %</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
<th>60%</th>
</tr>
</thead>
<tbody>
<tr>
<td>3k</td>
<td>Average</td>
<td>308.93</td>
<td>323.16</td>
<td>317.73</td>
<td>327.98</td>
<td>343.88</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>253.94</td>
<td>379.94</td>
<td>416.48</td>
<td>532.46</td>
<td>670.20</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>60.14</td>
<td>83.21</td>
<td>89.76</td>
<td>125.42</td>
<td>144.74</td>
</tr>
<tr>
<td>6k</td>
<td>Average</td>
<td>253.75</td>
<td>251.44</td>
<td>246.39</td>
<td>224.29</td>
<td>213.37</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>134.06</td>
<td>143.57</td>
<td>151.86</td>
<td>249.35</td>
<td>122.94</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>31.08</td>
<td>32.72</td>
<td>37.13</td>
<td>36.35</td>
<td>39.07</td>
</tr>
<tr>
<td>10k</td>
<td>Average</td>
<td>231.51</td>
<td>233.58</td>
<td>236.42</td>
<td>241.51</td>
<td>245.82</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>52.49</td>
<td>51.38</td>
<td>44.05</td>
<td>57.66</td>
<td>56.16</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>13.96</td>
<td>11.57</td>
<td>10.96</td>
<td>14.44</td>
<td>14.78</td>
</tr>
</tbody>
</table>

Table 33 Average and pk-pk temperature results using squared data at 373K in-pipe temperature

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Temperature (K) result at threshold %</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
<th>60%</th>
</tr>
</thead>
<tbody>
<tr>
<td>3kHz</td>
<td>Average</td>
<td>296.99</td>
<td>293.46</td>
<td>282.71</td>
<td>311.97</td>
<td>344.48</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>318.42</td>
<td>291.69</td>
<td>228.27</td>
<td>335.58</td>
<td>723.59</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>64.80</td>
<td>57.19</td>
<td>48.30</td>
<td>62.07</td>
<td>109.66</td>
</tr>
<tr>
<td>6kHz</td>
<td>Average</td>
<td>316.77</td>
<td>318.88</td>
<td>328.71</td>
<td>347.71</td>
<td>392.38</td>
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<tr>
<td></td>
<td>Pk – Pk</td>
<td>224.62</td>
<td>294.37</td>
<td>279.04</td>
<td>364.96</td>
<td>538.87</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>42.77</td>
<td>49.82</td>
<td>53.52</td>
<td>60.50</td>
<td>98.93</td>
</tr>
<tr>
<td>10kHz</td>
<td>Average</td>
<td>339.16</td>
<td>346.45</td>
<td>353.83</td>
<td>371.65</td>
<td>462.85</td>
</tr>
<tr>
<td></td>
<td>Pk – Pk</td>
<td>123.32</td>
<td>102.94</td>
<td>99.74</td>
<td>188.94</td>
<td>354.82</td>
</tr>
<tr>
<td></td>
<td>Std Dev</td>
<td>23.51</td>
<td>18.19</td>
<td>18.35</td>
<td>32.66</td>
<td>111.53</td>
</tr>
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Analysis of the experimental results shows that for low threshold settings, squaring of the normalized raw data, pre-threshold analysis, provides an improvement in the peak-to-peak results, versus those results obtained from a basic threshold analysis, as shown
in Table 18 and Table 19. Standard deviation results, especially at low threshold settings also show an improvement, as seen in Figure 92 and Figure 93, over the basic threshold results presented and plotted in Table 18, Figure 68 and Table 19, Figure 72.

Figure 92 Standard Deviation temperature results at various thresholds using squared acoustic data settings at 293K in-pipe temperature

Figure 93 Standard Deviation temperature results at various thresholds using squared acoustic data settings at 373K in-pipe temperature
8.8 Pre-Oscillation

During experimentation and subsequent analysis it was considered that the oscillation start up mechanism, resulting from the pressure wave, may have been a contributing factor to the observed variation in the measured values of $\Delta t$. One approach, that was presented earlier, was to square the acoustic data prior to analysis. However, another approach was considered that hypothesized that if the tone generators were constantly oscillating at a low intensity, then the pressure pulse or wave injected into the system through the solenoid valve would result in the amplitude of the constant oscillation increasing for the duration of the pressure wave. Since the tone generators oscillate constantly, there should be no phase shift due to start-up mechanisms. This approach required a change of hardware whereby the tubular blow-down pipe seen in Figure 56, was replaced with a square pipe, in conjunction with a rubber fitting to facilitate connection to the D892DPV valve, see Figure 94. A square blow-down pipe was used purely for the convenience of affixing a low pressure pre-oscillation air line on the tone generator side of the valve which is clearly seen in Figure 94 and Figure 95.

Figure 94 Apparatus showing pre-oscillation air line (Blue)
The purpose of the air-line attached post-valve was to provide a constant low pressure flow of air into the blow-down pipe that would cause the tone generators to oscillate constantly at a low acoustic intensity. A pressure regulator was used to regulate the constant flow of air into the blow-down pipe and hence control the level, and intensity of the continuous oscillation from the tone generator. This regulator can be seen attached to the pre-oscillation air-line in Figure 95.

With the tone generators oscillating constantly, a pressure wave induced into the blow-down pipe with the actuation of the valve, caused an amplitude change in the oscillation tone which was detected and analyzed using standard techniques. This approach is akin to the electronic engineering method of maintaining a trickle current in a circuit to assist in startup and reaction times. This *pre-oscillation* effect is seen in Figure 96, which shows two acoustic traces overlapped, one with pre-oscillation, and the second without pre-oscillation.
Figure 96 Combined plot showing normal and pre-oscillated acoustic results

With pre-oscillation, the transition from background oscillation to pressure wave response is smooth, as opposed to that without the pre-oscillation. This smoother transition meant that it was more likely that a specific and consistent threshold position would be achievable. This effect is seen in Figure 97, which shows a zoomed in version of the plots shown in Figure 96, with a 60% threshold position indicated. In Figure 97, the 60% threshold position triggers easily on the highlighted peak when pre-oscillation was used. However, when pre-oscillation was not used the same 60% threshold setting instead triggered on the next periodic peak. Such a threshold position slippage is not a problem, provided both TG1 and TG2 responses produce the same slippage. However, in reality this is an unlikely possibility.
However, there is one obvious shortcoming with the use of *pre-oscillation*, and that is that low values of threshold positions are meaningless, see Figure 96, where thresholds much below 50% are within the pre-oscillation region. However, despite this, the use of pre-oscillation of the tone generators has advantages.

Analyzing the experimental results captured using the setup shown in Figure 95 shows that the calculated temperature results are better with pre-oscillating tone generators than without. The results for the 30% and 40% thresholds when using pre-oscillation are ignored due to the action of the pre-oscillation. With low threshold settings, the trigger position occurred within the pre-oscillation region, meaning the amplitude change that resulted from the transit of the pressure wave was not detected. Figure 98 shows a typical pressure wave/pulse acoustic responses for both pre-oscillating and non pre-oscillating tone generators with a 50% threshold position indicated. Clearly seen is one of the benefits of pre-oscillation: the thresholder detects a peak on the pre-oscillating plot several periods before it detects a valid peak on the plot without pre-oscillation.
Table 34 shows a comparison between results obtained using the experimental apparatus shown in Figure 95, both with and without pre-oscillation. Only the 3kHz and 10kHz tone generators were used for this experiment.

Table 34 Average, pk-pk and standard deviation results both with and without pre-oscillation at different threshold positions at an in-pipe $T=293K$

<table>
<thead>
<tr>
<th>TG Frequency (Hz)</th>
<th>Pre-Oscillation</th>
<th>Temperature (K) at threshold %</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
<th>60%</th>
<th>70%</th>
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<tbody>
<tr>
<td>3k</td>
<td>No</td>
<td>Averag</td>
<td>253.10</td>
<td>260.62</td>
<td>257.77</td>
<td>269.80</td>
<td>238.93</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pk – Pk</td>
<td>223.29</td>
<td>201.37</td>
<td>178.79</td>
<td>231.93</td>
<td>288.59</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Std Dev</td>
<td>46.89</td>
<td>44.17</td>
<td>42.21</td>
<td>54.57</td>
<td>68.84</td>
</tr>
<tr>
<td></td>
<td>Yes</td>
<td>Averag</td>
<td>—</td>
<td>203.14</td>
<td>247.37</td>
<td>248.20</td>
<td>235.33</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pk – Pk</td>
<td>—</td>
<td>178.30</td>
<td>73.32</td>
<td>87.64</td>
<td>122.35</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Std Dev</td>
<td>—</td>
<td>48.16</td>
<td>20.18</td>
<td>24.38</td>
<td>26.88</td>
</tr>
<tr>
<td>10k</td>
<td>No</td>
<td>Averag</td>
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<td>206.02</td>
<td>206.03</td>
<td>204.53</td>
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<td>Pk – Pk</td>
<td>105.65</td>
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<td>106.84</td>
<td>123.91</td>
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<td></td>
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<td>Std Dev</td>
<td>23.39</td>
<td>23.14</td>
<td>21.75</td>
<td>21.79</td>
<td>27.63</td>
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<tr>
<td></td>
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<td>—</td>
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<td>295.81</td>
<td>302.01</td>
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<td>Pk – Pk</td>
<td>—</td>
<td>43.62</td>
<td>44.40</td>
<td>42.49</td>
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<td></td>
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<td>—</td>
<td>8.84</td>
<td>9.96</td>
<td>11.15</td>
<td>11.47</td>
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</table>
Plotting the average and peak-to-peak results for the 10kHz tone generators shows a dramatic improvement in results using pre-oscillation as opposed to not using it, see Figure 99 and Figure 100.

**Figure 99** Average and pk-pk results for 10kHz tone generators without pre-oscillation at an in-pipe T=293K

**Figure 100** Average and pk-pk results for 10kHz tone generators with pre-oscillation at an in-pipe T=293K
Figure 101 plots the standard deviation results from the data presented in Table 34, for the 3kHz and 10kHz tone generators. The improvement that was observed earlier for the peak-to-peak results is, once again clearly seen with the use of pre-oscillation.

Figure 101 Standard deviation results at various thresholds for 3kHz and 10kHz TGs both with and without pre-oscillation at an in-pipe T=293K

Ignoring the 40% threshold results for the plots with pre-oscillation applied, for reasons detailed earlier, the use of pre-oscillation is seen to dramatically improve the peak-to-peak spread and the standard deviation of the temperature results. This is an improvement over and above that seen solely due to increasing the tone generator oscillation frequency.

Interestingly, the peak-to-peak results in Table 34 for the normal condition, i.e. without pre-oscillation are on the order of 50% better than those given in Table 18 using the same tone generator and in-pipe temperature. This improvement could have been due to the differences in the experimental apparatus, i.e. square pipe versus round pipe or
possibly due to the use of the piece of rubber tubing, which is seen in Figure 95. The rubber tubing served two purposes on the apparatus:

1. it facilitated the connection between the square pipe with the solenoid valve
2. it helped reduce the solenoid impact impulse that was transmitted into the blow-down pipe with each actuation.

However, while this improvement in results between the data presented in Table 18 and Table 34 is interesting, it is not of concern due to the differences in the experimental setups. Rather the specific variations between the data presented in Table 34, both with, and without pre-oscillation are. The improvement in the peak-to-peak and standard deviation results with increasing tone generator frequency for that data without pre-oscillation, are commensurate with data presented previously, and as has been proven mathematically in equation (8.6) and shown in Table 17. In addition to the improvement seen due to tone generator frequency, there is a significant improvement in the results when pre-oscillation is applied as opposed to the case without pre-oscillation. For both tone generator frequencies tested, the results, when pre-oscillation is applied, shows an improvement of better than 50% compared to the case when the tone generators are not pre-oscillated.

It is also worth noting that the results for the 10kHz tone generators, with pre-oscillation applied, more accurately captured the in-pipe temperature during the experiment, ~293K, than any of the other results. Temperature differential measurements were not possible on this apparatus due to the mechanism of heating employed.
8.9 Summary

This chapter details several analysis techniques and presents the results of same. A contributed error analysis shows that the higher the oscillation frequency of the tone generators the better the resultant determination of the temporal difference will be. This is borne out by analysis using a basic percentage threshold operation. The 10kHz tone generators offered significant improvement over the lower frequency tone generators not only in terms of improved peak-to-peak and standard deviation results, but also in terms of being able to accurately track a deliberate temperature change.

Interestingly, the one analysis methodology that was expected to perform well, the sliding FFT did not perform as well as the other methodologies such as the basic threshold analysis. This was a surprising result given the spectral purity of the tone generators and the selectivity of the FFT algorithm. On the other, the use of pre-oscillation did offer an improvement to even the basic threshold operation. The effect of the pre-oscillation was to remove or help to remove issues associated with oscillation startup. Since the tone generators, with pre-oscillation, were already oscillating at a low level, the propagation of the high pressure pulse resulting from the actuation of the valve simply causes the amplitude of the oscillation to increase. It has been shown that this approach offers significant improvement over the non pre-oscillation case.
CHAPTER 9

9.1 Summary

An in depth examination of the pressure wave and its influence on passive acoustic transducers that could in turn be incorporated in a ‘blow-down’ system has been conducted (Chapter 3 and 4). A list of criteria, considered essential for any tone generator employed for the investigation of this work, was developed. The Helmholtz resonator was considered but deemed unsuitable for this investigation for several reasons, including its physical parameters, i.e. cavity size, neck opening area etc., versus oscillation frequency.

Based on the criteria introduced in chapter 4, the labial flue pipe edge tone generator was introduced in chapter 6 and it was deemed suitable for the investigation. While a discussion of the detailed operation of the jet mechanism involved in the pipe was beyond the scope of this work, some of the basic concepts surrounding the labial flue pipe as a tone generator were developed. This resulted in the decision to use a square rather than a tubular pipe for the investigation. Original software, developed to optimise the design of the flue pipe tone generator with reference to its physical geometry, pressure differentials, tonal frequency and efficiency or purity (Ising number), was introduced. This included the representative Reynolds number pertaining to the air jet through the air jet or wind sheet thickness (WST). A novel connection was made between the Reynolds number and the Ising number in relation to optimising the jet mechanism of the labial flue pipe for tonal purity. This was the first reported instance of this phenomenon.
Chapter 7 presented the hardware aspect of the research and detailed the advantages and disadvantages of several different pressure release mechanisms. These mechanisms included a diverse range including an engine head, a ball valve, a solenoid valve, and even an analogue of Blair’s QUB rig. The solenoid valve was chosen as the pressure release mechanism due to its operational simplicity, repeatability and controllability.

Chapter 8 presented the experimental analysis algorithms and their results. During this research several different analysis techniques were investigated, all based on a thresholding decision point. Three different frequency labial flue pipes were experimented with, 3kHz, 6kHz and 10kHz. An error contribution analysis indicated that the tone generators operating at the higher frequency (10kHz) should perform better than lower frequency tone generators. This result was borne out by the experimental results with the standard deviation of the results showing significant improvement with the use of the 10kHz tone generators as opposed to the use of the 3kHz or even the 6kHz tone generators. Integral to the research was the ability of the tone generators to detect a deliberate temperature differential in the blow-down pipe. Once again, it was found that the higher frequency tone generators, 10kHz, performed significantly better than the lower frequency ones, correctly indicating the temperature differential to within the actual temperature differential error, ±3K, especially for the basic threshold methodology at mid range thresholds.

The effect of pre-oscillating the tone generators was investigated and was demonstrated to have a dramatic effect on reducing the peak-to-peak and standard deviation errors for comparable apparatus.
9.2 Conclusion

A novel, constrained pulsed gas flow relative temperature measurement solution, using acoustic edge tone apparatus has been demonstrated. With economic and environmental pressures making power-plant combustion efficiency more and more important, the thermal monitoring of exhaust gases for optimizing of the Air-Fuel-Ratio in the combustion chamber is critical. The pressure waves that are induced into the exhaust pipe by the action of the valve, propagate at a velocity proportional to the square root of the temperature of the undisturbed gas ahead of it. By monitoring the velocity of these pressure waves the local temperature of the gas within the exhaust pipe could be determined. The measurement of absolute temperatures in the exhaust flow is dependant on many factors such as distance from the release valve, exhaust pipe insulation etc, therefore, it was not considered the primary focus of the work; rather the ability to monitor a temperature change was considered the more critical. As such, it has been shown that, for a deliberate and measureable temperature change in the proof-of-concept apparatus, the higher frequency labial flue pipe tone generators were easily able to accurately perform this measurement.

One of the ultimate goals in the area of pulsed flow gas temperature measurement is the ability to monitor temperature change on a cycle-by-cycle (CBC) basis. The apparatus outlined in this work required averaging of several exhaust valve cycles, to produce accurate results. However, with the improvements seen in the peak-to-peak and standard deviation results as tone generator frequency increased, the levels of averaging required are expected to reduce with continued further increases in tone generator frequency, if not to actual CBC rates, then certainly to low, single figure multiples of exhaust valve cycles. This reduction in averaging could be further enhanced through the demonstrated
improvement in peak-to-peak and standard deviation results seen with the introduction of pre-oscillation.
9.3 Future Work

Several interesting opportunities exist for future research:

1. It has been shown that higher frequency oscillation performs better from both a theoretical and practical point of view than low frequency oscillation. Therefore the tone generator frequency should be increased beyond the human audio range and into the ultrasonic, possibly through the use of Hartmann or some other edge-tone or jet-edge acoustic mechanism. This enhancement will involve the acquisition or development of suitable ultrasonic detectors. Ideally the acoustic range should be taken above 100kHz which is outside the hearing range of most animals.

2. System modelling. With a specific rather than a proof-of-concept blow-down system in place, modelling of the specific apparatus becomes a real possibility. With a consistent and defined apparatus in mind, a model of expected results could be developed, whereby a correlation based analysis methodology could be employed. This would enhance accuracy in the determination of $\Delta t$ and hence the value of the local acoustic velocity and resultant temperature.

3. The proof-of-concept presented, operated on a discrete, stand alone pressurization and release apparatus, at limited temperature ranges using the audio acoustic spectrum. Operations on an actual internal combustion (IC) engine exhaust pipe, while desirable, presented significant difficulties, with the audio acoustic spectrum employed being the primary difficulty due to the close proximity required between the audio microphones and the operating engine. Increasing the frequency range of the tone generators into the ultrasonic range, with suitable detection mechanisms would simplify band limiting techniques for
engine noise elimination from the tone generator acoustic signal. Equally important when operating on an IC engine are the rigorous health and safety requirements and the correct and appropriate venting of the exhaust gas emissions.

4. LabVIEW, running on a PC is ideal for investigative proof of concept purposes, but to achieve a real life cycle-by-cycle (CBC) rate will require a significant increase in algorithm processing power. The acquisition and analysis software employed in the work will need to be adapted to operate on a DSP style platform, e.g. running an executable version of LabVIEW.
References


References


[37] W. Enfield and S. Webber, "Institutes of Natural Philosophy: Theoretical and Practical," 3 ed. Cummings and Hilliard: Boston, 1820


[40] IPTS-68, "Echelle Internationale Practique de temperature de 1968. Comptes rendues de la 13e Conference Generales des Poids et Mesure et Comite Consultatif de Thermometrie, 8e session.," in International Practical Temperature Scale, 1968


References


# Appendix A

## Nomenclature

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<th>Name</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
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<td>Atmospheric Pressure</td>
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<tr>
<td>Driver Pressure</td>
<td>$P$</td>
<td>—</td>
<td>Pa or Bar</td>
</tr>
<tr>
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<td>—</td>
<td>kg</td>
</tr>
<tr>
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<td>K</td>
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<td>734</td>
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<td>m·s$^{-1}$</td>
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<tr>
<td>Force</td>
<td>$N$</td>
<td>—</td>
<td>Newton</td>
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<tr>
<td>Torque</td>
<td>$\tau$</td>
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</tr>
<tr>
<td>Cents</td>
<td>$\phi$</td>
<td>—</td>
<td>1 semitone = 100¢</td>
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Appendix B

Kulite XT-190-100A

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<th>MINIATURE RUGGEDIZED</th>
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<tr>
<td>XT-190 (M) SERIES</td>
</tr>
<tr>
<td>Easy Installation</td>
</tr>
<tr>
<td>High Natural Frequency</td>
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</tbody>
</table>

The ruggedness of this sensor has not compromised its performance. It was designed for ease of installation.

### Optional Connector Version

**APPENDIX B**

---

#### INPUT

- **Pressure Range**
  - Measured in PSI (bar)
  - Different ranges available

#### Operational Mode

- **Absolute**
- **Gage**
- **Sealed Gage**
- **Differential**

#### Over Pressure

- 2 Times Rated Pressure to a Maximum of 8000 PSIG (69 bar)

#### Burst Pressure

- 3 Times Rated Pressure to a Maximum of 8000 PSIG (69 bar)

#### Pressure Media

- All Non-conductive, Non-toxic Liquids or Gases

#### Read Electrical Excitation

- 3000 Ohms (Nom.)

#### Maximum Electrical Excitation

- 4500 Ohms (Nom.)

#### Input Impedance

- 1000 Ohms (Nom.)

#### Output Impedance

- 1000 Ohms (Nom.)

#### Full Scale Output (FSO)

- 100 mV (Nom.)

#### Residual Uncertainty

- ±5 mV (Typ.)

#### Combined Non-linearity, Hysteresis, and Repeatability

- ±0.1% FSO Full Range ±0.5% FSO (Max.)

#### Resolution

- Infinesimal

#### Natural Frequency (KHz) (Typ.)

- 150
- 175
- 241
- 300
- 390
- 590
- 760
- 1000
- 1400

#### Acceleration Sensitivity (%FSG/psig)

- 1.5 x 10^{-5}
- 1.0 x 10^{-5}
- 5.0 x 10^{-6}
- 3.0 x 10^{-6}
- 1.5 x 10^{-6}
- 0.5 x 10^{-6}
- 6.0 x 10^{-7}
- 4.5 x 10^{-7}
- 2.0 x 10^{-7}
- 2.0 x 10^{-7}

#### Insulation Resistance

- 100 Megohm/Min. @ 50 VDC

#### Environmental Operating Temperature Range

- -45°F to +250°F (-40°C to +120°C)

#### Comparted Temperature Range

- -45°F to +180°F (-40°C to +80°C)

#### Thermal Zero Shift

- ±1% FS of 0°F (Typ.)

#### Thermal Sensitivity Shift

- ±1% /°F (Typ.)

#### Steady Acceleration

- 11,000g (Max.)

#### Linear Vibration

- 10-1,000 Hz Sinus, 100g (Max.)

#### Physical Electrical Connection

- 4 Conductor 30 AWG Shielded Cable 30" Long

#### Weight

- 4 Grams (Nom.) Excluding Cable

#### Pressure Sealing Principle

- Fully Active Four-arm Wheatstone Bridge Electrically Isolated Silicon Si

#### Mounting Torque

- 15 Inch-Pounds (Max.) 17 Nm

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**Note:** Custom pressure ranges, accessory and mechanical configurations available. Dimensions are in inches, dimensions in parentheses are in millimeters. Conformal coating and hysteresis of our products may result in additional changes with or without notice. All dimensions nominal (C)

**KULITE SEMICONDUCTOR PRODUCTS, INC.**  One Willow Tree Road  •  Lenox, New Jersey 07850  •  Tel: 201 461-0600  •  Fax: 201 461-0900  •  http://www.kulite.com

---

**B1**
Appendix C

National Instruments PCI-6133 Specifications

<table>
<thead>
<tr>
<th>General</th>
<th>Counters/Timers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Form Factor</td>
<td>PCI</td>
</tr>
<tr>
<td>OS Support</td>
<td>Win, RT, Linux</td>
</tr>
<tr>
<td>DAQ Family</td>
<td>S-Series</td>
</tr>
<tr>
<td>LabVIEW Support</td>
<td>Yes</td>
</tr>
<tr>
<td>No. Cntrs/timers</td>
<td>2</td>
</tr>
<tr>
<td>Resolution</td>
<td>24-bits</td>
</tr>
<tr>
<td>Max Source Freq</td>
<td>20MHz</td>
</tr>
<tr>
<td>Min ip Pulse Width</td>
<td>10ns</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Analog Inputs</th>
<th>Logic Levels</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. Channels</td>
<td>8 SE/8 DI</td>
</tr>
<tr>
<td>Sample Rate</td>
<td>≤3MS/s/ch</td>
</tr>
<tr>
<td>Resolution</td>
<td>14 bits</td>
</tr>
<tr>
<td>Simul Sampling</td>
<td>Yes</td>
</tr>
<tr>
<td>Max Voltage Range</td>
<td>-10V..+10V</td>
</tr>
<tr>
<td>Range Accuracy</td>
<td>4660 uV</td>
</tr>
<tr>
<td>Range Sensitivity</td>
<td>432 uV</td>
</tr>
<tr>
<td>Min Voltage Range</td>
<td>-1.25V..+1.25V</td>
</tr>
<tr>
<td>Range Accuracy</td>
<td>740 uV</td>
</tr>
<tr>
<td>Range Sensitivity</td>
<td>68.8 uV</td>
</tr>
<tr>
<td>No. Ranges</td>
<td>4</td>
</tr>
<tr>
<td>On Board Memory</td>
<td>64MB</td>
</tr>
<tr>
<td>Max Range</td>
<td>0..5V</td>
</tr>
<tr>
<td>Timebase Stability</td>
<td>100ppm</td>
</tr>
<tr>
<td>GPS Sync</td>
<td>No</td>
</tr>
<tr>
<td>Pulse Generation</td>
<td>Yes</td>
</tr>
<tr>
<td>Buffered</td>
<td>Debounce/Glitch No</td>
</tr>
<tr>
<td>No. DMA Channels</td>
<td>1</td>
</tr>
<tr>
<td>Sync Bus</td>
<td>Yes</td>
</tr>
<tr>
<td>Triggering</td>
<td>Analog/Digital</td>
</tr>
<tr>
<td>Max Range</td>
<td>0..5V</td>
</tr>
<tr>
<td>Timing/Trig/Synch</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Analog Outputs</th>
<th>Physical Specs</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. Channels</td>
<td>0</td>
</tr>
<tr>
<td>Digital I/O</td>
<td>Length</td>
</tr>
<tr>
<td>No. Channels</td>
<td>8 DIO</td>
</tr>
<tr>
<td>I/O Connector</td>
<td>68pin M SCSI-II</td>
</tr>
<tr>
<td>Length</td>
<td>31.2 cm</td>
</tr>
<tr>
<td>Width</td>
<td>10.6 cm</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Timing &amp; Hardware</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Clock Rate</td>
<td>10 MHz</td>
</tr>
<tr>
<td>Logic Levels</td>
<td>TTL</td>
</tr>
<tr>
<td>Max I/O Range</td>
<td>0..5V</td>
</tr>
<tr>
<td>Input I Flow</td>
<td>Sinking, Sourcing</td>
</tr>
<tr>
<td>I Drive (Chn/Total)</td>
<td>24mA/192mA</td>
</tr>
<tr>
<td>Handskahing I/O</td>
<td>No</td>
</tr>
<tr>
<td>WatchDog Timer</td>
<td>No</td>
</tr>
</tbody>
</table>
Appendix D

D892DPV Datasheet

PILOT OPERATED SOLENOID VALVE G¼ ~ G1

Hot Water, Steam

<table>
<thead>
<tr>
<th>Wetted Materials</th>
<th>Brass CW614N EN 12165</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Stainless Steel, PTFE,</td>
</tr>
<tr>
<td></td>
<td>EPM PX 70/80 (Main Seal)</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>−10 ~ +70 °C</td>
</tr>
<tr>
<td>Media Temperature</td>
<td>−10 ~ +150 °C</td>
</tr>
<tr>
<td>Media Viscosity</td>
<td>37 mm²/s Max</td>
</tr>
<tr>
<td>Electrical</td>
<td>AC 36VA Inrush</td>
</tr>
<tr>
<td></td>
<td>AC 18VA Holding</td>
</tr>
<tr>
<td></td>
<td>DC 14W</td>
</tr>
<tr>
<td>Protection</td>
<td>IP65 with Connector</td>
</tr>
<tr>
<td>Flow Direction</td>
<td>Arrow on Body</td>
</tr>
</tbody>
</table>

2/2 Normally Closed
For use with Open Systems

<table>
<thead>
<tr>
<th>Connection</th>
<th>Part Number</th>
<th>Orifice mm</th>
<th>Operating Pressure Differential—bar</th>
<th>Flow (l/min)</th>
<th>Weight kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>G¼</td>
<td>D887DPV</td>
<td>11.5</td>
<td>Min: 0.3 AC: 4.5 DC: 4.5</td>
<td>35</td>
<td>0.5</td>
</tr>
<tr>
<td>G½</td>
<td>D888DPV</td>
<td>11.5</td>
<td>Min: 0.3 AC: 4.5 DC: 4.5</td>
<td>50</td>
<td>0.5</td>
</tr>
<tr>
<td>G¾</td>
<td>D889DPV</td>
<td>11.5</td>
<td>Min: 0.3 AC: 4.5 DC: 4.5</td>
<td>55</td>
<td>0.5</td>
</tr>
<tr>
<td>G1</td>
<td>D890DPV</td>
<td>11.5</td>
<td>Min: 0.3 AC: 4.5 DC: 4.5</td>
<td>70</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Standard Voltages: DC: 24V; AC: 24V 50Hz, 110V 50Hz, 230V 50Hz.
Other Voltages available to Special Order.

Dimensions

<table>
<thead>
<tr>
<th>mm</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>G¼</td>
<td>75</td>
<td>95</td>
<td>53</td>
<td>1.4</td>
</tr>
<tr>
<td>G½</td>
<td>75</td>
<td>95</td>
<td>53</td>
<td>1.4</td>
</tr>
<tr>
<td>G¾</td>
<td>75</td>
<td>95</td>
<td>53</td>
<td>1.4</td>
</tr>
<tr>
<td>G¹</td>
<td>85</td>
<td>105</td>
<td>53</td>
<td>22</td>
</tr>
<tr>
<td>G¹</td>
<td>85</td>
<td>105</td>
<td>53</td>
<td>22</td>
</tr>
</tbody>
</table>

Spares Kits

<table>
<thead>
<tr>
<th>All Sizes</th>
<th>SKP11JX-00</th>
</tr>
</thead>
</table>

Options—Change codes as follows

---

M&M International (UK) Ltd  www.mmint.co.uk  sales@mmint.co.uk  01234 855999  01234 856999
Appendix E

Netco Heat Trace

FSU

Electrical heating tape for process heating or temperature maintenance of pipework and vessels where high temperature withstand is required

- Automatically adjusts heat output in response to increasing or decreasing pipe temperature
- Can be cut to length with no wastage
- Will not overheat or burnout

FREEZSTOP ULTIMO

Self-Regulating Heating Tape

Withstands 250°C

- High power outputs up to 80W/m
- Full range of controls and accessories
- Available for 220/240VAC (110/120VAC on demand)

FEATURES

FREEZSTOP ULTIMO is an industrial grade, self-regulating heating tape that can be used for applications ranging from process heating or maintenance of temperature up to 200°C.

It can be cut to length on site and exact piping lengths can be matched without any complicated design considerations.

FREEZSTOP ULTIMO is used where high temperatures are required (up to 200°C) or where the heater must be capable of withstanding high exposure temperatures (up to 250°C un-energised).

Its self-regulating characteristics improve safety and reliability. FREEZSTOP ULTIMO will not overheat or burnout.

The installation of FREEZSTOP ULTIMO heating tape is quick and simple and requires no special skills or tools. Termination, splicing and power connection components are all provided in convenient kits.

OPTIONS

FSU
Unbrazed base heater protected against corrosive chemical solutions and vapours. Heaters must have additional protection from mechanical damage in service.

FSU.N
Nickel plated braid for where traced equipment does not provide an effective earth path, e.g. plastic pipework. Heaters must have additional protection from mechanical damage in service.

FSU.NF
Fluoropolymer outer jacket over nickel plated braid provides additional protection where corrosive chemical solutions or vapours may be present.

HEAT TRACE™

Setting the standards. Leading the way.
SPECIFICATION

MAXIMUM CONTINUOUS TEMPERATURE (energized) 200°C (392°F)

MAXIMUM PERMISSIBLE EXPOSURE TEMPERATURE 250°C (482°F)*

MINIMUM INSTALLATION TEMPERATURE -40°C (~-40°F)

POWER SUPPLY 220 - 240 VAC (110 - 120 VAC on demand)

MAXIMUM RESISTANCE OF PROTECTIVE BRAIDING 18.2 Ohm/km

WEIGHTS & DIMENSIONS

<table>
<thead>
<tr>
<th>Type</th>
<th>Nom. Dims. (mm)</th>
<th>Weight kg/100m</th>
<th>Min. Bending radius @ -20°C</th>
<th>Glend Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>FSU</td>
<td>13.4 x 3.4</td>
<td>7.6</td>
<td>20 mm</td>
<td>M20</td>
</tr>
<tr>
<td>FSU,N</td>
<td>11.4 x 4.4</td>
<td>11.7</td>
<td>25 mm</td>
<td>M20</td>
</tr>
<tr>
<td>FSU,NF</td>
<td>12.2 x 5.2</td>
<td>15.4</td>
<td>30 mm</td>
<td>M20</td>
</tr>
<tr>
<td>FSUw</td>
<td>12.4 x 3.4</td>
<td>9.5</td>
<td>20 mm</td>
<td>M20</td>
</tr>
<tr>
<td>FSUw,N</td>
<td>13.4 x 4.4</td>
<td>13.7</td>
<td>25 mm</td>
<td>M20</td>
</tr>
<tr>
<td>FSUw,NF</td>
<td>14.2 x 5.2</td>
<td>17.7</td>
<td>30 mm</td>
<td>M20</td>
</tr>
</tbody>
</table>

N denotes nickel plated copper braid

APPROVAL DETAILS

Testing Authority Certificate No. Standard
CENELIC EN60079-0/EN60079-7
ATEX 02ATX3012 EN60079-0/EN60079-7 EC82986
CSA 1295278 C22.2 No. 130.1
CSA 1295278 C22.2 No. 130.2
CSA 1295278 C22.2 No. 138

THERMAL RATINGS

Nominal output at 230V when FSU is installed on insulated metal pipe.
W/m

ACCESSORIES

Heat Trace supply a complete range of accessories including terminations, splice kits, and seals, junction boxes and controls. Such items carry separate approvals from the heating tapes.

FURTHER INFORMATION

Please consult the appropriate termination instructions and the Heat Trace Installation, Maintenance and Testing Manual (MWEHT01) for further details.
Appendix F

Presentations, Publications and Patents during this research


• B. Moss, E. Lewis, G. Leen, K. Bremer, and A. Niven, "Pre-oscillation superposition improves standard deviation results of temperature measurement in


- K. Bremer, E. Lewis, B. Moss, G. Leen, S. Lochmann, I. Mueller, “Micro fibre optic pressure sensor based on a glass capillary and a glass diaphragm”, First Asian Conference on e-Business and Telecommunication, 9-10th February, 2009, Changhua City, Taiwan

- K. Bremer, E. Lewis, B. Moss, G. Leen, S. Lochmann, I. Mueller, “Fabrication of a high temperature-resistance optical fibre micro pressure sensor”, IEEE Sixth International Multi-Conference on System, Signal and Devices (SSD'09), March 23rd -26th, Djerba, Tunisia


- K. Bremer, E. Lewis, B. Moss, G. Leen, S. Lochmann, I. Mueller “Conception and preliminary evaluation of an optical fibre sensor for simultaneous measurement of pressure and temperature”, 20th International Conference on Optical Fibre Sensors (OFS-20), October 2009, Edinburgh, Scotland