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SOURCES OF NOISE IN PROFESSIONAL CHAIN SAWS

by

Stephen Ernest Keith, BAsC, MAsC

A thesis submitted to

the Faculty of Graduate Studies and Research

in partial fulfillment of

the requirements for the degree of

Doctor of Philosophy

Department of Mechanical and Aerospace Engineering

Ottawa-Carleton Institute

for Mechanical and Aerospace Engineering

Carleton University

Ottawa, Ontario

September, 1992

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"SOURCES OF NOISE IN PROFESSIONAL CHAIN SAWS"

submitted by

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in partial fulfillment of the requirements
for the degree of Doctor of Philosophy

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Abstract

The sources of noise from two gasoline-powered chain saws (3 and 3.8 kW) were identified at the operator's ear and in the far field, by sequentially suppressing the most intense source. The powerhead noise sources were explored in a laboratory with the 3 kW saw mounted on a space frame test stand using mechanical models of the hand-arm system. Field measurements were made according to ISO 7182.

Field testing showed the relative noise source strengths on the two saws were comparable, with noise levels from the 3.8 kW saw about 5 dB greater than the smaller saw. The level differences were attributed to power, speed and source directivity.

Following are A-weighted sound pressure levels recorded at the operator's ear position on the 3 kW saw: As supplied, the exhaust noise was approximately 106 dB, or 116 dB with the muffler removed. The intake noise was approximately 104 dB, or 108 dB with the muffler removed. Fan noise due to turbulence and aeolian tones was 92 dB, determined by externally motoring the saw. A bearing impact noise level of 92 dB was deduced from increased bearing clearances. Chain impacts with the sprocket or guide bar are caused by polygonal action. Noise from cutting and impacts between chain and guide bar was 91 dB, measured using an acoustical enclosure to eliminate other noise sources. The chain produces noise levels of 87 dB through impacts with the sprocket. Piston slap impacts with the cylinder wall produced noise levels of 85 dB, estimated from measurements with increased piston running clearance. Theory estimating impact power and mobility overestimated measured levels for these impact sources. Low transmission loss through the intake system resulted in 102 dB noise levels, which was reduced to 81 dB by eliminating leakage. The reciprocating piston causes dipole radiation from rigid-body vibration of the chain saw and produced 81 dB of noise. Combustion noise due to the cylinder pressure and stiffness controlled transmission loss was estimated at 68 dB using reciprocity, and a specially modified cylinder pressure transducer.

Only 3-4 dB of noise reduction can be expected by improving exhaust and intake mufflers, before noise from the powerhead determines the overall noise of these saws. Powerhead modifications reduced the noise a further 10 dB, at which point the A-weighted sound level was 91-96 dB at the operator's ear, due to the cutting process.

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All necessary apologies are extended here, in writing, to my wife Renette.

Stephen Keith

Ottawa, 1992

Table of Contents

Abstract.....	ii
Acknowledgments	iii
Table of Contents	iv
List of Tables.....	vii
List of Figures	viii
List of Symbols.....	x
Chapter I Introduction.....	1
Chapter II Significant Noise Sources in Chain Saws.....	9
2.1. Exhaust and intake noise	9
2.2. Combustion.....	12
2.3. Fan noise.....	16
2.4. Rigid body motion.....	22
2.5. Impact noise.....	24
2.5.1. Production of impact noise	24
2.5.2. Structural response of the engine	27
2.5.3. Force spectrum.....	31
2.6. Piston slap.....	31
2.6.1. Effect of clearance on impact magnitude (and hence noise)	35
2.6.2. Effect of speed on piston slap	37
2.6.3. Production of piston slap impact noise	40
2.6.4. Force spectrum.....	41
2.6.5. Comments on the piston side force.....	44
2.6.6. Limitations of the above analysis.....	50
2.6.7. Implications for the measurement of piston slap	51
2.7. Bearing noise	52
2.7.1. Bearings requiring clearance	52
2.7.2. Zero clearance bearings	57
2.8. Chain noise	57
Chapter III Apparatus and Chain Saw Modifications.....	63
3.1. Apparatus and test site for field measurements.....	63
3.1.1. Test site.....	63
3.1.2. Acoustic enclosure for chain saw	63
3.2. Apparatus for laboratory measurements.....	64
3.2.1. Test stand	64
3.2.1.1. Dynamometer speed and load control	65
3.2.1.2. Hand models	65
3.2.1.3. Test stand performance	67
3.2.2. Laboratory environment.....	68
3.3. Chain saw modifications.....	71
3.3.1. Chain saw belt-drive attachments	72
3.3.2. Intake and exhaust modifications	73
3.3.3. Bearings	75
3.3.4. Fan	76

3.4. Vibration-isolated, high temperature cylinder pressure transducer	77
3.5. Computer based data acquisition system.....	80
Chapter IV Experimental Procedures.....	81
4.1. Field measurements.....	81
4.2. Laboratory measurements	82
4.2.1. Measurement of sound pressure	82
4.2.2. Measurement of sound intensity	83
4.2.3. Measurement of acceleration.....	84
4.3. Source-specific measurements	85
4.3.1. Measurement of cylinder pressure	85
4.3.2. Measurement of crankcase pressure	85
4.3.3. Measurement of structural attenuation	86
4.3.3.1. Crankcase and intake system	86
4.3.3.2. Combustion noise from cylinder	87
Chapter V Results and Discussion.....	89
5.1. Field measurements	89
5.1.1. Exhaust, intake and residual noise	89
5.1.2. Residual noise.....	90
5.1.3. Noise radiated by the cutting components	91
5.2. Laboratory measurements	92
5.2.1. Comparison of laboratory versus field measurements	93
5.2.2. Noise from the powerhead	94
5.2.2.1. Impact excitation of the powerhead	94
5.2.3. Sources internal to the powerhead	95
5.2.3.1. Fan noise.....	95
5.2.3.2. Combustion noise	97
5.2.3.3. Crankcase and intake system	98
5.2.3.4. Shaking forces from piston and crankshaft	99
5.2.3.5. Piston slap.....	100
5.2.3.6. Bearing noise	103
5.2.4. Bar chain interaction	107
5.3. Source summary.....	110
Chapter VI Conclusions.....	113
Appendix A. Measurement of Source Characteristics Using Reciprocity	117
A.1. Volume velocity in a small cavity.....	119
Appendix B. Effect of Transducer Movement on Pressure Measured with a	
Vibration Isolated Pressure Transducer.....	123
Appendix C. Other Analysis Techniques Explored	126
Appendix D. Prospects for Noise Reduction	128
D.1. Exhaust.....	128
D.2. Intake.....	129
D.3. Powerhead noise	129
D.4. Fan.....	131
D.5. Bearings.....	131

D.6. Bar chain and cutting.....	134
D.7. Piston.....	134
D.8. Combustion and intake system noise.....	135
D.9. Unbalanced shaking forces	136
Appendix E. Definition and explanation of functions, and equations.....	137
Appendix F. Reference values for decibel levels	141
Appendix G. Equipment Specifications	142
G.1. Chain Saws	142
G.1.1. Large-timber saw.....	142
G.1.2. Pulpwood Saw.....	142
G.1.2.1. General	142
G.1.2.2. Reference condition.....	143
G.1.2.3. Component values for equations.....	143
G.2. Adamson M200 midrange compression driver	143
G.3. Pressure measurement	144
G.3.1. Measurement microphones.....	144
G.3.2. Cylinder pressure	146
G.3.3. Miniature microphones	146
G.3.4. Dynamometer load cell	146
G.4. Acceleration measurement.....	147
G.4.1. Accelerometers	147
G.4.2. Charge amplifiers.....	147
G.5. Miscellaneous.....	148
G.5.1. Filter.....	148
G.6. 1/3 octave analyzer.....	148
G.7. FFT analyzer	148
Appendix H. Publications	150
Figures	152
References.....	189

List of Tables

1. **Field measurements of sound levels for pulpwood and large timber saws 90**
2. **Summary of sound pressure levels for all sources on the pulpwood saw 111**

List of Figures

frontispiece

	Cross cutting wood	xiv
1 1	Exploded diagram of typical chain saw	152
1 2	Cross section of stock chain saw	153
1 3	Spectra for three equal amplitude square waves	154
2.1.	Typical internal combustion engine cylinder pressure spectrum	155
2 2.	Fan blade vector diagram	155
2 3.	Piston and connecting rod geometry and forces	156
2.4.	Graph showing gas and inertial contributions to piston slap side forces	157
2 5	Graph showing piston slap side force	157
2.6.	Graph showing variation of piston slap vibratory power with speed	158
2 7	Polar diagram of connecting rod bearing forces	158
2.8.	Sketch showing comparison of saw chain and silent chain	159
2.9	Illustration of polygonal action when chain meshes with sprocket	159
3 1	Sketch of field test site	160
3 2.	Photo of acoustic enclosure used in cutting tests	161
3 3.	Level difference due to chain saw acoustic enclosure	161
3 4	Photo of laboratory test stand	162
3 5.	Block diagram of measurement system	163
3 6.	Sketch of hand model support for front handle	164
3 7	Graph showing effect of test stand support on saw vibration	164
3.8.	Sketch of laboratory	165
3.9.	Graph showing laboratory deviations from free field	166
3.10.	Graph of differences between laboratory and field acoustical conditions	166
3.11.	Sketch of modifications to stock saw	167
3.12.	Intake and exhaust ducting (field measurements)	168
3 13.	Acoustic intensity mapping of noise leaking from intake system	168
3 14	Graph of acoustical effectiveness of intake modifications	169
3.15.	Photo of dummy fan modifications	169
3 16.	Photo of external motor drive for fan	170
3 17.	Graph of acoustical effectiveness of fan modifications	171
3 18	Sketch of vibration isolated pressure transducer	171
3.19.	Graph of pressure transducer vibration isolation effectiveness	172
4 1.	Photo cutting wood with chain saw	173
4 2.	Photo of operator's ear microphone location (field measurements)	173
4 3.	Photo of laboratory setup for crankcase structural attenuation	174
4 4	Photo of laboratory setup for reciprocity measurements	174
5 1.	Graphs of pulpwood and large-timber saw sources (field measurements)	175
5 2.	Bar and chain noise (field measurements)	176
5 3	Graph showing range of SPL found in reference measurements	177
5.4	Graph showing SPL difference between laboratory and field measurements	177
5 5.	Graph showing difference in acoustic intensity of selected panels	178
5 6.	Graph showing acoustical response of pulpwood saw to hammer impacts	178
5 7.	Graph showing mechanical noise using dummy fan	179

5.8.	Graph of noise from motored fan	179
5.9.	Graph showing change in mechanical noise with speed	180
5.10.	Graph of full load vs. no load cylinder pressure	180
5.11.	Graph of full load vs. no load mechanical noise	181
5.12.	Graph of cylinder structural attenuation.....	181
5.13.	Graph of calculated SPL at operator's ear due to combustion	182
5.14.	Graph of full load vs. no load crankcase pressure	182
5.15.	Graph of calculated SPL at operator's ear from intake system	183
5.16.	Graph showing effect of piston clearance	183
5.17.	Graph showing effect of speed on noise due to piston.....	184
5.18.	Graph of high pass filtered cylinder acceleration time histories.	184
5.19.	Wigner distribution for cylinder acceleration time history	185
5.20.	Graph showing effect of gudgeon pin bearing clearance.....	186
5.21.	Graph showing effect of chain operation	186
5.22.	Graph showing acoustic intensity radiated by chain guide bar	187
5.23.	Bar chart of A-weighted sound pressure levels cutting wood	187
5.24.	Graph showing estimated levels from all significant sources.....	188

List of Symbols

<i>a</i>	constant
<i>acc</i>	acceleration
<i>A</i>	area
<i>AL</i>	acceleration level (dB re 10^{-6} ms^{-2})
<i>b</i>	constant
<i>B</i>	bending stiffness
<i>c</i>	velocity of sound
<i>c_b</i>	structural bending wave velocity
<i>C</i>	damping
<i>d</i>	piston-cylinder or bearing clearance
<i>D</i>	diameter
<i>D₁</i>	fan rotor inlet diameter
<i>D₂</i>	outer diameter of fan rotor
<i>D_{2a}</i>	outer diameter of fan <i>a</i>
<i>D_{2b}</i>	outer diameter of fan <i>b</i>
<i>D_B</i>	cylinder bore diameter (mm)
<i>D_{sp}</i>	specific diameter of fan (dimensionless)
<i>e</i>	internal energy per unit mass of fluid
<i>E</i>	Energy
<i>f</i>	frequency (Hz)
<i>f_n</i>	natural frequency
<i>f_o</i>	single frequency (Hz)
<i>f_Δ</i>	constant dependent on cylinder pressure spectrum slope
<i>f</i>	force in time domain
<i>f_x</i>	component of force in time domain oriented along <i>x</i> -axis
<i>f_y</i>	component of force in time domain oriented along <i>y</i> -axis
<i>f_{CP}</i>	force between piston and cylinder in time domain
<i>f_G</i>	gas force on piston in time domain
<i>f_P</i>	piston inertia force in time domain
<i>F</i>	force in frequency domain
<i>F_{cycle}</i>	force spectrum for a single engine cycle
<i>g</i>	acceleration due to gravity
<i>G</i>	shear modulus
<i>h</i>	material thickness
<i>H</i>	total pressure head
<i>i</i>	$\sqrt{-1}$
<i>I</i>	acoustic intensity
<i>IPN</i>	impact prediction number
<i>K</i>	stiffness
<i>l</i>	ratio of L_R/L_C (crank radius)/(connecting rod length)
<i>L</i>	characteristic length
<i>L_A</i>	perimeter of impact area
<i>L_b</i>	fan blade length, (measured in direction normal to blade chord)

L_c	fan blade chord
L_C	connecting rod length
L_p	chain pitch
L_P	length from center of gravity to piston end of connecting rod
L_R	length from center of gravity to crankshaft end of connecting rod
m	mass
m_C	connecting rod mass
m_g	mass of gas in cylinder
m_p	piston mass
M	Mach number
n	engine speed in revolutions per second (rps)
n_a	fan <i>a</i> speed in revolutions per second (rps)
n_b	fan <i>b</i> speed in revolutions per second (rps)
N	number of vibrational modes
p	pressure
p_{other}	pressure from other unknown sources
p_t	total pressure
p_a	total pressure of fan ' <i>a</i> '
p_b	total pressure of fan ' <i>b</i> '
$\hat{p}$	complex pressure amplitude
$p_R[1.0]$	normalized cylinder pressure source strength
P	power
P	sound power
P_v	vibratory power
P_{vsr}	vibratory power due to side force
PL	sound power level in dB re 10^{-12} W
q_{out}	heat transfer to surroundings per unit mass of fluid
Q	volume flow rate
Q_a	volume flow rate for fan <i>a</i>
Q_b	volume flow rate for fan <i>b</i>
$\hat{Q}$	complex volume flow rate
r	connecting rod radius of gyration
R	gas constant
R	Radius
R_{obs}	radial distance to observer
R_p	chain sprocket pitch radius
R_S	crank radius,
$\mathbf{R}$	position vector
$\mathbf{R}_a$	initial vector position of joint <i>a</i> in chain
$\mathbf{R}_a'$	final vector position of joint <i>a</i> in chain
$\mathbf{R}_b$	initial vector position of joint <i>b</i> in chain
$\mathbf{R}_b'$	final vector position of joint <i>b</i> in chain
$\mathbf{R}_c$	initial vector position of point <i>c</i> on sprocket
$\mathbf{R}_c'$	final vector position of point <i>c</i> on sprocket

R_1	vector position used in reciprocity calculation
R_2	vector position used in reciprocity calculation
Re	Reynolds number
s	Poisson's ratio
SPL	sound pressure level in dB re 2×10^{-5} Pa
$SPL(A)$	A-weighted sound pressure level in dB re 2×10^{-5} Pa
St	Strouhal number
t	time
t_m	measurement duration
T	temperature
u	gas particle velocity in time domain
U	acoustic particle velocity in frequency domain
U_1	tangential velocity of inner edge of fan blade
U_2	fan blade outer tip velocity
v	volume per unit mass of fluid (see also v , v)
V	velocity
V_1	flow velocity at fan blade inlet
V_2	flow velocity at fan blade exit
w	work per unit mass of fluid
x	x axis displacement
y	y axis displacement
Y	Young's modulus
z	structural impedance
z_a	complex acoustical impedance
Z	structural impedance in frequency domain
Z_a	complex acoustical impedance in the frequency domain
Z_s	specific acoustic impedance ($=c/A$)
Z_A	engine structural attenuation

Greek Symbols

α_1	fan streamline inlet angle
α_2	fan streamline exit angle
β	incremental change in crankshaft angle
β_1	fan blade inlet angle
β_2	fan blade exit angle
γ	specific heat ratio
γ	ratio of crankshaft radius to connecting rod length (R_S/L_C)
Δf	frequency band (i.e., 1/3 octave band)
ΔN	number of vibrational modes in a given frequency band (i.e., 1/3 octave band)
η	structural damping
θ	angle
θ_s	angular position of crankshaft

κ	fan noise constant
μ	coefficient of viscosity
v	dimensionless constant $(1-2l^2-r^2/L_C)/l\gamma$
π	pi
ρ	mass density
ρ_a	inlet air density for fan <i>a</i>
ρ_b	inlet air density for fan <i>b</i>
ρ_m	density of material in structure
ρ_o	density of standard air at 20° C
σ	radiation efficiency,
u	ratio of m_C/m_P (connecting rod mass)/(piston mass), (see also v, v)
ϕ	bearing force angle
ϕ	angle
ϕ_C	connecting rod angle
Ψ	dimensionless force
Ψ_C	dimensionless connecting rod inertia force
Ψ_G	dimensionless gas force
Ψ_P	dimensionless piston inertia force
Ψ_S	dimensionless side force
Ω	specific speed of fan

Common Subscripts

<i>C</i>	connecting rod
<i>j</i>	index
<i>P</i>	piston
<i>S</i>	crankshaft
<i>SF</i>	side force
<i>o</i>	reference condition
Δf	measurement spans a finite frequency band

Operators

'	differentiation with respect to time or frequency
·	differentiation with respect to crankshaft angle θ_s
	(ellipsis) absolute value
—	(over bar) time average
Δ	value of quantity measured over a finite interval (finite difference)
$\mathcal{F}\{ \}$	Fourier transform
$\mathcal{H}\{ \}$	Hilbert transform
$\Re\{ \}$	real part of complex number

Cross cutting (bucking) in an Algonquin park logging operation.

Chapter I Introduction

The intense noise produced by chain saws has long been recognized as a potential source of hearing loss to the operator and annoyance to nearby observers¹. Though hearing protectors do much to reduce the risk of hearing loss, questions concerning their performance and comfort during heavy manual work in extremes of temperature and humidity remain, as does the ability of the operator to hear warning sounds. A basic understanding of chain saws and noise source mechanisms is required to develop a strategy for noise reduction. The first step is to identify significant sources and source mechanisms, which requires the ability to compare predictions with the results of experiments. The reduction of chain saw noise, however, remains a formidable challenge due to constraints of size, weight and extreme operating conditions.

Chain saws (fig 1.1, 1.2, frontispiece) represent the evolution of small internal-combustion engines within constraints that define absolute limits for power and weight, together with a need for simplicity and durability under extreme operating conditions. The chain saw is a small, lightweight, portable device, consisting of a chain for cutting wood and a drive engine. The saw is completely self contained; it includes a fuel tank, chain oil tank, and pull-cord rewind starter. Single cylinder, air-cooled reciprocating engines are almost universally employed for this application, though twin-cylinder and rotary engine saws have been produced.

The cutting system is lightweight and compact. The wood is cut and the chips removed by cutting teeth attached to a moving chain. The chain is guided in a grooved track around the edge of a long flat plate, or "guide bar," and is kept constantly lubricated by an oil pump that is gear driven from the crankshaft (fig. 1.2). The chain is driven directly from the crankshaft, at over 16 ms^{-1} , through a centrifugal clutch. The effectiveness of this system is demonstrated by the large-timber saw used in this study. Although the main body of the saw weighs under 10 kg, it can cut down trees up to 2 m in diameter, using a guide bar 0.9 m long, 8 cm wide and 0.6 cm thick. The saw can cut through 0.1 m^2 of hardwood in 6 seconds.

The powerhead is a high speed, single cylinder, two-stroke cycle, crankcase scavenged gasoline engine. Engine lubrication is provided by oil mixed with the gasoline. A diaphragm carburetor pumps and meters fuel from the gas tank using crankcase pressure, allowing operation of the carburetor when inverted. Fuel and air are drawn (through one-way reed valves) into the crankcase by the upward movement of the piston (fig. 1.2).

Next, as the piston moves downward, the crankcase mixture is pressurized. The downward movement of the piston first exposes the exhaust port (not shown), releasing the spent gases from the cylinder. The transfer ports are next exposed, allowing the pressurized mixture in the crankcase to flow into the cylinder around the sides of the piston. Then the piston moves upwards, closing the ports and compressing the charge in the cylinder (fresh mixture is also simultaneously drawn into the crankcase for the next operating cycle). The compressed charge is then ignited by a spark plug which is energized by a flywheel magneto. The piston is again forced down, once more exposing the exhaust port, which releases the spent charge, completing the cycle. The mechanical energy developed by the reciprocating motion of the piston is stored in the flywheel.

Two high-performance, professional chain saws were available for the present study. Though of different design, from two manufacturers, both were powered by single cylinder, two-stroke gasoline engines with intake facing the rear handle (normally held by the right hand). One was a 3 kW, 67 cm³ swept volume, vertical-cylinder, pulpwood saw with forward facing exhaust (fig. 1.1, frontispiece), while the other was a 3.8 kW, 82 cm³ capacity, horizontal cylinder, large-timber saw, exhausting to the right side, away from the operator. The handles of both saws were vibration isolated from the crankcase. In addition, the fuel and oil tanks of the large-timber saw were attached to the crankcase rather than the handles.

In the study of noise radiated from internal combustion engines, combustion has been referred to as the direct exciting force. But, there are also many indirect exciting forces (producing noise), which depend on combustion in some non linear manner². For example, in the pulpwood saw, the rotating components and flywheel have a mass of 0.7 kg and approximate moment of inertia of .002 Nms². At 7500 rpm, the flywheel stores about 600 J of energy. Assuming a 3 kW output power, and 20% thermal efficiency, the intermittent energy input with each cycle is about 120 J. Thus the speed and inertia of the components is controlled by the stored energy in the flywheel. This energy is converted to noise by other components, such as the fan.

When dealing with machinery that contains many separate noise sources, a convincing demonstration of the noise from each source can be obtained by source subtraction. In concept, source subtraction eliminates all but one noise source, allowing a measure of the

noise from that source. Measuring each source in turn allows the total noise to be estimated from the sum of the individual sources.

Implementation of source subtraction is often straightforward. Sometimes noisy components can be completely removed, or replaced by a quiet component. Sources that cannot be physically removed can be effectively eliminated by silencing treatments. Common methods of silencing are: ducting of aeroacoustic noise (such as exhaust noise), and lead wrapping or enclosure of noise radiating surfaces.

In practical cases it may not be possible or even desirable to quiet all sources at once. For example, enclosing or shielding many individual parts may also partially shield the noise source being measured. There is often little advantage to quieting sources with sound pressure levels more than 10 dB quieter than the source under study, as they cannot affect the total noise by more than 1 dB.

Selective source enhancement or reduction is used to determine the noise from individual sources when noise from other parts of the machine is present and it is not possible to use source subtraction. In this method, the absolute noise levels from a given source may not be known, but the exciting force can be changed by a known amount. The effect on the total noise can be predicted and measured, and thus, the contribution of the source can be determined.

Calculations comparing the expected change in one source with the measured change in total noise from the machine, give the absolute level of the individual source. The total noise can be measured easily, but the change in source strength can be more difficult to measure. A common method of changing source strengths in engine work is to change the engine speed. This has the advantage that the engine is operating within normal limits. Different sources have different speed dependencies. For example, fan noise commonly increases at about 50 dB per decade increase in speed. Thus a 1/3 octave speed increase will raise each 1/3 octave sound pressure level by 5 dB. Some engines have shown an 86 dB noise increase per decade increase in engine speed³, which has been attributed to bearing impacts. This is equivalent to an 8.6 dB change in sound pressure level per 1/3 octave change in speed.

For other sources, such as combustion, the rate of noise increase with speed is dependent on the slope of the forcing function spectrum. Before being radiated as noise, the forcing function is filtered by the structural response and radiation efficiency of the engine, which is unaffected by speed. An increase in engine speed compresses the time scale of the

source waveform. This effectively translates the spectrum toward higher frequencies. If, for example, the spectrum of the forcing function has a constant slope, a horizontal translation is equivalent to shifting the spectrum vertically. The amount of vertical translation is dependent on the slope of the spectrum, and the relative speed increase. In this manner a complicated spectrum may have a simple response to a change in speed. For a gasoline engine, the noise changes approximately 63 dB per decade (6.3 dB per 1/3 octave) increase in speed. This change approximates the decrease in cylinder pressure spectrum level with frequency.

An example illustrates the effect of changing the speed or repetition rate. Assume a force that varies like a square wave with a fundamental frequency at 20 Hz. From Fourier transform tables⁴ the unweighted 1/3 octave spectrum slope decreases 10 dB per decade increase in frequency. Decreasing the fundamental frequency by a factor of 10 shifts the unweighted spectrum to the left by a factor of 10. This shift effectively drops the source level, and hence, the radiated noise by 10 dB, as shown in figure 1.3. The response of a structure to the square wave is approximated by A-weighting the spectrum. The A-weighted spectra for equal amplitude square waves with fundamental frequencies of 20 Hz, 2 Hz, and 0.2 Hz are shown. A 10 dB level decrease for each frequency is apparent from the change in the A-weighted levels.

Changing the engine speed affects all sources at once. Changes can also be made so that only a single source is affected. One example of a source-specific change is to operate at different load conditions, keeping the speed constant. As the speed remains constant, a source, such as the fan, which depends only on speed, should not be affected. A change in load will increase or decrease the cylinder pressure levels, which will cause an identical change in radiated combustion noise (discussed in section 2.2 on combustion noise). The source strength of the cylinder pressure can then be measured with a pressure transducer directly in the cylinder⁵. The change in radiated noise also can be measured, and the combustion noise level calculated. Note the gas forces contribute to other sources, such as piston slap. Thus a change in load may have unwanted effects, if combustion is not the dominant source.

The impact magnitude for impact sources can be changed by increasing the separation between parts. This larger distance between parts allows a greater terminal velocity as components accelerate across the clearance space, increasing the energy of each impact. For large accelerations across short distances, the time to traverse the clearance space is relatively short so that other components are not affected. For example, an empirical

study of clearance effects in bearings showed no interaction between impacts in bearings⁶. Thus a small change in clearance space between parts only affects impact energy and hence the noise due to the impact. The effect on the impact strength can be calculated based on appropriate theory or experimental results.

The most effective means of detecting a source is to increase its noise output. The variability in sound generated by noise sources is estimated to be ± 2 dB for the sources considered in this thesis. A 10 dB reduction of an individual source can only be detected if the source originally accounted for over 40% of the total noise (-4 dB re total noise) of the machine. For comparison (with ± 2 dB uncertainty) a 10 dB increase in an individual source strength is detectable for a source that originally accounted for only 6% of the noise (-12 dB re total noise) from the machine.

A common form of data analysis obtains fast Fourier transforms (FFT) of time-windowed data. Although FFT analysis provides detailed narrow band spectra, transient events, and non stationary signals must be analyzed with caution⁷.

Coherence (appendix E) provides a measure of the linear dependence between two signals, allowing an estimate of the effectiveness of dual channel comparisons. A dual channel analyzer providing FFT analysis was available for this study (Brüel & Kjær type 2032). Dual channel analysis can provide a direct comparison of source and receiver, but may be of limited use for machinery noise. Linear comparisons require high coherence between a single source and receiver, and low coherence between separate sources. Although measurements of coherence can be low in an engine⁸, most sources are coherent due to the cyclic nature of operation. True coherence between channels is lost through dispersion and resonance in the structure.

There are a number of reasons why coherence may be reduced in machinery structure. Noise from machinery surfaces is primarily radiated by bending waves, which are dispersive⁹. The vibration energy propagates at different wave speeds⁹ for longitudinal, shear, torsion, and bending waves within the engine structure. As well there is reverberation, or ringing in the structure. All these factors help to make coherence low⁷, reducing the similarity between measurement channels.

Conversely, there are a number of reasons why coherence between sources will be increased on a saw, making the measurement channels appear more similar¹⁰. Combustion is the direct forcing function supplying energy to all other sources in the chain saw, which makes all linearly related sources coherent. All processes occur cyclically at the

crankshaft rotation rate, which also increases the coherence. Any two (non sinusoidal) periodic waveforms with the same period will be coherent at all harmonics. For example, piston slap impact vibration at the side of an engine cylinder was found to be coherent with combustion pressure below 2 kHz¹¹. These factors suggest it will be difficult to distinguish between sources using coherence, though if a single source dominates, the contribution from other sources can be reduced by using coherence¹².

In another application of digital signal processing, individual cylinder pressure time histories have been recovered from the acceleration of the engine surface using filtering techniques¹³. This requires knowledge of the magnitude as well as the phase between cylinder pressure and radiated sound.

The narrow frequency bandwidth provided by digital signal processing is useful in modal analysis, but not always in noise source identification. Due to the cyclic operation of the chain saw, the noise is harmonically related to the engine firing frequency, and is reduced at intervening frequencies. At higher frequencies, small changes in rpm cause the harmonics to smear out, until at some frequency they blend. For an ideal analyzer, a $\pm 2\%$ variation in rpm may allow only 25 harmonics to be visible. In practice, the harmonics may give little information other than the speed of the engine.

An exception is if another source operates at a non integer multiple of the firing rate. In this case a modulation of the signal results. From elementary trigonometry we find the sum of two sine waves¹⁴:

$$\sin at + \sin bt = 2 \sin \frac{(a+b)t}{2} \cos \frac{(a-b)t}{2} \quad 1$$

Thus the waveform amplitude is modulated as a function of time (t). If frequency b is a small fraction of a , the waveform is best described by the left side of the equation. Signal a drifts sinusoidally about the zero axis with the period of the modulation equal to that of signal b . The modulation of the signal is antisymmetric about the zero axis. When the two frequencies are similar, the right side of the equation best describes the waveform. The signal amplitude is modulated symmetrically about the zero axis, with the amplitude maxima occurring with period defined by the frequency difference $((a-b)t/2)$. This is the familiar phenomenon of "beats"¹⁵.

Thus a modulation of the time history indicates processes that occur at frequencies other than the engine firing rate. For example, chain saw rolling element bearings rotate at a

different rpm than the engine, and a modulation of the time history will be apparent if defects in the bearings significantly affect its vibration.

Another useful tool is the principle of reciprocity (see appendix A). Stated in brief, the ratio of source strength, and received pressure, is unchanged if the source and receiver position are switched. This allows added freedom in selection of source characteristics. The dynamic range of some measurements can also be increased using reciprocity. For example, poor results would be obtained with a tiny speaker inside a chain saw, and a pressure measurement in a noisy laboratory. Much better results would be obtained using reciprocity. A large powerful speaker could be located in the laboratory, and a small microphone placed inside the chain saw.

A good choice for the use of reciprocity is combustion noise. Combustion noise is linearly related to the cylinder pressure. The peak exciting force occurs at a well-defined position (top dead center), so measurements can be made with the chain saw crankshaft fixed in this position. This will eliminate other sound sources.

In this thesis, the nature of chain saw noise sources is described in chapter II. The available literature on the subject of chain saws is incomplete. Using experimental results from other comparable mechanisms and appropriate theory, a more complete description is proposed. This includes some new suggestions for the effect of piston slap side force. Next is a description of the apparatus required to measure contributions of the chain saw noise sources (chapter III). In this chapter, the vibration isolation of a pressure transducer allowing previously unobtainable cylinder pressure measurements is described. In addition, the design of a rubber and metal biodynamic model that successfully mimicked the vibrational response of the human hand arm system is also described. Chapter IV summarizes the experimental procedures used. The major sources were first determined when cutting wood at an outdoor field test site. Establishing the contributions to the mechanical noise of various components of the machine required laboratory experiments with the saw mounted on a dynamometer. Combustion noise is a major source on large reciprocating internal-combustion engines; however, the measurement of combustion noise in a chain saw required a novel use of the principle of reciprocity. Finally the results, discussion and conclusions (chapters V and VI) indicate the relative importance of each source and the implications for noise control. The appendices contain additional theoretical analysis (appendices A, and B), other analysis techniques used (appendix C), a discussion of the prospects for noise reduction (appendix D), explanation of equations used (E), reference quantities (appendix F), equipment details (appendix G), and related

publications (appendix H). Preliminary reports of this work have appeared elsewhere^{16,17,18,19}.

Chapter II Significant Noise Sources in Chain Saws

2.1. Exhaust and intake noise

Noise produced by the exhaust is normally quieted using an exhaust muffler. Both the exhaust and intake, radiate noise associated with combustion and unsteady gas flow within the engine²⁰. Although temperature and pressure differ in the exhaust and intake, control of intake noise is possible by using the same muffling techniques as for the exhaust.

The lowest frequency noise produced by the exhaust is due to the periodic gas emission with each engine cycle. Sound at higher frequencies results from the abrupt release of cylinder gases, due to piston porting of the cylinder^{20,21}. The most intense frequencies in exhaust noise have been modeled²² as due to a Helmholtz resonator effect controlled by the cylinder volume and exhaust outlet dimensions. The noise levels of muffled chain saw exhaust have been reported to be between 96 and 120 dB^{20,23,24,25} (A weighted) at the operator's ear.

The amount of noise reduction provided by an exhaust muffler is the choice of the saw manufacturer. Chain saw mufflers are restricted in terms of their length, volume, and complexity. Although exhaust noise can, in principle, be controlled to any acceptable level, output power reductions, or increases in muffler volume are tradeoffs associated with large reductions in exhaust noise. Increasing reductions in exhaust noise will not result in an equal reduction in the overall noise of the chain saw. Other sources will ultimately dominate and hence determine the overall noise.

A common device for reducing exhaust noise is a reactive muffler. The reactive muffler does not absorb acoustical energy: Impedance mismatches between muffler components reflect energy back to the cylinder, and change the impedance seen by the exhaust²⁶.

The action of a reactive muffler is seen by considering an energy balance between the cylinder and its surroundings. First assume an ideal gas and all the gas in the cylinder enters the surroundings. The change in internal energy (Δe) of the cylinder gas is related to the energy transferred to the surroundings by²⁷:

$$\Delta e = w + q_{out} + RT \quad 2$$

where w is the work per unit mass done on the surroundings, q_{out} is the heat transferred to the surroundings, R is the gas constant, and T is the final temperature of the cylinder gas.

For simplicity, consider a single event, the thermodynamic work (w) is given by²⁷:

$$w = \int_{v_i}^{v_f} p \, dv \quad 3$$

where p is pressure, v is the volume per unit mass of fluid, and the limits of integration are the initial and final specific volume.

Next, assume plane wave propagation through a control surface with area A , and change the integration of equation 3 from volume to time (t):

$$w = \int_0^{\tau} p \frac{A u}{m} \, dt \quad 4$$

where u is the particle velocity in the fluid, and m is the mass of the fluid originally contained in the cylinder.

Finally equation 4 can be solved to give the acoustical energy (E) in the frequency domain. Multiplication of equation 4 by the mass m , results in the acoustical energy (E) entering the surroundings through the control surface A ²⁸. Assuming the engine to be a volume velocity source, the pressure (p) can be expressed as $p = u z_a$, where z_a is the complex acoustic impedance, (the subscript is used to distinguish between acoustical and mechanical impedance, which differ by the dimension length squared). From equation 4, using Parseval's identity, the acoustical energy (E) expressed in the frequency domain for a single engine cycle is:

$$E = w m = 2 A \int_0^{\infty} \Re \{ |U|^2 Z_a \} \, df \quad 5$$

Where f is the frequency in Hz, $\Re \{ \}$ indicates the real part of a complex quantity, U and Z_a are the Fourier transforms of u and z_a . For any given frequency (f_o) in a bandwidth Δf the acoustic energy is:

$$E(f_o) = 2 A \Re \{ |U(f_o)|^2 Z_a(f_o) \} \Delta f \quad 6$$

Comparison of equations 2 through 6, allows an accounting for the energy flow from the cylinder to the surroundings. Without a muffler the engine exhaust sees the characteristic

impedance of the atmosphere (the product of air density ρ_a and sound speed c). To reduce the acoustical energy produced at a given frequency, the reactive muffler must reduce the acoustical resistance ($\Re\{Z_a\}$) seen by the engine exhaust below that of the atmosphere²⁶. This reduces the amount of acoustic energy flow that is generated at that frequency.

Note that if the engine was assumed to be a pressure source, the condition for reduction of acoustical energy is more obvious. The velocity (u) in equation 4 would be expressed as $u=p/z_a$. Generation of acoustical energy could then be stopped if the impedance at the exhaust port were infinite (i.e., seal the exhaust ports).

The simplest reactive mufflers act as low pass filters that have little effect on a steady gas flow²⁸, and progressively reduce the high frequency content of the exhaust. These mufflers only reduce the acoustical resistance ($\Re\{Z_a\}$) at higher frequencies, where the exhaust noise is unacceptably loud. At low frequencies, the exhaust noise will usually increase.

Controlled reduction of exhaust noise by changing the acoustical resistance at the exhaust port requires an accounting for each muffler element, at each frequency. The task is made simpler using transfer matrices (also called transmission matrix, two port or four pole parameter representation)²⁶. The transfer matrix approach provides a reasonable model of the muffler system with elements that can be combined to provide the desired muffler characteristics²⁶ (an example is shown in appendix A.1). Each muffler element is represented as having one inlet and one outlet, and is described by a 2×2 matrix. The state of the exhaust flow at the inlet or outlet of any given muffler element is described in the frequency domain by two state variables, acoustic pressure and mass velocity, which are assumed constant over a given cross section. A muffler element can be very complex, or as simple as a cylinder with inlet and outlet at opposite ends. Mathematical models are available for many elements, while the parameters of others can be measured to produce the required matrix for the element. Complex muffler elements consisting of many components can be built up by multiplication of individual element matrices.

Measurement of exhaust noise is straightforward if it is the most intense source. Then the noise can be measured directly and checked by ducting the exhaust away. Care must be taken that the ducting does not change engine operation, and hence engine noise. In cases where exhaust noise has been muffled below that of other sources, it can be identified by quieting other sources (source subtraction). A relative increase in exhaust noise levels

compared to those of other sources can be accomplished by lead wrapping the engine, or by using acoustic intensity²⁹. Alternately, the exhaust noise can be measured without a muffler and, provided the muffler characteristics are known, the muffled exhaust noise can be calculated (selective source enhancement and reduction). Errors can arise if the exhaust noise measurement is not made with the normal exhaust system termination, as any additional exhaust components will influence the frequency response of the engine and duct system. Thus, measurements of exhaust noise should be made without significant modifications to the exhaust.

Similar considerations apply to measurement and control of intake noise, although the reduced temperature and noise at the intake simplify both experiments and analysis.

2.2. Combustion

The energy released by combustion is the source of all the noise produced by the chain saw. However, combustion noise (by common definition) consists only of the noise linearly related to the cylinder pressure.

With this definition, the cylinder pressure spectrum can be used to infer the combustion noise. Typical cylinder pressure spectra are shown in figure 2.1. The pressure at the fundamental frequency is over 200 dB (re 2×10^{-5} Pa), with the measured higher frequency levels decreasing by up to 20 dB per decade of frequency change. This spectrum is attenuated by the engine structure before it is radiated as sound. The linear relationship between cylinder pressure and radiated combustion noise, represents the transmission loss (or structural attenuation) of the engine structure. Transmission loss is frequently controlled by the mass of the intervening structure³⁰, which leads to the sound pressure levels decreasing with increasing frequency by 6 dB per octave. As the cylinder pressure in figure 2.1 is seen to decrease at about 17 dB per octave, mass controlled transmission loss would result in a measured spectrum decreasing at 23 dB per octave. This is not the case with radiated combustion noise. The measured noise is broadband and relatively flat between 500 Hz and 3 kHz^{31,32}. This suggests that combustion noise is controlled more by the stiffness of the engine, which is inversely proportional to frequency. Thus, the radiated combustion noise spectrum is essentially flat, limited at low frequencies by the radiation efficiency of the engine structure, and at high frequencies by the cylinder pressure excitation.

A chain saw is smaller than other engines, and will produce less low frequency combustion noise. This is due to a reduction in the radiation efficiency of the structure, and a lack of

bending modes at lower frequencies. At a frequency inversely proportional to the characteristic dimension of the engine the radiation efficiency drops with decreasing frequency³³. For one 1.5 liter diesel engine the radiation efficiency was estimated to decrease at frequencies below about 400 Hz, which was slightly lower than that of the first resonance of the structure (about 650 Hz)³¹. A chain saw is smaller than a diesel engine, and thus the reduction in radiation efficiency would occur at proportionately higher frequencies (at about 630 Hz).

At most frequencies, a gasoline engine has lower cylinder pressure levels than a diesel, and should produce lower combustion noise levels. Figure 2.1 shows that for a gasoline engine the combustion noise excitation force is lower at all frequencies (especially higher frequencies). The radiation of noise at frequencies above about 400 Hz is determined by the rate of change of the pressure rise during ignition of the fuel. It has been reported that diesel engines produce significant combustion excitation upwards to 30 kHz³⁴.

In the past, studies of engines with significant combustion noise have led to empirical relations between engine parameters and combustion noise levels ($SPL(A)$ in A-weighted decibels re 2×10^{-5} Pa). One such relation based on years of experience with internal combustion engines is³⁵:

$$SPL(A) = p_r[1.0] + 10 f_\Delta \log(60n / 1000) + 50 \log(D_B) - Z_A \quad 7$$

The first term on the right hand side ($p_r[1.0]$) is related to the cylinder pressure source strength, with a value of 130 dB for gasoline engines. The value is up to 30 dB higher for diesel engines, due to a sharper cylinder pressure rise, and higher peak cylinder pressure (especially since gasoline engines are usually run at part load). The second term in equation 7 is based on the shape of the combustion pressure spectrum and relates the cylinder pressure slope at high frequency (f_Δ) to the engine speed (n) in revolutions per second, or rps. In gasoline engines the high frequency slope decreases at a higher rate than a diesel, and results in a value for f_Δ of 6.3 for gasoline engines and as low as 3.4 for some diesel engines. The next term uses the cylinder bore diameter (D_B in mm). It is based on the recognition that the forces on the piston and cylinder head are related to the diameter of the cylinder. The final term Z_A expresses the approximate attenuation provided by the structure. The value is 175.5 dB for gasoline engines and up to 10 dB less for diesel engines. The accuracy of equation 7 is about ± 2.5 dB for diesel engines, and ± 6.5 dB for gasoline engines.

Using equation 7 to predict levels for a chain saw produces interesting results. For lack of a better estimate, it was assumed the chain saw was most similar to the 4 stroke, water-cooled, automotive gasoline powered engines used to derive equation 7. An application of the formula to the pulpwood chain saw leads to a predicted level of 94.5 ± 6.5 dB (A-weighted) at 1 m, or about 97 dB (A-weighted) at the operator's ear (see appendix G.1.2 for pulpwood saw specifications). This level is surprisingly close to the mechanical noise levels of 94 and 95 dB (A-weighted) found in the literature for chain saws^{20,23}.

The magnitude of the noise level prediction from equation 7 suggests there would be no noise reduction if the chain saw engine was changed from the present 2 stroke air cooled engine to a very small automotive engine. This statement may seem difficult to reconcile with observations of automobiles. However, a chain saw typically operates under different operating conditions. Automotive engines are normally operated at low speed and part load (see fig. 2.1) as well as being partially enclosed, which significantly reduces the exciting pressure spectra, and radiated noise. A chain saw always operates at full load and high speed, which proportionately increases the noise.

Although it is relatively simple to identify and control sources external to the engine surfaces by ducting, or lead wrapping, measurement of combustion noise is complicated by the presence of other sources of engine noise that are radiated by the same surfaces. Complicating matters, the intermittent force of combustion interacts with the inertial forces to affect other noise producing components³⁶. This subject is discussed in detail in section 2.6 (on piston slap). Thus, noise related to crankshaft rotation, cannot easily be separated from the noise radiated from within the combustion chamber, as the surfaces that radiate the noise are the same in both cases.

Experiments on some engines have shown that the main path of combustion sound energy transmission is through the piston and crankshaft^{37,38}. In one case study³⁹, the force on the crankshaft caused deformation of the journal bearing bulkheads. Vibrations transmitted through the piston pass through a more complicated system than simple walls, making predictions difficult.

A convincing measurement of the noise of combustion has been reported by Priede³². In diesel engines the rapid rise in cylinder pressure produces high frequency pressure fluctuations that are transmitted through the structure and radiated as noise. To identify the contribution of combustion noise to the total noise (as a function of frequency), Priede adjusted the rate of increase of cylinder pressure during combustion (thus increasing the

magnitude of the high frequency pressure fluctuations) and monitored the radiated noise. To prevent other noise sources from being changed, engine rpm was maintained constant, but changes in the ignition timing and injection characteristics were used to alter the character of combustion and increase the high frequency noise in the cylinder. When the increase in cylinder pressure (at a given frequency) was proportional to the increase in the radiated noise (at that frequency), the radiated noise was assumed to be due to the noise in the cylinder. As already noted, the difference between the radiated noise and the cylinder pressure (as a function of frequency) is commonly called the structural attenuation. As the cylinder pressure is linearly related to the combustion noise¹², the structural attenuation depends on the construction and materials of the engine structure⁴⁰. Using structural attenuation and cylinder pressure, combustion noise can be calculated under any operating conditions.

Direct measurement of combustion noise is difficult as this requires a comparison of the forcing function in the cylinder with the radiated noise. A pressure transducer mounted within the combustion chamber will be exposed to flame, high temperature, and high pressure. It must also be able to measure a full range of audio frequencies. This is much more difficult than measurement of the sound field external to the engine, which can be measured using standard techniques.

An alternative measurement, is to use a non functioning engine and provide an alternate source of pressure, or vibration. This has an added advantage for situations where the vibration and sound are propagated through many paths. Limiting the excitation to the piston, or cylinder wall allows an estimate of the relative importance of each component. Note that reciprocity (appendix A) can be used for mechanical as well as acoustical systems, allowing more flexibility in positioning of source and receiver.

One example of combustion noise measurement with a non functioning engine, was obtained by the Institute of Sound and Vibration Research in Southampton England (ISVR). They constructed a "banger rig" to allow excitation of a stationary engine^{41,38}. This eliminated additional noise sources normally present in a running engine. The crankshaft was held stationary and an explosion was created in the combustion chamber using propane gas ducted into the cylinder through an injector hole. The resulting engine excitation and noise were then measured.

2.3. Fan noise

The cooling airflow over the chain saw cylinder is provided by a centrifugal fan with forward curved blades, driven at high speed directly from the crankshaft (fig. 1.1, 3.15). In general, the noise from this type of fan is broadband, and diminishes with increasing frequency. The broadband component is due to the formation of turbulent eddies⁴². Noise at discrete tones is related to the blade passage frequency.

The noise of similar fans of this type can vary significantly according to the installation⁴². Because this type of fan is commonly used for cooling small machines, minimum size may be more important than minimum noise and optimal airflow. The relative clearance between parts can also be critical⁴³.

The volute that collects the discharge air from the fan can affect the noise. The volute is a duct around the outside of the fan that collects the air from the fan blades and directs it to the cylinder fins. The duct size corresponds to the volume of air it carries, thus it is initially narrow, and progressively widens. The shape approximates a spiral with the fan at the center. The narrowest point in the volute duct is called the cutoff. Eck indicates that the cutoff can produce a howl if it is too close to the impeller⁴⁴. On the pulpwood saw, the narrow section of the volute extends for about 125 degrees of arc, at a distance of 3% of the fan diameter. This distance may be somewhat low, as Jorgensen suggests this distance should be 5% to 20% of the impeller diameter⁴³.

The volute directly affects the amount of air discharged from the fan. On the pulpwood saw, for approximately 160 degrees of arc after the cutoff, the volute side wall spirals away from the fan at angle of 6.6° . The remainder is open to the cylinder fins and has no side wall. In the latter open section, streamlines from the fan could extend radially from the fan, the streamlines in the former spiraling section are limited by the wall angle.

Using known properties of the pulpwood saw volute, estimates can be made for the flow through the fan, and hence the effect on noise. For example, in the narrowest section of the volute, due to the closeness of the volute wall to the fan, the flow would be essentially zero, except for leakage around the fan, and ignition module. This would stall the fan blades producing turbulence. Additional turbulence due to the presence of the irregularly shaped ignition module would also produce noise.

In the next section of the volute, for about 160 degrees of arc, the volute wall has an angle of inclination to the fan periphery of 6.6° . The flow through the fan blades can be predicted, if the blade exit streamlines are assumed parallel to the volute wall. This is

shown in figure 2.2, which has been drawn with scale appropriate for the pulpwood saw (see also fig. 3.15). The blade exit angle (β_2), blade tip speed (U_2), and the streamline angle ($\alpha_2=6.6^\circ$) are known. Vector addition allows calculation of the flow velocity (V_2) out of the fan:

$$V_2 = \frac{U_2}{\cos \alpha_2 \left(1 + \frac{\tan \alpha_2}{\tan \beta_2} \right)} \quad 8$$

The resulting flow velocity (V_2) out of the fan blades on the pulpwood saw is 48 ms^{-1} . Assuming incompressible flow, the volume flow (Q) through a fan with outer diameter (D_2) and an infinite number of blades of length (L_b , measured in direction normal to blade chord) is given by:

$$Q = \pi D_2 L_b V_2 \sin \alpha_2 \quad 9$$

Using equation 9, the calculated volume flow for the pulpwood saw is $0.06 \text{ m}^3 \text{ s}^{-1}$. However, the equation is only applicable for the 45% of the volute with a 6.6° side wall angle (equal to α_2). Allowing for the reduced area of the volute affected, the local volume flow would be about $0.03 \text{ m}^3 \text{ s}^{-1}$.

The remaining 15% of the volute, has no side wall. Thus the only the momentum of the flow around the fan would prevent the streamlines from becoming radial. As the streamlines can exit at a greater angle, the volume flow through the blades should be greater through this section. This will compensate for the restricted flow in the section where the volute wall blocks the fan.

For simplicity in calculation, the entire volute will be considered to have a side wall angle of 6.6° . The volume flow through the section with no side wall is assumed to compensate for the section where the flow is blocked.

In an incompressible flow the inlet volume flow equals that of the outlet. Since the tangential velocity of the inner edge of the fan blade (U_1) and blade inlet angle (β_1) are known, the inlet streamline angle (α_1) is:

$$\alpha_1 = \text{atan} \left(\frac{Q \tan \beta_1}{\pi D_1 L_b U_1 \tan \beta_1 - Q} \right) \quad 10$$

where D_1 is the fan blade inlet diameter.

Substituting values appropriate for the pulpwood saw (appendix G.1.2.1) the resulting inlet streamline angle is 16° . The common approximation is an inlet streamline angle of 90° although the true value will be lower⁴⁴. As seen in figure 3.15 the fan blades are aligned essentially radially. Thus the approximate angle of attack of the fan blades is 74° ($=90^\circ - 16^\circ$). Centrifugal fans are known to reduce the size of the boundary layer compared to a single airfoil, thus the boundary layer will not be as thick as a single airfoil. But, due to the angle of attack at the blade inlet, there will be significant turbulence, and a large boundary layer.

Assuming an average streamline angle of 6.6° at the fan blade outlet, the total pressure head (H) can be calculated from⁴⁴:

$$H = \frac{U_2^2}{g \left(1 + \frac{\tan \alpha_2}{\tan \beta_2} \right)} \quad 11$$

where g is the acceleration due to gravity.

The value of the total head H calculated for the pulpwood saw is 212 m. The total pressure is 2.5 kPa, obtained from $p_t = H\rho g$, (ρ is air density).

Comparison with other fans can be obtained by dimensional analysis of fans with geometric, kinematic and dynamic similarity. In this way a number of generalizations, or fan laws have been developed relating the performance of fans with geometric similarity in the air flow. Two common non-dimensional quantities are the specific speed (Ω), and specific diameter (D_{sp}) of the fan. These are given by⁴⁵:

$$\Omega = \frac{2\pi n \sqrt{Q}}{(gH)^{1/4}} \quad 12$$

where n is the fan speed in rps, and:

$$D_{sp} = \frac{D_2 (gH)^{1/4}}{\sqrt{Q}} \quad 13$$

The specific speed and diameter of the pulpwood saw are typical for a centrifugal fan. The specific speed of the pulpwood saw at 7500 rpm is 0.62, which is appropriate for a centrifugal fan⁴⁵. The specific diameter of 3.1 is comparable with other forward curved fans operating at the same specific speed⁴⁶.

The efficiency of comparable forward curved fans is 70% to 75%, about 10% lower than similar backward curved blade fans⁴⁶. A fan operating at maximum efficiency, also produces near minimum noise^{43,44}, with sound power levels increasing approximately 4 or 5 dB for each 10% reduction in efficiency⁴³.

Dimensional analysis also allows an estimate of fan noise. Using the fan law that depends on volume flow (Q) and total pressure (p)⁴³:

$$PL_a = PL_b + 10 \log \left(\frac{Q_a}{Q_b} \right) + 20 \log \left(\frac{p_a}{p_b} \right) \quad 14$$

where PL is the acoustical power in dB re 10^{-12} W, the subscript b refers to the fan with known conditions, and subscript a refers to the fan with unknown noise levels.

Equation 14 can be used to predict the noise at the operator's ear. An example of the noise produced by a typical forward curved blade centrifugal fan is given in Harris⁴². Values are given for a well-designed fan operating at volume flow rate (Q_b) of $1 \text{ m}^3\text{s}^{-1}$, and total pressure (p_b) of 1 kPa. The 1/3 octave band power levels (PL_b) for such a fan start around 94 dB re 10^{-12} W at 63 Hz, and decrease steadily at about 4.7 dB per octave increase in frequency. Using a calculated Q_a of $0.06 \text{ m}^3\text{s}^{-1}$, and p_a of 2.5 kPa, the calculated pulpwood saw fan noise is 81 dB (A-weighted) at the operator's ear (from eqn. 14). This sound level is comparable to that found in an experimental study of another chain saw (87 dB A-weighted²⁴).

As already noted, fan noise is dependent on characteristics of individual installations. Thus a measurement of fan noise under one condition on a chain saw could be used with fan laws to improve the noise predicted under other engine operating conditions (on the same chain saw). For example, the effect of speed (n) on fan noise can be determined directly from the following relationship for sound power level (PL)^{42,43}:

$$PL_a = PL_b + 70 \log \left(\frac{D_{2a}}{D_{2b}} \right) + 50 \log \left(\frac{n_a}{n_b} \right) + 20 \log \left(\frac{p_a}{p_b} \right) \quad 15$$

where the subscript *b* refers to the fan with known conditions, and subscript *a* refers to the fan with unknown noise levels, other symbols as defined previously.

This simply states that for a given rotor diameter, the fan noise increases at 50 dB per decade increase in speed. Jorgensen, who derived this relationship, indicates that other authors show variations of about 20% in the parameters of equation 15⁴³. Thus the difference between predicted and actual levels could be up to $\pm 20\%$ of the predicted levels. He also recommends that use of the equation be limited to overall sound produced by fan. However, equation 14, which is based on the same principles as equation 15, is used without any caveats in Harris⁴².

Deviations from fan laws are expected⁴³ below Reynolds numbers of 8×10^5 . The Reynolds number (*Re*) is defined as⁴⁷:

$$Re = \frac{VL\rho}{\mu} \quad 16$$

where *V* is relative the velocity between fluid and body, *L* is a characteristic dimension of the body, μ is the coefficient of viscosity of the fluid and ρ is the mass density of the fluid.

On the pulpwood saw, the Reynolds number was calculated using *L* equal to fan diameter (*D*₂), *V* equal to blade tip velocity (*U*₂), and fluid conditions at the inlet⁴³. The Reynolds number was 4×10^5 , thus some deviation is expected from the fan laws, with the difference increasing as speed is reduced.

One identifying characteristic of fan noise is the presence of tones at harmonics of the blade passage frequency. For the pulpwood saw, handbook spectra suggest the 1/3 octave-band level at the blade passage frequency is increased above broadband noise by about 4.5 dB⁴². Thus, in the presence of other noise sources, the blade passage frequency will not be apparent unless the fan noise is a dominant source.

There are in effect, 3 differently shaped fan blades on the pulpwood saw fan: 12 narrow blades, 12 wide blades, and 2 large protrusions on the fan (see fig. 3.15) resulting in 3 blade passage frequencies. Fan noise will be radiated with increased levels at the 3 blade passage frequencies and their harmonics. When the fan back plate rotates at 125 Hz, the fundamental blade passage frequencies are 250 Hz, 1500 Hz and 3 kHz.

The broadband noise produced by an improperly designed fan can result from von-Kármán vortices shedding off the blades⁴⁸. These vortices are produced when the blade leading

edge angle does not match the air incidence angle. They produce aeolian tones that occur⁴⁷ for flow Reynolds numbers between 40 and 4×10^5 . As the Reynolds number approaches 4×10^5 the flow in the wake of the body becomes turbulent and the aeolian tones change to broadband noise⁴⁹. For the pulpwood saw, the fan Reynolds number based on blade tip velocity (U_2) and blade chord (L_c) is about 7×10^4 (note this is lower than a Reynolds number used for comparison of different fans, that is based on tip velocity and fan diameter⁴³). This indicates broadband aeolian noise can occur. The total broadband noise intensity (I) is⁴⁹:

$$I = \frac{\kappa Sr^2 L^2 \rho_o V^6 \sin^2 \theta \cos^2 \phi}{32 c^3 R_{obs}^2 (1 - M \cos \theta)^4} \quad 17$$

where the characteristic dimension of the flow obstruction is L ; the constant κ is taken as 1⁴⁹; ρ_o and c are the inlet air density and speed of sound respectively; the flow velocity is V ; M is the flow Mach number; (R_{obs}, θ, ϕ) is a vector in spherical coordinates ending at the observer position, with origin at the fan hub, and $\theta=0$ along the fan axis; the Strouhal number Sr is given by:

$$Sr = \frac{fL}{V} \quad 18$$

To solve equation 17 or 18 for a single blade on the pulpwood saw, the characteristic dimension of the flow obstruction L is determined by the boundary layer thickness⁴⁹ at the blade exit, say 1/3 the blade chord (L_c) (see figure 3.15 for relative sizes); the flow velocity V was taken to be the blade tip speed (U_2); R_{obs} was the distance to the operator's ear position; θ and ϕ were taken respectively as $\pi/2$ and 0 to produce the maximum value for the equation. The Strouhal number associated with aeolian tones⁴⁹ is 0.2.

As seen in appendix G.1.2.3, there are two sizes of blades on the pulpwood saw fan. Twelve blades have a constant chord, the remaining 12 have a varying chord. Using the approximations above and a Strouhal number of 0.2 (eqn. 18), the main frequencies affected by fan noise should be at 1.5 kHz and extend up to 2.6 kHz. Using a boundary layer thickness of 1/3 the larger blade chord and multiplying the resulting intensity by 24 (for 24 blades) results in a maximum of 116 dB (A-weighted) at the operator's ear. This is only 5 dB below the previous calculation based on fan laws using similar fans.

Along with the approximations made, the noise levels will be affected by factors that are difficult to calculate. Due to the θ and ϕ dependency, the intensity at an arbitrary location would be a few dB lower (ideally there exists a plane within which the intensity would be zero). This will be offset by increased noise levels from the section of the volute that is open. The largest velocity in the fan will be through this section, and equation 17 is strongly affected by velocity. For example (as assumed earlier) if 4 fan blades produced triple the average air velocity, they would produce a sound pressure level of 97 dB.

Allowing for the approximations made, the intensity that could be expected from aeolian tones is comparable to that found on another chain saw (87 dB A-weighted)²⁴. The aeolian tone noise varies as the 6 th power of fan speed, which would change the third term in equation 15 to $60\log(n_d/n_b)$. This value has been used by some authors in the fan law for sound prediction⁴³.

Most of the 24 fan blades can be completely removed to reduce fan noise and determine its effect on the noise of the pulpwood saw. However, the magneto and counterweight produce protuberances in the back plate that are similar to fan blades (fig. 3.15). Thus removing the fan blades will leave two lobes on the back plate, reducing the number of blades from 24 to 2. Assuming equal contributions to noise energy are produced by each blade or lobe, reducing the number of blades to 2 would reduce the expected noise radiated by 11 dB. Measurement of fan noise with the blades removed should indicate if the fan is a dominant source. Yet, if the fan is comparable to other sources, a reduction of 11 dB will not have much effect on measured noise.

The fan noise is mainly dependent on its rotational speed. Thus a better method of measuring fan noise would be to drive the fan with an external motor.

2.4. Rigid body motion

The rigid body motion of a chain saw radiates noise, and is therefore a noise source. Most chain saws have only a single cylinder and thus a simple counter weight on the crankshaft cannot completely balance the reciprocating mass of the piston and connecting rod⁴⁶. For example, if a rotating mass was used to balance the reciprocating mass along the cylinder axis, an unbalance of equal magnitude would be introduced normal to the cylinder axis.

Noise produced by shaking motion at the engine rotation rate will be strongly reduced by the radiation efficiency of the chain saw. Thus harmonics of the rotation rate will produce the most audible noise. Due to the angular displacement of the connecting rod (commonly

modeled as a slider-crank mechanism) additional shaking forces exist at harmonics of the rotation rate, with the main component existing at the first harmonic frequency. For example, the large-timber chain saw operates most efficiently at 9600 rpm. Thus the second harmonic is at 315 Hz, at which frequency the sound radiation efficiency (based on saw dimensions) starts to become significant⁵⁰.

Rigid body motion has been studied in connection with operator exposure to hand arm vibration. The fluctuating forces acting on the chain saw at low frequencies are mostly due to the reciprocating piston, connecting rod, and crankshaft^{51,52}. To evaluate the vibration exposure of the operator's hands, ISO 7505⁵³ emphasizes the lowest frequencies of saw vibration, and thus a number of attempts have been made to model the low frequency vibration. The most significant difficulty is in modeling the mass distribution of the chain saw, as the forces from the slider crank mechanism are well understood, and the chain saw is essentially a rigid body at low frequencies. The results of these studies tended to underestimate vibration on average by 6 dB. The differences were attributed to deformation of the structure at frequencies below the lowest assumed vibrational mode, and improper modeling of the support provided by the human operator^{51,52}.

In this thesis, a simple model is developed for the shaking forces experienced by the powerhead, which is vibration isolated from the handles. A number of simplifications are made. The powerhead is assumed to float freely in space, unaffected by handle mass, or vibration isolators. All forces are considered to act through the center of mass of the powerhead, and the piston and connecting rod masses are lumped together ($=m_p+am_c$, and a is typically between 1/3 and 2/3). Then the force due to the reciprocating motion of the piston and connecting rod (fig. 2.3) is⁵⁴:

$$F \approx (2\pi n)^2 (m_p + am_c) R_s \left(\cos \theta_s + \frac{R_s}{L_c} \cos 2\theta_s \right) \quad 19$$

where n is the angular velocity of the crankshaft in rps, θ_s is the instantaneous position of the crankshaft in radians, R_s is the radius from crankshaft axis to the big end of connecting rod in meters, and L_c is the length of the connecting rod in meters. The resulting force F is in newtons.

Using values appropriate for the pulpwood saw (appendix G.1.2), 50% balance of the reciprocating mass, and an assumed powerhead mass of 6 kg (excluding handle mass), the powerhead vibration is calculated to be about 165 dB at the fundamental and 164 dB at

the first harmonic. If dipole radiation is assumed, and the radiation efficiency increases asymptotically at 40 dB per decade up to unity at 630 Hz⁵⁰, then the levels at 0.75 m distance, in the direction of maximum sound radiation are 90 and 95 dB for the fundamental and first harmonic. Calculations of this type are discussed in more detail in the next section.

Measurement of sound radiation from rigid body vibration can vary significantly depending on the position of the microphone and the presence of reflecting surfaces. The assumption of dipole radiation implies there exists a plane in which no sound is radiated. The best accuracy would be obtained by measuring under actual conditions of operation.

2.5. Impact noise

The remaining noise sources on the chain saw produce noise through vibration of the external surfaces of the engine. An experimental study using acoustic intensity showed the level of panel vibration noise is comparable to that of the other significant sources on the chain saw²⁹, (such as the exhaust, intake, and fan). The forces that produce this vibration can be modeled as the result of impacts. The following discussion is sufficiently general that it may be applied to other sources, such as combustion noise, (with a force pulse similar to that of an impact³¹), or even rigid body motion.

2.5.1. Production of impact noise

Vibration normal to the surface of the exterior panels of the saw produces noise. For this reason, sound is radiated most effectively by bending waves. The efficiency of the noise production depends on panel damping, shape, and density of the bending wave modes⁹. At low frequencies where no flexural panel modes are present, the sound is radiated by rigid body modes, which radiate very poorly. As the frequency is increased the number of flexural modes increases, and the radiation efficiency also increases. The sound power ($P(f)$) as a function of frequency radiated by the modes is defined by⁹:

$$P(f) = AV^2(f)\rho_c c\sigma(f) \quad 20$$

Thus the sound power is proportional to the surface area (A), the radiation efficiency ($\sigma(f)$), the time and space averaged mean square of the normal component of surface velocity ($V^2(f)$), and the characteristic impedance of the air ($\rho_c c$). The radiation efficiency of the surface ($\sigma(f)$) is roughly dependent on surface area⁵⁰: it is small at low frequency, and increases asymptotically to unity with increasing frequency.

The surface velocity produced by an impact is due to a translation of the surface, and excitation of vibrational modes. In a series of articles on impact noise Richards has referred to the initial translation of the structure as acceleration noise, the noise following was termed ringing noise. At frequencies where flexural modes are present, the ringing noise energy produced by an impact can be 20 to 30 dB more intense than the noise due to the initial acceleration⁵⁰. Thus the acceleration noise can be ignored in favor of the ringing noise.

The acoustical power radiated by ringing noise is a fraction of the vibration power within the structure. The remainder of the vibration power is converted to heat due to damping within the structure. The sound power contribution from ringing noise in a bandwidth Δf at frequency f_o is given by^{34,55}:

$$P(f_o)_{\Delta f} \approx P_v(f_o)_{\Delta f} \frac{\rho_o c \sigma(f_o)}{\rho_o c \sigma(f_o) + \rho_m h \eta 2\pi f_o} \quad 21$$

where P_v is the vibratory power h is the material thickness; η is the structural damping factor; and ρ_m is the mass density of the material in the structure. The bracketed quantity, f_o indicates a value at a specific frequency, and subscript Δf indicates measurement spans a finite frequency band.

From this equation an estimate can be obtained for the radiated noise spectrum. The impact energy P_i is directly related to the radiated noise energy P . The impact energy is broadband with the high frequency limit determined by the duration of the impact. Low frequencies are reduced due to the radiation efficiency (σ) of the source (in the numerator of eqn. 21). Assuming the chain saw acts as an oscillating sphere (i.e., a dipole)⁵⁰, the radiation efficiency (σ) varies as f^4 at low frequency.

As could be expected, if there is no energy dissipation in the structure ($\eta=0$), equation 21 predicts all the impact energy would be radiated as noise. However, η is not normally zero, and the term $\rho_m h \eta 2\pi f_o$ usually dominates the denominator at audible frequencies³⁴. Thus there is reduction in sound power of 10 dB per decade increase in frequency, due to the frequency dependence of equation 21 (f_o term in the denominator).

The structural damping loss factor η is approximately constant with frequency, with values for bolted structures ranging between 0.01^{34,54} to as much as a few tenths of a percentage (if there is relative sliding between machine parts⁵⁶). Although no

measurement of structural damping is available for the pulpwood saw, one measurement³¹ on a diesel engine found a loss factor for the bare engine frame to be 0.01. When the engine was completely assembled the structural damping loss factor increased to 0.1 below 2500 Hz. Aluminum and magnesium have a higher loss factor than steel, but, as the material loss factor is small compared to the structural loss factor, a reasonable estimate for η would be 0.05, similar to a steel engine.

From the above considerations, the impact noise increases at low frequencies, levels off, then eventually drops off due to the frequency response of the original impact. A chain saw is smaller than most other engines, so the radiation efficiency will be smaller at low frequencies. Whereas a diesel engine may radiate noise down to 200 Hz, the lower frequency limit of the chain saw radiation efficiency is about 630 Hz (due its smaller size). A chain saw is hence unlikely to produce much noise below 500 Hz.

The vibratory power must be known to calculate the acoustical power using equation 21. The total mechanical vibratory power can be obtained from consideration of the magnitude of the energy input to the system. This is easier than calculation of the associated spectrum. The vibratory power P_v , integrated over all frequencies is defined by^{34,55}:

$$P_v = 2\pi \int_0^\infty \Re\left\{\frac{1}{Z}\right\} |F_{cycle}|^2 \partial f \quad 22$$

The frequency spectrum of the vibration can be measured, or approximated. The vibration spectrum is determined by the functions $\Re\{1/Z\}$ and $|F_{cycle}|^2$. Z is the impedance of the structure at the impact point as a function of frequency. F_{cycle} is the spectrum of the force versus time history for a single engine cycle, and represents the spectrum of the force of the individual impacts occurring with every cycle. The assumption (also made previously) is that each impact is independent and the energies are additive.

Low frequencies associated with an impact force have little effect on the radiated noise. Due to the reduced radiation efficiency, and sparsity of radiating vibrational modes at low frequencies, higher frequencies of the force spectra contribute most to the radiated noise. For this reason, Richards suggests using a time derivative of the force spectrum (i.e., $F' = Fi2\pi f_o$) to provide an intuitive picture of the frequencies important to the generation

of impact noise. Expressing equation 22 in terms of vibratory power in a small bandwidth Δf at frequency f_o results in:

$$P_v(f_o) = 2\pi \Re \left\{ \frac{1}{Z(f_o)} \right\} \frac{|F'(f_o)|^2}{4\pi^2 f_o^2} \Delta f \quad 23$$

2.5.2. Structural response of the engine

Assuming the total vibratory power (P_v) is known, the associated frequency response requires an estimate of the forcing function and the structural response (or impedance) of the engine. The structural response will be considered first. Note that since the application of a force can produce translation as well as rotation, the impedance will be a combination of the driving point and moment impedance.

First, a simple system will be considered. If the chain saw is a single degree of freedom system, the displacement (x) is given by⁵⁷:

$$x = \frac{-iF}{2\pi f Z} \quad 24$$

where the impedance Z is:

$$Z = C + i \left(2\pi f m - \frac{K}{2\pi f} \right) \quad 25$$

where K is the mounting stiffness, C is the damping, and m is the chain saw mass.

The power (P_v) absorbed by the system depends on the damping (C) and is related to the applied force (F) and structural impedance (Z). The power (P_v) expressed for a sinusoidal force (F) is⁵⁵:

$$P_v = \frac{1}{2} |F|^2 \Re \left\{ \frac{1}{Z} \right\} \quad 26$$

Upon expansion of equation 26 it is apparent³⁴ the vibrational power in the system is controlled in the low frequency limit by the stiffness of the chain saw support, varying as $C(2\pi f/K)^2$. At the high frequency limit $\Re\{1/Z\}$ (the real part of the inverse of the impedance) will be controlled by mass, varying as $C/(2\pi f m)^2$. While the presence of

damping increases the energy absorbed by the system away from resonance, at resonance, damping decreases the absorbed power.

A single degree of freedom system analogy is most applicable at lower frequencies, where the vibration isolators control the stiffness of support, and the saw is approximated as a rigid body. The resonant frequency of the vibration isolators is set below the engine firing frequency. Thus at, and below the resonant frequency of the rigid body motion the forcing function will be negligible. The single degree of freedom vibrational power response of the saw above the firing frequency will vary as f^{-2} .

At frequencies where the saw is no longer rigid, a single degree of freedom system will not be an appropriate model. The lowest structural resonance of the pulpwood saw will be due to bending modes, but, will not be well defined due to the complexity of the structure. Assuming uniform thickness (h) and bending stiffness (B) in the chain saw an estimate for the lowest bending mode can be made based on the chain saw dimensions, and bending wave speed. The bending wave speed (c_b) is given by⁵⁵:

$$c_b = \sqrt[4]{\frac{(2\pi f)^2 B}{\rho_m h}} \quad 27$$

where the bending stiffness B is:

$$B = \frac{Y}{1-s^2} \frac{h^3}{12} \quad 28$$

where s is Poison's ratio, and Y is Young's modulus.

For the pulpwood saw, assume a behavior similar to a plate with typical linear dimension of 12 cm, and thickness 5 mm (see fig. 3.11). Support conditions differ on different sections of the chain saw. From standard tables for beam bending, the first natural frequencies for various support conditions are⁵⁸:

free-free	1800 Hz
clamped-clamped	1800 Hz
simply supported	800 Hz
cantilever	290 Hz
cantilever (6 cm length)	1100 Hz
clamped-pinned	1250 Hz

From the above values, the lowest frequency where any flexural response is expected from the saw would be 290 Hz. At approximately 1.25 kHz large areas of the saw could be expected to become excited. This frequency should describe the general response of the saw, even though the saw may not be completely rigid at lower frequencies^{51,52}, due to localized masses and stiffnesses.

The response of the structure at frequencies above the first bending mode resonance (say 1250 Hz) is the most important, since bending modes radiate noise most effectively. If the impact time is relatively short, (or the size of the impact area is relatively small⁵⁵) the system boundaries will be less important and the equation for impedance of a comparable infinite structure will have some applicability. At lower frequencies, the structural response of a finite system differs from an infinite system as reflections from the boundaries influence the impedance seen by a continuous force excitation. These reflections at the system boundaries produce discrete resonances (or vibrational modes).

A finite homogeneous system with uniform thickness can be approximated by an infinite system. The impedance term $\Re\{1/Z\}$ that controls the power absorbed (eqn. 26) may be approximated for an arbitrary frequency band Δf by^{55,9}:

$$\Re\left\{\frac{1}{Z(f_o)}\right\} = \frac{\pi}{2m} \frac{\Delta N}{2\pi\Delta f} \quad 29$$

where N is the number of vibrational modes present, ΔN is the number of modes in frequency band Δf , centered at frequency f_o .

The term $\Delta N/\Delta f$ is the modal density of the system, which for simple structures such as beams, plates cylindrical shells and rings varies proportionally to $\Re\{1/Z\}$ of a similar infinite structure. In a finite structure, $\Re\{1/Z\}$ depends only on the mass and modal density of the structure. The response of a finite system corresponds to that of an infinite number of independent single degree of freedom systems with the total power absorbed dependent only on the number of modes present⁵⁵. For a finite excitation area the spatial variation of $\Re\{1/Z\}$ also depends on the number of modes excited, but, there is no spatial dependence with ideal point excitation⁵⁵.

Equation 29 can be applied to the pulpwood saw, as it is constructed of aluminum and magnesium, metals with similar mechanical properties, and the thickness of the casting is approximately uniform (see fig. 3.11). The power absorbed by the system only requires an

estimate of the real part of the structural impedance (eqn. 26). The term $\Re\{1/Z\}$ for infinite homogeneous plates, beams, and cylindrical shells is found to vary between f^{-1} to f^1 ⁵⁵. For example, the point impedance of a thick plate is influenced more by mass than stiffness, with the absorbed power varying as f^{-1} (and the inverse of the mass per unit area). The moment impedance for a thin plate is most influenced by stiffness, with the absorbed power varying as f^1 (and the inverse of the plate stiffness). As could be expected from an infinite system, many frequencies are simultaneously affected, and the change in impedance with frequency is not as great as for a single degree of freedom system.

Thus based on the variation with frequency of the simple structures mentioned above it is reasonable to expect $\Re\{1/Z\}$ to be relatively constant with frequency (f^0) above the lowest structural resonance of the pulpwood saw (say 1200 Hz). At lower frequencies $\Re\{1/Z\}$ should vary as f^{-2} . Although only the shape of the frequency response is required in this analysis, an estimate⁵⁹ for the engine impedance (Z) is the point impedance of a simple thin isotropic plate⁵⁵:

$$Z = 8\sqrt{B\rho_m h} \quad 30$$

where the flexural stiffness of the plate (B) is given by equation 28.

Substituting values appropriate for the pulpwood saw (appendix G.1.2) and assuming the material is aluminum, the bending stiffness from equation 28 is 800 Nm, and Z is 800 kg s^{-1} .

The impedance for a plate (eqn. 30) can be compared with that of a single degree of freedom system (eqn. 25) at 125 Hz. Using values appropriate for the pulpwood saw, an assumed isolator resonance of 60 Hz (very conservative), and critical damping of the isolators (very conservative), the vibratory power from equation 26 is approximately 10 times higher in the plate. A less conservative estimate would increase this value to a factor of 100. Thus the value of $\Re\{1/Z\}$ at 125 Hz should be about 20 dB below the value at 1250 Hz. Between the lowest structural resonance and vibration isolator frequency, it is likely there will be some response of the saw to vibration, especially since the lowest resonance frequency is not well defined⁵¹. A conservative estimate for $\Re\{1/Z\}$ of f^2 would assume a larger response to vibration, and would not significantly affect the radiated noise (due to the reduced radiation efficiency at low frequency). This provides a conservative upper bound for weak localized vibrational modes in the structure, as well as

the acceleration noise, which is lower in amplitude and frequency than the ringing noise⁵⁰. Above 1250 Hz $\Re(1/Z)$ should be relatively constant with frequency (f^0).

2.5.3. Force spectrum

At this point an estimate has been obtained for all parameters affecting the radiated noise except the impact force spectrum. Equations 23 for vibratory power or 21 for acoustical power can be viewed as filters or frequency weightings affecting the force spectrum.

The vibratory power (eqn. 22) is dependent on the impulse force spectrum as well as the real part of the impedance presented by the structure. The impulse force spectrum is related to the imaginary, or spring like, part of the structural impedance, and the mass of the impacting body. An estimate of the imaginary part of the structural impedance, as well as a geometric, and dynamic model of the impact is required.

The exact form of the impact time history is unimportant to the present analysis, but an estimate of the spectrum is required. Typically, an individual impact time history can be pictured as a half cycle of a sine wave, with zero amplitude at the time of impact^{34,60} (thus if impact occurs at time equal zero, the impact would appear as one half cycle of a cosine wave). The Fourier transform spectrum (F) of such a force is flat at low frequencies, and begins to decrease as f^{-1} above the sine wave frequency. At approximately double the sine wave frequency, the spectrum decreases as f^{-2} . Similar frequency dependence is also observed if the original force behaves like a damped sine wave as it approaches zero force. Thus the force derivative spectrum, F' will increase as f^1 at low frequencies, becomes flat at the sine wave frequency, and at approximately double the sine wave frequency, the spectrum decreases as f^{-1} .

At this point all parameters required for a general estimate of the pulpwood saw response to an impact have been approximated. The next three sections deal with application of this information to piston slap, bearing noise, and chain impacts.

2.6. Piston slap

Piston slap noise occurs due to piston impacts against the cylinder wall. A small amount of clearance (0.07% piston diameter cold clearance⁶¹) is required between the piston and cylinder wall to allow free movement. Piston slap results when a transverse force causes the piston to move into this clearance space and subsequently impact the cylinder wall.

Measurements of piston slap noise on diesel and gasoline engines have shown the affected frequencies can lie between 250 Hz and 10 kHz^{31,62,63,64}. The piston and cylinder are metal, producing hard impacts, with a Fourier force spectrum that can be flat up to and beyond 1.5 kHz. As the radiation of the noise by the engine at low frequencies is reduced by the radiation efficiency of the structure, and sparsity of radiating vibrational modes, the higher frequencies radiate the most noise.

Piston slap noise is audible as a short duration impulsive sound^{61,63} that can subjectively give the impression of poor mechanical reliability⁶³. Intense levels of piston slap noise are not only damaging to hearing. It has been reported that cavitation due to piston slap vibration has produced up to 2.5 mm of cylinder liner erosion per 1000 hr running in a wet liner diesel engine⁶⁵.

The level of the piston slap noise depends mainly on the side force and clearance in the cylinder. These parameters control the piston acceleration and distance over which the acceleration acts. The problem may be reduced to a single dimension, involving only the side force and lateral movement of the piston. Longitudinal forces along the cylinder axis do not directly contribute to the piston movement across the clearance space.

The piston side force is due to the varying angular offset of the connecting rod. The side force due to gas pressure varies with the cylinder pressure and connecting rod angle. Cyclic changes in linear and angular momentum of the piston and connecting rod provide another contribution to the side force. When the net side force acting on the piston passes through zero, it is commonly accompanied by a change in direction. Thus a piston initially positioned against the cylinder wall is forced into the piston and cylinder clearance space. As the side force changes, the piston accelerates within the clearance space. The piston energy at impact is equal to the product of force and distance through which the force acts, and can be approximated using the clearance space and the rate of change of side force curve at the zero crossing.

An intuitive accounting for the important factors affecting piston slap was provided by Ungar and Ross³⁶. Figure 2.3 illustrates the significant dimensions and forces. In this analysis γ ($\equiv R_S/L_C$) is the ratio of crank radius (R_S) and connecting rod length (L_C). Assuming γ is small and ignoring higher order terms, the equations can be reduced to a function of the crankshaft position θ_S using the following relations³⁶:

$$\begin{aligned}
\sin \phi_c &= \gamma \sin \theta_s \\
\cos \phi_c &= 1 - \frac{\gamma^2}{2} \sin^2 \theta_s \\
\tan \phi_c &= \gamma \sin \theta_s \left(1 + \frac{\gamma^2}{2} \sin^2 \theta_s \right) \\
\phi_c'' &= -\gamma(2\pi n)^2 \cos \phi_c \sin \theta_s (1 - \gamma^2 \cos 2\theta_s)
\end{aligned}
\tag{31}$$

With these assumptions, the inertia forces acting on the piston and connecting rod are described by³⁶:

$$\begin{aligned}
F_p^x &= -m_p x_p'' = (2\pi n)^2 m_p R_s (\cos \theta_s + \gamma \cos 2\theta_s) \\
F_c^x &= -m_c x_c'' = (2\pi n)^2 m_c (R_s \cos \theta_s + L_R^2 \gamma \cos 2\theta_s) \\
F_c^y &= -m_c y_c'' = (2\pi n)^2 m_c L_p \gamma \sin \theta_s
\end{aligned}
\tag{32}$$

where F_p^x is the piston inertia force along the cylinder axis (x); F_c^x and F_c^y are the connecting rod inertia forces parallel to and normal to the cylinder axis; x_p'' is the piston acceleration along the cylinder axis (x); x_c'' and y_c'' are the connecting rod accelerations parallel to and normal to the cylinder axis; m_p is the piston mass; m_c is the connecting rod mass; L_p is the length from the connecting rod center of gravity to the piston end of the connecting rod; $L_R (=L_c - L_p)$ is the length from the connecting rod center of gravity to the crankshaft end of the connecting rod; and n the engine speed.

The equations can be simplified by normalizing with respect to the force from a rotating mass equivalent to that of the piston, (and an additional factor γ). The normalized side force (Ψ_s) due to the piston, acting on the cylinder wall is³⁶:

$$\Psi_s \equiv F_{cp}^y(\theta_s) \left(\frac{1}{m_p R_s (2\pi n)^2 \gamma} \right) = (\Psi_G - (\Psi_p + \Psi_c)) \sin \theta_s
\tag{33}$$

where $F_{cp}^y(\theta_s)$ is the force between piston and cylinder.

The gas forces $F_G^i(\theta)$ act normal to the piston face and contribute to the side force due to the connecting rod angle ϕ_C . Normalizing in the form of equation 33, the contribution to the side force due to cylinder pressure is³⁶:

$$\Psi_G \equiv F_G^i(\theta_s) \tan \phi_C \left[\frac{1}{m_p R_s (2\pi n)^2 \gamma \sin \theta_s} \right] = \frac{F_G^i(\theta_s)}{m_p R_s (2\pi n)^2} \quad 34$$

Similarly, the inertia force of the piston ($F_p(\theta)$) along the cylinder (x) axis contributes to the side force as³⁶:

$$\Psi_p \equiv F_p^i(\theta_s) \tan \phi_C \left[\frac{1}{m_p R_s (2\pi n)^2 \gamma \sin \theta_s} \right] = \cos \theta_s + \gamma \cos 2\theta_s \quad 35$$

Allowing for the normalization factor, and including the connecting rod mass, the approximate expression above is equivalent to equation 19, (the shaking forces due to the piston and connecting rod).

Piston slap is determined by the difference between inertial and gas forces (eqn. 33), and accuracy is more important than in equation 19. Therefore the connecting rod forces are considered separately here. Due to the rotational inertia of the connecting rod, there is an additional side force component normal to the cylinder axis, as well as the component along the cylinder axis. These forces are taken into account by the following equation:

$$\Psi_C \equiv \frac{(F_C^j + F_C^i \tan \phi_C) L_R \cos \phi_C + (r^2 + L_R^2) \phi_C'' m_C}{L_C \cos \phi_C [m_p R_s (2\pi n)^2 \gamma \sin \theta_s]} \approx v l (v + \cos \theta_s + 2l\gamma \cos 2\theta_s) \quad 36$$

In the equation 36, r is the connecting rod radius of gyration, about an axis through the connecting rod center of gravity and parallel to the crankshaft. The remaining terms are defined by³⁶:

$$v \equiv m_C / m_p, \quad l \equiv L_R / L_C, \quad v \equiv (l - 2l^2 - r^2 / L_C^2) / l\gamma \quad 37$$

Equations 35 and 36 have essentially removed any direct dependency on speed from the estimation of side force Ψ_C in equation 33. This is shown by the approximate expressions

for equations 35 and 36, which although not used in calculations, would only result in a small error (typically less than 5% of the actual values).

The parameters affecting the side force in equation 33 can be seen in figure 2.4 by combining the inertial terms $\Psi_i = \Psi_p + \Psi_c$, and then comparing them to the combustion term (Ψ_c). The inertial terms are obtained by using the dimensions and masses of the pulpwood chain saw components (appendix G.1.2), and an assumed speed of 7500 rpm. The gas forces in the chain saw (Ψ_c) were more difficult to determine. Owing to the limited temperature rise and high compression ratios used in gasoline engines running with a small load, the cylinder pressure indicator diagram can be approximated by compression alone. An estimate for the gas forces has been obtained by considering the cylinder compression only, (dashed line in fig. 2.4). However, because of the similarity of the inertial and gas forces, the latter was determined from measurements of the pulpwood chain saw running at no load and 7500 rpm. The solid line represents the average value calculated for the normalized gas force (Ψ_c). The measured values did little to improve the previous estimate obtained using compression (dashed line), but they did indicate the lack of cycle to cycle repeatability under the measurement conditions. The shaded area in figure 2.4 is representative of the maximum and minimum measured combustion force (Ψ_c). These values were determined from 10 firings of the cylinder with the saw operating at no load and 7500 rpm. From consideration of the forces acting on the piston (fig. 2.4), at 7500 rpm and no load, the gas and inertial forces appear to contribute approximately equally to the side force the piston exerts on the cylinder wall.

The estimated side force Ψ_s experienced by the cylinder wall, is shown in figure 2.5 (obtained by substituting the gas and inertial forces from figure 2.4 into equation 33 for normalized side force). The variation in Ψ_s due to the gas force variation shown shaded in figure 2.4, is also shaded in figure 2.5.

Each time the gas and inertial forces (fig. 2.4) intersect, the side force (eqn. 33) on the piston (fig. 2.5) becomes zero. A subsequent change in the sign of the side force allows piston movement across the cylinder, resulting in a piston slap. Due to the $\sin\theta_c$ factor in the side force (eqn. 33), additional piston slaps can occur at crankshaft positions $\theta_c = 0$ and π (top and bottom dead center as shown in fig. 2.5).

2.6.1. Effect of clearance on impact magnitude (and hence noise)

The instantaneous acceleration of the piston as it crosses the clearance space is related to the rate of change of side force Ψ_s , which can be estimated from equation 33. Assuming

the piston flight is sufficiently short, the side force will change linearly with crank angle. The side force for a small change in crank angle (β), after the piston enters the clearance space at crank angle $\theta_s = \theta_o$ is given by:

$$F_{CP}'(\theta_s) = F_{CP}'(\theta_o + \beta) = \beta \left(\frac{\partial F_{CP}'(\theta_o)}{\partial \theta_s} \right) \quad 38$$

The equation of motion for the piston after it has entered the clearance is:

$$F_{CP}'(\theta_o + \beta) = \beta (m_p R_s (2\pi n)^2 \gamma \dot{\Psi}_s(\theta_o)) = m_p y'' = m_p 4\pi^2 n^2 (d^2 y / d\beta^2) \quad 39$$

where y is the lateral displacement of the piston, and $\dot{\Psi}_s(\theta_o) = \partial \Psi_s / \partial \theta_s$.

Using equation 39, the change in crank angle over which the piston is in free flight ($\Delta\theta_o$) and velocity (V_o) across the clearance space (d) can be estimated for each piston slap (occurring at crank angle θ_o) as follows³⁶:

$$\Delta\theta_o = \left(\frac{6}{\gamma \dot{\Psi}_s(\theta_o)} \left(\frac{d}{R_s} \right) \right)^{1/3} \quad 40$$

and:

$$V_o = R_s (2\pi n) \left(\frac{9}{2} \left(\frac{d}{R_s} \right)^2 \gamma \dot{\Psi}_s(\theta_o) \right)^{1/3} \quad 41$$

Substituting values appropriate for the pulpwood saw (appendix G.1.2), for the bottom dead center piston slap, the velocity (V_o) at initial impact is 0.2 ms^{-1} . The change in crankshaft angle ($\Delta\theta_o$) is found to be 15° .

The values of $\Delta\theta_o$ were calculated for each piston slap event, and used to predict the thick solid line in figure 2.5, which shows the side force on the walls allowing for zero wall force during the time the piston is in free flight. Note that the impulsive force due to the piston slap impact(s) at the end of the piston's flight is not shown.

In a typical elastic impact, the force pulse is similar to a half sinusoid, after impact the impacting body and impacted surface separate. Some of the initial kinetic energy is

transferred to the impacted structure, and some energy is retained by the impacting body. Piston slap differs from a single impact, in that after the impact, the increasing side force on the piston will help to hold the piston against the cylinder wall, creating multiple impacts, or possibly an extended force from the impact with no bounce^{64,66}. Simulated piston slap excitation using an electrodynamic shaker to reproduce the force on the piston has been found to result in multiple impact force peaks³¹, that diminish in amplitude with time.

The kinetic energy of the piston is expended in impacts with the wall and is either dissipated as heat, or transmitted into the chain saw. Assuming all the kinetic energy enters the saw, the kinetic energy of a piston of mass m_p and V_o (eqn. 41) produces a total vibratory power given by:

$$P_v = n \frac{m_p}{2} \sum_{cycle} V_o^2 \quad 42$$

where the summation represents the contribution for each piston slap event that occurs in a single engine cycle. Substituting the piston impact velocity (eqn. 41) yields:

$$P_v = nm_p R_s^2 (2\pi n)^2 \left(\frac{9}{2}\right)^{\frac{2}{3}} \left(\frac{d}{R_s}\right)^{\frac{4}{3}} \gamma^{\frac{2}{3}} \sum_{cycle} (\psi_s(\theta_o))^{\frac{2}{3}} \quad 43$$

The clearance space (d), determines the time of flight of the piston. A larger clearance allows more time for piston acceleration, and results in a higher impact velocity. In agreement with other theoretical studies^{62,66,67}, equation 43 predicts the piston vibratory power and radiated noise to be dependent on the clearance raised to the exponent 4/3. Thus, the impact vibration energy, and radiated noise, are predicted to increase at 4 dB per doubling of piston and cylinder clearance. This value is also supported by experimental studies that show changes up to 3 to 7 dB^{68,62,63} (depending on frequency).

2.6.2. Effect of speed on piston slap

If the work per engine operating cycle is assumed to be constant, then the cylinder indicator pressure (F_o) versus crankshaft position θ_s will not change significantly with speed, and the output power will be linearly related to engine speed. Due to normalization, at very low speed, the dimensionless gas force (Ψ_o , eqn. 34) has an

amplitude proportional to $1/n^2$, and can be many times greater than the normalized inertial force Ψ_i , which is essentially independent of speed (eqn. 35). Thus, at very low speed, the inertial forces (Ψ_i) can be ignored, and the side force will be dominated by the gas force (i.e., $\Psi_s \approx \Psi_g \sin \theta_s$). This condition will produce the minimum number of piston slaps, one at top dead center and one at bottom dead center, due to the $\sin \theta_s$ factor in side force (eqn. 33). It can be seen from figure 2.4 that if the side force is dominated by the gas force Ψ_g (Ψ_i negligible) then the slap at bottom dead center is insignificant compared to the slap at top dead center. The vibratory power, and hence noise, will be dominated by the slap at top dead center (θ_{TDC}), and the vibratory power P_v will be defined in the low speed limit by:

$$P_v = nm_p R_s^2 (2\pi n)^2 \left(\frac{9}{2}\right)^{2/3} \left(\frac{d}{R_s}\right)^{2/3} \gamma^{2/3} (\dot{\Psi}_s(\theta_{TDC}))^{2/3} \quad 44$$

At low speed $\dot{\Psi}_s(\theta_{TDC})$ is proportional to $1/n^2$, thus P_v (and hence the radiated noise) will be proportional to $n^{5/3}$. In other words P_v will increase by 5 dB per doubling of speed.

Conversely at very high speeds, the piston side force will be dominated by the inertial forces of the piston and connecting rod. As shown in equation 34, the normalized gas force, Ψ_g is proportional to $1/n^2$, and so, will become vanishingly small at high speed. At high speed, the piston slap forces depend on the inertial forces ($\Psi_i = \Psi_p + \Psi_c$). There will be four piston slap events, one each at top and bottom dead center (where $\sin \theta = 0$, eqn. 33), and two additional slaps at the points where $\Psi_i = 0$ (see fig. 2.4). As the normalized inertial forces (Ψ_i) are independent of speed, each $\dot{\Psi}_s(\theta_s)$ event in equation 43 is of order n^0 , and P_v is proportional to n^3 . Thus P_v increases at 9 dB per doubling of speed (for n large).

In summary, at low speed the vibration energy due to piston impacts (eqn. 44) has less dependence on engine rpm than at high speed. Normalization of equations 34 to 36 makes the gas forces Ψ_g proportional to n^{-2} , and Ψ_g determines $\dot{\Psi}_s(\theta_o)$. At high speeds the inertial forces Ψ_i determine $\dot{\Psi}_s(\theta_o)$, and normalization makes Ψ_i independent of speed. Thus, at low speeds the impact power is proportional to the engine speed raised to the exponent 5/3, or 5 dB per doubling of speed. At high speed the power is proportional to the cube of the engine speed, or 9 dB per doubling of speed. These low and high speed

asymptotes are shown by the dashed lines in figure 2.6 that show the total vibratory power (from kinetic energy) of the piston slaps.

At intermediate speeds where the gas (Ψ_g) and inertial (Ψ_i) forces are similar, (as in fig. 2.4) the intersection of the curves can produce additional piston slap events, which will increase the vibratory power (P_v). Calculation of the side force as in figure 2.5 for a number of different speeds allowed an estimate of $\dot{\Psi}_s(\theta_o)$ for each piston slap event, and hence a calculation of P_v as a function of chain saw speed. The resulting vibratory power P_v has been plotted in figure 2.6. Due to the variations in the no load gas force levels of the chain saw, figure 2.6 has been plotted for the average, maximum and minimum (compression) gas force levels found in figure 2.4. This figure indicates the variation in vibratory power levels ($10\log P_v$) due to gas force fluctuations, is about ± 4 dB, at speeds near the best cutting speed (7500 rpm) of the chain saw. Due to the complexity of the interaction between gas and inertial forces, the power calculated from the averaged gas forces actually falls outside the limits defined by the minimum and maximum gas force (fig. 2.6). Thus, in the limited speed range that can be examined with the pulpwood saw (3800 to 9600 rpm), it is not possible to accurately predict the change in vibratory power with speed, other than to note a general increase in vibratory power output with speed.

Note also, in this intermediate speed range, the rate of change in side force $\dot{\Psi}_s(\theta_o)$ is small for the additional piston slaps. The piston will likely be affected to a greater extent by the oil film, friction from piston rings, and rotation of the piston within the cylinder, leading to additional uncertainty. These factors cannot be easily estimated, and may require measurements to determine the correct values. This was done in one study⁶⁹ by measuring the structural response to cylinder impacts, and estimating the effect of the oil film and piston rings. This produced good agreement for the timing of experimentally measured vibration, although of course the analysis was only applicable to the engine studied. At lower speeds Griffiths and Skorecki⁷⁰ were able to predict the timing of occurrence piston slap events using an analysis similar to that above.

One final caution, as shown by the solid line in figure 2.5, near top dead center, the crank angle over which the piston is in free flight (indicated by the solid line coinciding with zero force), may exceed the time required for the force to reverse again. This would extend the time the piston is in free flight. Note that rotation of the piston about the gudgeon pin can produce an impact on the corner of the piston⁷¹. As this does not require the entire mass of the piston to cross the clearance space, a short duration force reversal may result in

only one corner of the piston impacting the opposite wall. Thus, at speeds where the gas and inertial forces are similar, additional uncertainty will be present in the number and magnitude of piston slaps.

Measurement of the pulpwood saw piston slap noise at full load would help to separate the gas and inertial forces, although the difference would not be large. It would only increase the gas forces by a factor of two, which would shift the region of variation in figure 2.6 to the left (by about 2000 rpm).

2.6.3. Production of piston slap impact noise

Ungar and Ross describe a procedure for predicting the noise produced by the piston slap events. In following their procedure the estimated noise of piston slap in the pulpwood saw would be about 73 dB (unweighted) at the operator's ear. Their procedure calculated the conversion of impact energy to radiated noise using an estimated impedance due to an infinite plate in series with the impedance of the air (with allowance for the radiation efficiency and area of the engine). There was no explicit accounting for dissipation of energy by structural damping, which directly affects the ratio of energy absorbed in the structure versus energy transported away from the structure. For example, zero structural damping would result in the radiation of all impact energy as sound³⁴. Ungar and Ross may have allowed for the structural damping by choosing a low value for the radiation efficiency. The value they used was empirically chosen as 0.03, averaged over all frequencies. This value is low, since measurements on even a small diesel engine showed the radiation efficiency dropped below 1 only for frequencies below 400 Hz³¹. Larger engines (as discussed by Ungar and Ross) would further reduce the frequencies where the radiation efficiency became low⁵⁰. Although it is unclear how Ungar and Ross obtained the estimate of acoustical power, it may be an empirical estimate which would apply to the large engines they studied. Assuming the chain saw is similar to those engines, the predicted value of 73 dB is a reasonable estimate of piston slap noise from the chain saw. This estimate will be reduced somewhat by the A-weighting function.

A less empirical estimate of the radiated noise spectrum can be obtained using Richards' analysis as discussed in section 2.5 on impact noise. The total mechanical vibratory power of the piston slaps may be calculated by equation 44, based on the kinetic energy of the piston. This corresponds to equation 22 for the vibratory power produced by an impact. The spectrum of equation 22 is determined by the functions $\Re\{1/Z\}$ and $|F_{cycle}|^2$. Z is the impedance of the cylinder wall as a function of frequency, and represents the structural

response of the engine. F_{cycle} is the Fourier transform of the force versus time history, and represents the spectrum of the force of the individual impacts occurring with every cycle.

2.6.4. Force spectrum

At this point an estimate has been obtained for all parameters except the piston slap impact force spectrum. The force exerted by the piston on the cylinder wall is a combination of the impact force, and the side force on the cylinder walls (fig. 2.5). The impact of the piston is assumed dominant, and the additional effect of side force will be discussed in the next section.

The vibratory power (eqn. 22) is dependent on the impulse force spectrum as well as the real part of the impedance presented by the structure. The impulse force spectrum is determined by the piston and cylinder wall properties, and the surface area involved in the impact. It is related to the imaginary (spring like) part of the structural impedance.

Unfortunately, the corner frequencies where the force derivative spectrum (F') decreases are not known for the piston and cylinder impact. Rather than attempt to calculate these frequencies, a rough similarity is assumed between piston and cylinder wall impacts in diesel engines and chain saws. Spectral estimates of the latter are available for the force provided by piston slap in diesel engines^{31,34}. The chain saw and diesel engine structures are assumed geometrically similar, all dimensions are proportional to some characteristic length (L). The frequencies are related to the spring-like component of the structural impedance, and can be expected to scale as some function of the Young's modulus (Y), material density (ρ_m), and characteristic length (L). These three quantities can be combined to represent the impact area (A), perimeter of the impact area (L_A), piston mass (m_p), bending stiffness (B) (eqn. 28), shear modulus (G), and material thickness (h) i.e.:

$$YL^3 \propto B \quad Y \propto G \quad L \propto h \quad L \propto L_A \quad L^2 \propto A \quad \rho_m L^3 \propto m_p \quad 45$$

where the unknown constant of proportionality is different for each of the above. However, due to the assumed geometric similarity, each equation has the same constant of proportionality for the chain saw and diesel engine.

Thus, depending on the wall construction, the spring-like stiffness (K) may be any combination of:

$$K \propto \frac{YL^2}{L} \text{ or } K \propto GL \text{ or } K \propto \frac{BL}{L^3} \quad 46$$

Then assuming the corner frequencies are roughly proportional to some equivalent single degree of freedom system with stiffness (K) and mass (m) defined by:

$$YL \propto K \text{ and } \rho_m L^3 \propto m \quad 47$$

The natural frequency (f_n) of a single degree of freedom spring mass system is given by:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{K}{m}} \quad 48$$

And thus:

$$f_n \propto \sqrt{\frac{Y}{\rho_m L^2}} \quad 49$$

Assuming the diesel engine is made of steel and the pulpwood saw is mainly aluminum, the ratio of Y/ρ_m is approximately equal for both the diesel and chain saw. Thus, the impact frequencies are inversely proportional to a characteristic dimension of the engine (L^{-1}). So, due to the small size of the chain saw, the impact frequencies would be about a factor of 2 larger. Richards³⁴ gave values for the characteristic upper and lower frequencies for diesel engine piston slap, the high frequency limit being 1.5 kHz, and the low frequency limit as 2 Hz. Ignoring the low frequency limit as representative of the side force on the cylinder wall (discussed in next section), suggests that the characteristic sine wave frequency of a single impact event is about 1.5 kHz. If a half sinusoid pulse is further assumed for the impact event, the upper limiting frequency of the force derivative (F' used in eqn. 23) would be 3 kHz for the diesel engine. The corresponding values for the pulpwood saw would be 3 kHz and 6 kHz, representing the flat part of the spectrum F' . At higher and lower frequencies the F' spectrum will fall away as f^1 and f^{-1} respectively. These values should be considered as very rough estimates, as Richards did not discuss the configuration of the diesel engines, other than to mention a typical material thickness of 1 cm for most of the machinery studied³⁴ (note that this is twice the typical material thickness of the pulpwood chain saw).

Comparison with experimental studies allows an appreciation of the frequencies that may affect piston slap. In two studies where piston slap noise was increased by enlarging the piston/cylinder clearance, the radiated noise increased between 1500 to 20000 Hz with the peak acceleration recorded between 2500 and 4000 Hz⁶² for a single cylinder diesel. For a four cylinder gasoline engine the largest effect on block vibration was between 800 to 4000 Hz⁶³. Thus based on the acceleration measurements, it would appear the lower limiting frequency for the force derivative was around 4 kHz. In a wet liner diesel engine, the small thickness of the liner could be expected to provide a reduction in liner stiffness, leading to lower frequencies for the impact pulse⁶¹. This was apparently the case in one wet liner diesel engine, where the piston slap force derivative lower and upper limiting frequencies were measured to be 250 Hz and 500 Hz³¹. On another wet liner engine the equivalent stiffness for a piston-liner impact was measured to be 400 MNm⁻¹, and 25 MNm⁻¹, at the top and bottom of the piston respectively. The piston mass was not reported, but appeared to be about 3 kg (based on its reported speed and inertia)⁶⁹. The resulting lower limiting frequency of the force derivative spectrum would be 1.8 kHz, or 460 Hz, for an impact at the top or bottom of the piston skirt respectively. There thus appears to be a factor of up to 10 difference in frequencies affected between wet liner, and cast engine blocks.

The engines listed above probably used steel cylinders and steel pistons. In one other study of a 2.37 l, 4 stroke wet liner diesel, the engine had piston and connecting rod masses similar to the pulpwood saw. The piston was likely made of aluminum, making the results more applicable to the pulpwood saw. On the basis of vibration velocity measurements or calculations the lower corner frequency for the force derivative appeared to be around 2000 Hz⁶⁴. However, as this was a wet liner engine, the cylinder wall stiffness would be low, and the corresponding frequency on the pulpwood chain saw would be higher. The mass and impedance of each component were measured individually, thus the lower limiting frequency for the force derivative was recalculated with infinite impedance in all components except the cylinder skirt. Using only the compliance of the piston skirt, and ignoring any other stiffness measured, the lower corner frequency for the force derivative would be at 6500 Hz.

Thus an average lower limiting frequency for the force derivative (F'') estimated from the above engine measurements with steel pistons is 4000 Hz, or 500 Hz for wet liner diesel engines. Richards' estimate of 1.5 kHz thus appears to be an appropriate approximation, although the actual value can vary significantly. From the above assumption of geometric

similarity between the chain saw and diesel, the corresponding estimate of lower limiting frequency of the force derivative on the chain saw would be 3 kHz using Richards' estimate, or 8 kHz based on engines without wet liners. The best estimate should be based on the aluminum piston and skirt compliance (6500 Hz). However, the large variations in data suggest that measurements of the piston slap noise of the pulpwood saw are required to clarify the frequencies at which the piston slap force spectrum rolls off.

Calculated spectra for a 3 kHz, and 8 kHz lower limiting frequency with other parameters appropriate for the pulpwood saw are shown later in figure 5.16. Since the vibrational energy is the same in each calculation, the 3 kHz estimate exceeds the 8 kHz estimate below 3 kHz, and vice versa at higher frequencies.

2.6.5. Comments on the piston side force

Priede has made reference to piston side force affecting engine noise⁷², stating that for one small engine studied the side force was more significant than piston slap, and for a large engine the piston slap was more significant. However, on these two engines, based on a comparison with cylinder pressure amplitude, the side force was assumed not to contribute to the noise. A later paper by the same authors suggested through qualitative arguments that the side force affects engine noise at high speeds⁶⁷.

For most engines the side force variation (fig. 2.5) is at a frequency too low to affect significantly the noise from the structure. However, the flight of the piston temporarily removes the side force from the cylinder wall creating a discontinuity in the applied force. This creates higher frequency excitation that extends up to the piston impact frequencies⁶⁷. Thus the piston flight across the clearance is linked to the noise radiated due to the side force.

Richards' analysis (section 2.5 on impact noise) could be used to predict the effect on noise. He estimated the frequencies affecting the derivative of the piston slap force spectrum to peak between 2 Hz and 1.5 kHz³⁴. No distinction was made between side force and impact force, thus the impact force would sum with the side force. In fig. 2.5, which shows the side force, the piston impact force is not shown. The impact would produce an impulse in the side force at the time the piston first contacts the wall. Thus the high frequencies are related to the piston slap, and the low frequencies are associated with the side force.

The frequency response of the side force and impact force is filtered through the mass of the piston and stiffness between the piston and the cylinder wall, which reduces the high frequency content of the side force. As the high frequencies from the side force are due to a discontinuity in the time history, the force spectrum will decrease as f^{-1} above the fundamental repetition rate of the time history (approximately equal to the engine rotation rate, see fig. 2.5). The effect of filtering through the piston and cylinder wall will cause a further decrease with increasing frequency. Thus the frequency derivative of the side force will vary as f^0 between the engine rotation rate, and the characteristic frequency of the piston and cylinder wall vibration. At higher frequencies the side force will decrease with increasing frequency. The side force derivative spectrum begins to decrease at the same frequency the piston slap force derivative spectrum attains its maximum value. Thus the impact and side force affect separate frequency ranges. The frequencies involved depend on the imaginary, or spring like part of the wall impedance.

The relative magnitudes of the impact and side force also depend on the cylinder wall impedance (or mobility). This is best illustrated through some examples. For simplicity, assume that the piston is completely rigid. If the impedance of the cylinder wall is low and well damped, the decay of the impact force would smoothly blend into that of the side force. The side force versus time history would form a smooth continuous curve, which would produce a smooth continuous spectrum as Richards describes. Thus for a low impedance cylinder wall, the impact and side force spectrum would overlap and the side force would smoothly extend the piston slap impact noise to lower frequencies.

For a high impedance wall with high damping, the force pulse experienced by the cylinder wall would appear as an initial spike superimposed on the (roughly sinusoidal) side force curve (of fig. 2.5). In this case the spectrum would have a different form. The effect of cylinder side force and impact pulse spike could be considered as separate events. The energy of the impact would be concentrated at a high frequency related to the width of the impact force spike. The side force spectrum would be the spectrum of the side force time history (fig. 2.5), which can be loosely approximated as a truncated sine wave. Thus the combined force spectrum would be discontinuous, consisting of a peak at the engine rotation rate (fig. 2.5), while at a much higher frequency, another peak would correspond to the characteristic frequency of the piston cylinder impact. The force spectrum at intermediate frequencies between the peaks would decrease smoothly as f^{-1} due to the discontinuity in the side force.

Finally, for a high impedance wall with relatively low damping, the piston could bounce repeatedly off the wall. The side force would simply serve to decelerate the piston, and then return it to the wall; a process that would have little effect on the piston energy. The kinetic energy of the piston would be gradually transferred to the wall with each bounce, and the energy would be transferred to the wall at frequencies characteristic of the piston impact. Thus the piston impact frequencies would dominate the spectrum.

Hence, the cylinder wall impedance can significantly affect the frequency distribution of the piston slap impact and side force. In the above cases all the initial kinetic energy of the piston, will be either dissipated within the structure (structural damping), or radiated as noise by the chain saw. The amount of energy due to the side force decreases with increasing wall impedance, and the side force spectrum levels can exceed that of a piston impact for a low impedance wall. In other words, if the cylinder wall moves easily (low impedance), more work will be done by an applied force, and more energy will be transferred. If the cylinder wall does not move (infinite impedance), no work will be done by an applied force, and an impacting (rigid) piston will bounce until all its initial kinetic energy is eventually transferred to the wall.

The noise radiated by the side force can be compared to that of piston impacts. Measurements of piston-cylinder impacts with simulated side force on a non running engine show that the piston can bounce on impact³¹. This makes the piston impact frequencies dominant. In any case, as the side force produces an equal and opposite force acting through the crankshaft, its effect on the chain saw will be minimal below the lowest bending mode in the saw structure. As discussed in the previous section, experimental studies have shown piston slap is most apparent at frequencies well above the engine rotation frequency (which is comparable to the fundamental frequency of the side force)^{31,62,63,64,69}. Possible effects of side force have only been found at low frequencies (around 500 Hz⁶⁹). This indicates side force is less important than the kinetic energy of the piston. On the chain saw, the lowest bending mode was previously estimated as 1.2 kHz, and the piston impact noise was estimated to extend to 6.5 kHz. Thus on the pulpwood saw, the side force frequency response will show a drop above 6.5 kHz, and the effect on noise due to side force will be most apparent between 1.2 kHz and 6.5 kHz.

Although a quantitative estimate of the noise levels due to the piston side force requires an estimate for the magnitude of the wall impedance, it is possible to determine a qualitative measure of the side force noise. The harmonics of the side force will contribute to the radiated noise above the lowest bending mode of the structure. As the side force and

piston slap mainly affect different frequency ranges, they can be considered independently. An estimate will be made to calculate the sound directly related to the side force time history. To assume the largest possible contribution to the noise, implies a further assumption that the piston does not bounce off the cylinder wall.

The discontinuity in the side force is equivalent to removing a triangular area from the side force time history (i.e., see fig. 2.5 BDC slap). The harmonic levels will be directly related to the area omitted from the side force curve due to the flight of the piston (note, any two curves that sum to form a sine wave, have identical spectral magnitudes at all frequencies except the sine wave frequency). For example, if the piston is always in contact with the cylinder wall, the side force curve will have negligible harmonics. The piston time of flight occurs over a relatively small crank angle rotation, where the rate of change of side force can be assumed constant. When the piston impacts the wall the side force will increase to the level it would have with zero clearance. As the piston is in flight for longer times, the triangular area removed from the side force curve increases in size, and more energy is supplied to the harmonics. Note that since the area removed is discontinuous, the harmonics in the frequency domain decrease as f^{-1} .⁵⁹

The previous analysis of impact noise (section 2.5) can be applied to the triangular discontinuity in the side force. Equation 23 is repeated here for the vibratory power due to side force (P_{vs}):

$$P_{vs}(f_o) = 2n\Re\left\{\frac{1}{Z(f_o)}\right\} \frac{|F'(f_o)|^2}{4\pi^2 f_o^2} \Delta f \quad 50$$

Substituting equation 23 in equation 21 will allow an estimate of the noise due to the side force. The vibratory power $P_{vs}(f_o)$ scales directly as $|F'(f_o)|^2$, the square of the magnitude of the derivative of the side force spectrum. Since Z was approximated as constant with frequency (section 2.5.3) equation 22 can be changed to the time domain using Parseval's identity. The result can then be expressed in terms of crankshaft angle by transforming variables from time to $\theta_s = \theta_o + \beta = 2\pi nt$:

$$P_{vs} = 2n\Re\left[\frac{1}{Z}\right] \int_0^{2\pi} |F(f)|^2 \partial f = \frac{1}{2} \frac{1}{\pi} \Re\left[\frac{1}{Z}\right] \int_0^{2\pi} |f(\theta_s)|^2 \partial \beta \quad 51$$

note the factor $1/2$ on the RHS, since F is a one sided spectrum from 0 to ∞ .

Substituting the value (and spectral shape) from equation 51 into equation 21 provides an estimate of the radiated noise spectrum. Note that all parameters except F (or $\mathcal{F}$) are unchanged from the previous considerations of piston impact noise, and the relative amplitudes of noise due to side force versus impact force can be obtained by consideration of the force function alone. Substituting the force versus crank angle (eqn. 39) for $\mathcal{F}(\theta_s)$ in equation 51, gives the vibratory power of the chain saw due to the triangular discontinuity in the side force (fig. 2.5). The solution to the integral is:

$$P_{vs} = \frac{m_p^2 R_s^2 (2\pi n)^4 \gamma^2 (\dot{\Psi}_s(\theta_o))^2}{2\pi} \Re \left\{ \frac{1}{Z} \right\} \frac{\Delta\theta_o^3}{3} \quad 52$$

Then substituting for the piston time of flight $\Delta\theta_o$ from equation 40.

$$P_{vs} = (2\pi n)^4 \dot{\Psi}_s(\theta_o) \left(\frac{d}{R_s} \right) \frac{m_p^2 R_s^2 \gamma}{\pi} \Re \left\{ \frac{1}{Z} \right\} \quad 53$$

When comparing this equation to equation 43 (P_v for piston slap), there is a larger dependency on speed n and a lesser dependency on clearance (d). This is shown by a ratio of the power due to the piston kinetic energy (P_v) and the power given above in equation 53:

$$\frac{P_{vs}}{P_v} = m_p 2\pi n \Re \left\{ \frac{1}{Z} \right\} \left(\frac{32}{81} \left(\frac{R_s}{d} \right) \gamma \dot{\Psi}_s(\theta_o) \right)^{1/3} \quad 54$$

As discussed previously, the higher the wall impedance (Z), the lower the effect of side force, i.e., an infinite impedance wall would not be affected by the side force. The kinetic energy of the piston is more strongly influenced by clearance than the side force on the wall. This is to be expected as the piston accelerates under a constantly increasing force, a larger clearance provides a higher impact velocity.

What may be surprising is the explicit dependence of the side force energy on piston mass (m_p). This can be seen from equation 53. The side force is influenced by the piston mass due to the inertia of the piston acting through the angle of the connecting rod, and the amplitude of the harmonics of the side force increase as the time of flight of the piston lengthens.

Note that $\Psi(\theta_o)$ is a function of rpm, previous considerations in section 2.6.2, shows the relation to be given by:

$$\Psi(\theta_o) \propto \begin{cases} n^{-2} & \text{low speed asymptote} \\ n^0 & \text{high speed asymptote} \end{cases} \quad 55$$

Thus the effect of speed on the ratio of side force to impact power (eqn. 54) reduces to:

$$\frac{P_{vs}}{P_i} \propto \begin{cases} n^{1/3} & \text{low speed asymptote} \\ n^1 & \text{high speed asymptote} \end{cases} \quad 56$$

So as speed is increased the side force vibratory power could eventually exceed the impact power. From equations 34 and 56, at high speeds the side force vibratory power could increase up to 12 dB per doubling of speed.

The ratio P_{vs}/P_i in equation 54 is 0.5 using values appropriate for the pulpwood saw (from appendix G.1.2), the impedance given by equation 30 (estimated at 800 kg s^{-1}), and approximating Ψ as 1 (see fig. 2.5). This indicates the side force should be less significant than the kinetic energy of the impact. Note the side force should decrease at 20 dB per decade increase in frequency, making the high frequencies less significant. The radiation efficiency should reduce the low frequency levels below 630 Hz. Also, if the piston bounces off the cylinder wall, the side force will be even less significant, and the frequency response of the impact and side force will overlap.

An equation for side force energy based on highly simplified empirical arguments has been suggested by Lalor and Grover as an explanation for the large increase in noise noted with diesel engines operated at relatively high speed⁶⁷. Due to the lower speeds of engines used for transportation, the ratio P_{vs}/P_i would likely be lower than found in the pulpwood saw, and the side force consequently less significant. Lalor and Grover dropped this explanation in favor of bearing noise in subsequent papers.

As the side force is slowly varying, it affects frequencies around the firing frequency. This was indicated in two experimental studies where noise below 500 Hz was attributed to the side force^{69,73}.

2.6.6. Limitations of the above analysis

Since the present calculations rely on the analysis of Ungar and Ross, they may exceed measured results by about 10 dB. In one study⁶⁴ of piston slap noise, the analysis of Ungar and Ross was used to determine the piston velocity. The engine vibration was calculated by using the predicted piston velocity and measurements of the mobility of the piston and cylinder wall. The calculated vibration exceeded measured values by up to 10 dB at most frequencies above 1 kHz. Differences were attributed to a failure to account for losses affecting piston velocity⁶⁴. Other factors that may contribute to discrepancies between measurement and theory can be deduced from comparison with other literature.

As mentioned briefly above, besides the approximations made, the above analysis does not include effects due to piston rotation, or oil film damping. There is also friction between the piston, gudgeon pin, cylinder bore, and piston rings. While these have been qualitatively considered in other studies^{31,66,69}, the errors due to their omission have not been quantified. As these factors can be varied to reduce radiated noise, an indirect measure of their importance can be obtained by examining their effectiveness in noise reduction.

Offsetting the gudgeon pin with respect to the piston centerline, or center of gravity is a common noise control measure: this can be expected to affect the rotation of the piston (as well as the timing of the events). The effect of moving the gudgeon pin within reasonable limits has shown noise variations of about 3 dB^{62,66}, with the effect on individual slaps somewhat higher.

The damping between piston and cylinder provided by the oil film can be made significant. Forced oil injection along the side of the piston has produced a 10 dB total engine noise reduction attributed to elimination of piston slap^{74,75}. In one study, normal lubrication was found to produce rotation of the piston, which did not occur without lubrication⁷⁴.

Friction between the cylinder and piston (or piston rings) adds an additional force not considered in the above analysis^{66,75}. This friction can also damp the movement of the piston in the clearance space. A similar effect can be produced by squeezing the oil within the grooves for the piston rings. Removal of the piston rings increases the noise: more rings tend to reduce noise⁷⁴ (although this also can change the effective piston wall clearance³¹). The added friction can also affect the time of flight of the piston within the clearance. The value for the friction was calculated in one study by varying the assumed

friction until the piston slap event timing matched experimental data⁶⁸. The timing error associated with an assumption of zero friction was up to 20° of crank rotation.

The frequency range affected by the impacts directly affects the A-weighted noise energy received by the operator. For example, frequencies beyond the audible range can be ignored. Plastic inserts in the piston affect the effective stiffness between the cylinder and piston, (as well as clearance and friction) and have reduced noise by 4 dB⁷⁶. Ohta⁶⁹ also found the mobility (inverse of impedance) differs at top and bottom of the piston, which will also affect the frequency response. As seen in section 2.6.4, the frequencies affected by piston slap can vary by a factor of 10 depending on engine design.

In view of the considerations above, it is likely that the radiated noise energy and frequencies affected could vary by as much as 10 dB and a factor of 10 respectively. Predicted levels will most likely cause an overestimate of the measured values by as much as 10 dB. Simple noise measurements on the pulpwood saw can be used to obtain a better estimate of the approximations made.

2.6.7. Implications for the measurement of piston slap

Using equation 21 the piston slap acoustic power has been calculated at the operator's ear, for the pulpwood saw at 7500 rpm. The A-weighted sound level is 97.9 dB, and somewhat exceeds the mechanical noise levels for chain saws found in the literature (95, 94 dB A-weighted)^{20,23}. Since errors on the order of 10 dB are expected, the difference is not surprising. Measurements on the pulpwood saw will be required to clarify the significance of piston slap noise.

The measurements will be complicated by the interaction between combustion and inertial forces. The gas and inertial forces on the piston are similar on the pulpwood saw (fig. 2.4). Thus due to the limited speed range possible on the pulpwood saw the vibratory power due to piston slap is not easily predicted. Changing the engine speed would not likely be useful in separating the effect of piston slap. Externally motoring the engine is possible, and would roughly approximate the combustion forces during the compression cycle. However, the lubrication and thermal effects on clearance would not be present.

One method of identifying the noise contribution of piston slap is to increase the clearance between piston and cylinder. The effect of clearance (d) on vibratory power scales as $d^{4/3}$ (eqn. 43). Although piston slap noise may be masked by other sources, increasing the

piston and cylinder clearance by a factor of 5 will increase the noise in each frequency band by 9 dB. If piston slap noise is at all significant, a 9 dB change will make it apparent.

Although too large a clearance between the piston and cylinder produces piston slap noise, too little clearance can also apparently increase noise. In two separate case studies, the piston to cylinder clearance was decreased to the point that the piston and cylinder fit was potentially an interference fit. The tight fitting pistons apparently produced a metallic diesel sounding knock⁷⁷ at idle. As the knock could be eliminated by reducing the friction between piston and cylinder, this noise was dubbed piston stick slip noise⁷⁸.

2.7. Bearing noise

The sources listed so far do not account for all the noise of the chain saw. After elimination of exhaust, intake, chain and fan, the noise of chain saws has been measured at 95, 94 dBA^{20,23}. The causes of this noise are not clear from the literature, although the bearings are commonly mentioned in studies of transportation engine noise^{63,79}. It has been suggested that bearing impacts are the main cause of noise for engines operated at high speed³.

Different noise generation mechanisms are present depending on the bearing design. A ball bearing operates with essentially zero clearance, it is the defects in the bearing that produce noise. Condition monitoring using the noise and vibration of such bearings is extremely useful in installations where shutdowns due to repairs, or failures can be expensive. In other bearings, clearances are required. This allows contact loss and subsequent impacts, just as can occur for piston slap.

2.7.1. Bearings requiring clearance

Plain bearings and needle bearings require clearance to allow lubrication and avoid binding between bearing surfaces. Impacts in bearings are much more complicated than piston slap impacts. With piston slap, the problem is reduced to a single dimension, only movement across the clearance is significant. In a bearing, the movement is significant in two directions.

The net force on the shaft must be away from the surrounding "wall" for contact loss. The initial velocity is tangential to the wall (i.e., immediately preceding contact loss). Once the shaft has entered the clearance space, its impact energy is dependent on the final velocity normal to the wall. The normal component of velocity depends on the direction of flight

and impact location on the wall. The motion of the shaft, thus depends on the history of force, velocity and position. Small changes in any of these three factors can change the motion from cycle to cycle. There are also additional complications, for example, if there is friction between shaft and wall, the angular velocity of the shaft also affects the normal and tangential velocity⁸⁰.

Dynamic modeling of bearing motion usually involves numerical solution of sets of highly nonlinear differential equations^{81,82}. Thus the factors affecting bearing motion have not been identified as completely as for piston slap. Earles et al.⁸³ used a numerical solution to determine contact loss, and then attempted to make generalizations with the aid of an experimental model. As in piston slap, contact loss was expected when the component of bearing force normal to the wall passed through zero, producing an instantaneous change in the direction of force. Unlike piston slap, the normal force depends on the shaft position within the bearing. The normal force on a bearing may be zero, even if the total force on the bearing is non-zero. To simplify the problem, Earles et al. attempted to relate the zero clearance bearing force polar to the occurrence of contact loss (i.e., fig. 2.7).

Two main factors were significant to the occurrence of contact loss⁶: the bearing force (F) becoming very small, or a large rate of change of direction of force with time (ϕ'). On the basis of their experimental model, Earles et al.⁶ found contact loss was more likely if $\phi'/F \geq 1$ rad/Ns. This dimensional value was also verified by other researchers⁸⁴. The importance of these parameters has been verified in other studies⁸⁵. In one study, a reduction of these parameters was used to reduce the noise of the connecting rod bearings⁸⁶.

The pulpwood saw connecting rod big and little end bearing forces are shown on a polar diagram in figure 2.7. The forces are normalized and derived from equations used in the previous section on piston slap, for a no load speed of 7500 rpm. Dashed lines show where $\phi'/F \geq 1$ rad/Ns, and thus contact loss may occur. These results indicate contact loss can occur at the big end of the connecting rod at top dead center. At the little end bearing, contact loss is only indicated around $\pm 50^\circ$ before and after top dead center.

At low speed, the impact intensity was found to be proportional to the natural logarithm of ϕ'/F .^{84,87}

$$I = a(b^{1000 d/D} - 1) \ln \left(\frac{\phi' \text{ Ns}}{F \text{ rad}} \right) \quad 57$$

where I is the impact acceleration level in g's, a , and b are empirically determined system and bearing factors, d is the bearing clearance, and D is the bearing diameter. The constant b was found to be 1.28. Of note, the factor a depended on the bearing lubrication⁸⁷:

$$a = \begin{cases} 0.61 & \text{unlubricated} \\ 0.48 & \text{boundary lubrication} \\ 0.14 & \text{film lubrication} \end{cases} \quad 58$$

Assuming the vibration spectrum shape is independent of the bearing clearance, and the number of impacts is unchanged by an increase in clearance, the effect on noise of such a change can be predicted. Substituting values appropriate for the pulpwood saw gudgeon pin bearings, equation 57 predicts an increase in vibratory power with clearance of 5.4, 14, and 29 dB respectively if the clearance is increased by a factor of 2, 4, or 8. These changes are much greater than would be expected with piston slap where theory predicts 4, 8, or 12 dB for the same increases in clearance. However, Earles used smaller d/D ratios in the bearing clearances tested. For the values he used, the change in impact power appeared to be about 4 dB per doubling of clearance, just as found for piston slap. Thus, if Earles et al. misinterpreted the functional relationship between noise and clearance, equation 57 may not be applicable to the larger clearances used on the pulpwood saw.

Summarizing the remainder of the findings of Earles et al.: In a two bearing rig no interaction between bearing impacts was found. At higher speeds, the occurrence of impacts was still accurately predicted if $\phi'/f \geq 1$ rad/Ns, but the logarithmic dependency of impact intensity was not. It is for this reason that the authors suggested using a numerical approach to confirm the results.

Another predictor for bearing noise impacts was suggested by Dubowsky⁸¹. On the basis of experimental and theoretical studies of a bearing moving in an elliptical path, he calculated the motion of a mass inside the bearing. Impacts occurred when an impact prediction number (IPN) exceeded 0.2. The impact prediction number was defined by⁸¹:

$$IPN = \left(\left(1 - (R_s/R_d)^3 \right) (d/2R_s) \right) / (n/f_n) \quad 59$$

where R_b is the minor axis of the elliptical bearing path, R_a is the major axis of the bearing path, d is the clearance between mass and bearing, n is the rotation rate of the bearing along the elliptical path, f_n is natural frequency of the mass within the bearing.

Equation 59 only accounts for gravitational and inertial forces on the mass due to the bearing path. It predicts impacts always occur at the little end of a connecting rod, and impacts never occur at the big end. This makes the model appear to be of little use. Dubowsky, however, indicates that the model suggested by Earles et al. (eqn. 57) does not predict impacts in his experimental model.

Some additional insight into bearing impacts may be obtained by examining equation 59. It is not immediately obvious from the analysis of Earles et al., that in an internally controlled system, increasing speed n can reduce the ratio ϕ/ϵ , thus helping to eliminate impacts. Equation 59 indicates the natural frequency of impact has an effect, although this is not considered by Earles et al.. Thus less stiffness in the bearing, or a larger mass could reduce the likelihood of impacts. Dubowsky's model also indicates that increasing the bearing clearance d increases the likelihood of impacts, thus making measurements using increased bearing clearances difficult to interpret.

Impacts across bearing clearances have been identified as a primary noise source in automotive gasoline engines⁸⁸. The rate of increase in bearing noise with engine speed has been shown to be up to 86 dB per decade³.

Demonstrations of bearing noise have relied on increasing the bearing clearances. Increased journal bearing clearances have resulted in up to 10 dB noise increases when the bearing clearance was doubled⁸⁸, and up to 10 dB increase has been noted with a tripling of gudgeon pin clearance⁸⁹. These values are similar to the effect of clearance on piston slap. It is noteworthy that a reduction of gudgeon pin clearance by a factor of four, produced no effect on noise⁸⁹.

An estimate of the frequencies affected by gudgeon pin bearing impacts is difficult to obtain without a model of the impacts. As the force on the bearings increases, the bearing element can accelerate across the clearance space resulting in an impact. The noise is broadband, with a frequency response influenced by the hardness of the impacting materials at high frequencies, and by the structural response of the machine at low frequencies. In an internal combustion engine, the bearing forces are similar to those on the piston, and depend on balance of combustion and inertial forces. The impact intensity may be affected by lubrication or grease packing, if the lubricant reduces the amount of

energy gained when clearance is crossed, or softens the impact and redistributes the energy to lower frequencies³⁴.

Although both journal and needle bearings can produce impacts, needle bearings differ from plain bearings in that: the bearing surfaces are harder, there is more than one body involved in an impact, and a thinner film of lubricant is required. In addition, individual needles may skew, then the shaft load is applied in the center of the needle, and only the ends of the needle contact the outer race. This can reduce the stiffness of the bearing, and also causes binding.

An estimate of the frequencies affected by bearing impacts is difficult. The natural frequency of the impact is related to the masses involved. A gudgeon pin bearing impact involves the piston and the connecting rod, which have similar masses. For impacts normal to the longitudinal axis of the connecting rod, the connecting rod can rotate about the crankshaft and the effective mass of the connecting rod is reduced by a factor of $(r/L_c)^2$, where r is the radius of gyration of the connecting rod, and L_c is the length of the connecting rod. The natural frequency (f_n) of such an impact is given by:

$$f_n = \frac{1}{2\pi} \frac{L_c}{r} \sqrt{\frac{K \left(\left(\frac{r}{L_c} \right)^2 m_c + m \right)}{m_c m}} \quad 60$$

where K is the stiffness at the point of impact, and m is the effective mass of the piston or crankshaft.

The term $(r/L_c)^2 m_c$ can be ignored in equation 60. Thus the natural frequency is determined by the connecting rod mass and is moved to higher frequencies by the ratio (L_c/r) .

The impacts are not necessarily in a direction normal to the longitudinal axis of the connecting rod. At the small end bearings, if the impact is along the axis of the connecting rod, the piston is moving against the combined mass of connecting rod and crankshaft. At the big end bearings, the piston and connecting rod move against the crankshaft. The natural frequency of these impacts would be about 3 times lower than that given by equation 60 (using values appropriate for the pulpwood saw). Thus assuming a sinusoidal impact pulse, the bearing noise can be associated with frequencies spanning a bandwidth of over 2.5 octaves, depending on the direction of the impact.

Since the chain saw runs at high speed, bearing noise should be significant, making identification easier. On the pulpwood saw, it is likely that bearing noise can be increased by increasing the bearing clearance. While this should be the most effective method of identifying bearing noise, the number of impacts may also increase; which could affect the distribution of bearing noise frequencies. Thus without more detailed calculations, it may be difficult to identify bearing frequencies, and the levels of bearing noise using increased clearances.

2.7.2. Zero clearance bearings

The main crankshaft bearings of the pulpwood saw are ball bearings. Roller, or ball bearings with preload can be operated at effectively zero clearance. Thus contact between the rolling elements and races is always maintained, making impacts due to contact loss unlikely. Because of the lack of clearance between elements, imperfections due to damage or contaminants cause small impacts or vibration to occur each time a rolling element contacts a defect. Thus an increase in noise could indicate an impending bearing failure.

Due to the lack of slippage between bearing components, the impacts can occur with well-defined frequencies related to the bearing geometry. The bearings are similar to a planetary gear system, thus the impact frequency is not normally harmonically related to the shaft frequency. Defects in the inner race, outer race, or rolling elements, will produce unique frequencies that can be identified with narrow band or digital analysis. Monitoring a machine for slight changes or abnormally high levels at bearing related frequencies, allows preventive maintenance to be performed¹²¹.

Unless there are defects or contaminants in the main bearings of the pulpwood saw, it is unlikely that the main bearings will make a significant contribution to the chain saw noise.

2.8. Chain noise

The noise of the cutting chain has been measured at levels comparable to exhaust, intake, and fan noise^{24,29,90}. Case studies have measured the noise at 91, 92, or 96 dB^{23,91,24} (A-weighted) at the operator's ear. The noise due to the cutting process was also found to be insignificant compared to other sources⁹².

The saw chain used for cutting the wood is a cross between a roller chain and a silent chain (see fig. 2.8). The chain is held by tension around a stationary rigid guide bar along its entire length. It is similar to a single strand roller chain (like a bicycle chain), except

that the rollers and inner link plates are replaced by a single (roughly triangular) inner drive link plate. This is similar to a silent chain, in that the drive sprocket does not extend through the chain, but enlarged links on the inner surface of the chain, mesh into the sprocket. The drive link plates slide inside a groove in the edge of the guide bar to prevent lateral movement of the chain. The final components of the chain are cutting teeth (cutters) formed by extended side link plates on one in four side link plates.

Noise can be produced when the chain impacts against the drive sprocket, or guide bar. The guide bar impacts can occur upon initial contact, and by slapping along the length of the bar. The metal to metal scraping contact between guide bar and chain, although lubricated with chain oil, will also produce noise. A further noise source, is the cutting teeth along the chain, that periodically interact with the wood.

A saw chain, although similar to a silent chain, has the noise characteristics of a roller chain. A silent chain has design improvements not found in roller chains or saw chains (fig. 2.8). The reduced noise of silent chains, is due to the use of articulated joints instead of pin joints, and two part drive links, that expand in a scissor like fashion into the sprocket. Compared to pin jointed chains, this produces smoother meshing of chain and sprocket, allowing higher speed and power capacity, as well as reduced noise^{93,94} (hence the name silent chain).

Experiments have shown that the most significant noise producing mechanism in roller chains, is the noise occurring due to the impacts between chain rollers and sprocket teeth^{95,96}. The impacts are due to fluctuating forces arising from so called polygonal (or chordal) action. When the chain is wrapped around the sprocket, the rigid links form straight line segments, with vertices located at the joints between each link. As shown in fig. 2.9, by extending the straight line segments around the sprocket, a closed polygon is formed (dotted lines), explaining the origin of the term polygonal action. Note, that on a saw chain, every second link is a drive link. Thus a 7 tooth sprocket, would be represented by a 14 sided polygon (double that shown in fig. 2.9).

Chain impacts can be visualized using figure 2.9. In the figure, the chain is represented by the heavy dark lines, and the chain joints are shown as circles. The chain is wrapped partially around a sprocket (shaded). The chain pitch (L_p) and pitch radius (R_p) are also shown in fig. 2.9. The source of the impacts is seen by noting that the chain moves radially as the sprocket rotates. As the sprocket rotates clockwise through an angle θ , joint a at position R_a (in contact with the sprocket) moves to R_a' , following the path

described by the dashed line. For a long, slow moving chain, joint **b** at position R_b will precisely mimic the movement of joint **a**. The chain impacts the sprocket when joint **b** reaches $R_{b'}$. The point of impact on the sprocket is **c**, located at position $R_{c'}$, which has rotated from its original position at R_c .

The relative velocity between sprocket and chain (when joint **b** contacts the sprocket at $R_{b'}$) determines the energy transferred. This is shown by the vector diagram in figure 2.9. The magnitude of the velocity (V_a) of joint **a**, is the same as that of the velocity (V_b) of joint **b**, since they are connected by a rigid link. The velocity (V_c) of point **c** on the outer edge of the sprocket is approximately the same (ignoring the small difference between sprocket radius and R_p). Thus the magnitudes of the velocities are given by⁹⁷:

$$|V_c| = |V_a| = |V_b| = 2\pi n R_p \quad 61$$

The normal impact velocity will be given by the vector difference between the speed of point **c** on the sprocket and rivet **b**. As shown in the vector diagram in figure 2.9, the magnitude of the impact velocity (V_i) is⁹⁷:

$$|V_i| = |V_c| \sin \theta + |V_b| \sin \theta \quad 62$$

Substituting from eqn. 61, and noting that chain pitch (L_p) is given by $2R_p \sin \theta$, the normal impact velocity reduces to:

$$|V_i| = 2\pi n L_p \quad 63$$

The kinetic energy of the chain that may be transferred to the sprocket also depends on the effective mass of the impacting body (joint **b**). Although figure 2.9 shows the chain moving as a rigid body, the flexibility of the chain must be taken into account. Using Lagrange's equations, Chew⁹⁷ has shown the chain flexes locally, and, in theory, the effective impact mass is equivalent to that of only one or two links, and depends on the ratio of roller mass and side plate mass. For most roller chains, the effective mass decreases with increasing length to that of a single roller. Similar results were found by Ryabov⁹⁸, with theory indicating that the effective mass increases with length to a limit of 1.67 times a roller mass. The effect of friction in the joints of the chain was expected to increase the effective mass slightly. Ryabov describes experimental results⁹⁸ that indicate an effective mass equivalent to two links.

Impact noise analysis can be used to determine the noise produced by chain impacts. Using equation 63 for a 9.5 mm (3/8") pitch chain with the sprocket rotating at 7500 rpm, the impact velocity is 7.5 m/s. Assuming the equivalent mass of the impacting chain is approximated by two chain links, (i.e., a drive link, two side plates, and a connecting pin), Richards' analysis (section 2.5 on impact noise) can be applied to obtain a rough estimate for the noise due to sprocket and chain impacts.

The impact frequencies can be estimated from published data on a roller chain. The peak narrow-band noise experimentally measured and associated with roller and sprocket tooth impacts on a 25 mm pitch chain, occurred with a flat narrow band spectrum between 4 kHz and 12 kHz⁹⁵. Thus the lower limiting frequency of the force derivative spectrum should occur around 12 kHz. For a 9.5 mm saw chain, an initial assumption of geometric similarity with the 25 mm pitch roller chain, based on link dimensions, results in the high frequency roll off occurring 8/3 times higher (see section 2.6.4, piston slap force spectrum). The rim drive sprocket impacts against side plates that have an area say 4 times larger than a roller chain roller. Resulting in the natural frequency being increased by a factor proportional to the root of the impact area ratio. Thus the lower limiting frequency of the force derivative would occur at 64 kHz (i.e., $(8/3)\sqrt{4}$ times 12 kHz which allows for impact area and smaller chain pitch). Thus the chain noise, due to polygonal action, would be expected to increase with frequency beyond the audible range.

Substituting these values into equation 23 to obtain the vibratory power, and using equation 21 to obtain the radiated noise, indicates the spectrum should produce 1/3 octave band levels of 95 dB at all frequencies between 1.2 kHz, and 64 kHz. The corresponding total A-weighted noise due to polygonal action is about 106 dB. This suggests polygonal action may dominate the noise of chain and cutting. However, the A-weighted level exceeds measurements of 91 or 96 dB as reported by other researchers^{24,23}. More significantly, the level equals the maximum noise the chain saw should produce (106 dB A-weighted). Thus the calculated levels significantly overestimate the noise due to polygonal action.

Note that additional factors can reduce the above estimate of noise. Friction between the rim drive sprocket and chain links can reduce the impact velocity, while chain lubricating oil can also reduce the affected frequencies³⁴ as well as the impact velocity⁹⁵. The impact energy must travel across the interface from the sprocket to the clutch, and crankshaft to reach the large radiating surfaces. Thus the coupling between impact and radiating surface may be poor.

Fluctuating speed and tension in the chain also can affect the noise. It has been suggested from experimental observations that fluctuating engine torque can increase the transverse vibration of the chain and magnify the chain and sprocket impact noise⁹⁹. A theoretical study has shown a non-linear coupling between longitudinal and transverse vibrations of the chain⁹⁹. Thus the torque fluctuations in the chain drive can magnify the chain and sprocket impact energy. The speed and torque fluctuations and the resulting chain movement would produce additional impacts between the chain and guide bar, leading to additional noise.

The speed of the chain and the torque on the sprocket are directly dependent on the effective sprocket radius. The effective sprocket radius changes due to polygonal action. Referring to figure 2.9 the change in effective radius is:

$$\Delta R_p = R_p(1 - \cos(\theta)) \quad 64$$

The pulpwood saw has a seven tooth sprocket with two chain links per sprocket tooth: thus the chain would form a 14 segment polygon around the sprocket (instead of 7 segment as shown in fig. 2.9). Using equation 64 with $2\theta = 2\pi/14$, the percentage change in the torque arm (or effective pitch circle radius), is 2.5%, varying from high to low 14 times per revolution. Thus the saw chain speed and torque are expected to vary 2.5% solely due to polygonal action. For a 9.5 mm pitch chain, with 7 tooth sprocket operating at 7500 rpm, the chain speed is 16.7 ms^{-1} (60 kph), resulting in a speed variation of about 0.4 ms^{-1} at 1750 Hz. A velocity of 0.4 ms^{-1} is twice that calculated for the piston slap impact velocity. This speed variation can therefore cause additional interactions with the guide bar.

As the saw chain is guided along its entire length by the guide bar, any transverse movement of the chain will produce impacts. The momentum of the chain links as they leave the sprocket may cause them to separate from the guide bar. Changing the chain speed or tension can also cause the chain to slap against the guide bar. For lack of a better estimate, the impact between chain link and bar can be assumed to be similar to that of an impact with the sprocket. Thus as a first approximation, the spectrum of the noise produced would be the same for impacts of the chain with the sprocket or bar.

Case studies of roller chains provide additional information on noise producing mechanisms. Lightly lubricated chains have shown that individual impact amplitudes vary approximately 20 dB per decade increase in speed^{96,100} (a somewhat larger effect of speed

was noted with chains lubricated at 3 liters oil per minute¹⁰⁰). The impact rate would likely add another 10 dB of noise per decade speed increase, resulting in a 30 dB noise increase per decade speed increase (for lightly lubricated chains). Tension had little effect on impact intensity⁹⁶. A roller chain was found to vibrate transversely (similar to a heavy bar with fixed ends⁹⁵) along its unsupported length. This directly affects noise through additional impacts between the chain and chain guides.

One study of saw chain vibration examined the effects on hand arm vibration at frequencies ranging from 6.3 Hz to 1.6 kHz. Chain sharpness was found to have an effect on the vibration along the axis of the bar¹⁰¹, and normal to the plane of cutting¹⁰². The use of a slightly dull chain (cutting rate 10% lower than obtained with a sharp chain) caused the vibration at the engine firing frequency to increase along the bar axis by 6 dB^{101,102} (resulting in a total increase of 3 dB). The effect on vibration velocity (related to hand arm vibration hazard) below 1.6 kHz, was found to be more significant than the effect on vibration acceleration (related to radiated noise)¹⁰¹, suggesting that the sharpness mainly affects low frequencies and may not have a significant effect on noise.

Chapter III Apparatus and Chain Saw Modifications

The chain saw was modified in numerous ways to suit the purposes of experiments conducted in the field and in the laboratory. In this chapter these modifications are described in detail, together with other apparatus developed for the experiments.

3.1. Apparatus and test site for field measurements

3.1.1. Test site

The outdoor field test site was located at the National Research Council, Montreal Rd. campus on a small asphalt parking lot, (fig. 3.1). It provided a free field above a horizontal reflecting plane covered directly under the saw with up to 3 cm of freshly accumulated sawdust. The only acoustically reflecting landscape feature within 50 m was the laboratory that sheltered the measuring equipment. To avoid reflections from the laboratory building, the test site was located on a diagonal to the corner of the building. Background noise levels (, fig. 3.3) were influenced by light traffic on nearby roads, normal environmental noise, and machinery operated at other locations in the Council.

3.1.2. Acoustic enclosure for chain saw

Separation of the noise associated with the cutting process from that radiated by the surfaces of the machine, was achieved by placing the saw in an acoustic enclosure, (fig. 3.2). The enclosure permitted an operator to cross-cut wood as in other field experiments, yet it reduced the noise in the near and far field from the powerhead surfaces. It was constructed from an aluminum frame to which the saw was compliantly attached (isolators just visible under ends of saw in fig. 3.2). The walls were fabricated from sheet aluminum with an inner lining of fiberglass (13 mm thick). Access to the rear handle and engine controls was provided by a felt-lined rubber boot, while the cutting components (chain and its supporting plate, or "guide bar") protruded outside the enclosure through a rudimentary lined duct. A similar lined duct (long black rectangle visible in lower middle section of enclosure in fig. 3.2) allowed fresh air to enter for combustion and cooling. Exhaust and intake noise were also eliminated by ducting (using exhaust tubing, as described later).

The acoustical enclosure for the chain saw provides a substantial noise reduction at frequencies above 250 Hz. The effectiveness of the enclosure is shown in figure 3.3. A point source was located at the normal position of the powerhead when cutting wood at the field cutting site. Measured sound pressure levels at the operator's ear (°) and at 5 m

(⊙) are shown in figure 3.3. Next, the point source was placed in the chain saw acoustical enclosure, and the measurement repeated. The resulting sound pressure levels at the operator's ear (⊙), and at 5 m (⊙) are reduced to the levels expected for background environmental noise (). Although results will differ somewhat when the enclosure is used with a chain saw, the enclosure is highly effective at frequencies of interest (above 500 Hz).

3.2. Apparatus for laboratory measurements

To investigate the chain saw noise more thoroughly, a controlled laboratory environment was required to eliminate variations due to changes in engine speed, load, and source-receiver geometry. The laboratory also removed the necessity to cut wood. Modifications to the saw or operating conditions that would be unsafe, or impractical in the field, became possible: the human operator was replaced by a mechanical test stand. The apparatus designed and constructed to fulfill these special requirements is described here.

3.2.1. Test stand

A test stand (fig. 3.4) was designed to support the saw in the same manner and position used when a human operator cross cuts wood. The saw was compliantly attached to a rigid space frame by its front and rear handles, 0.9 m above a concrete floor, and was oriented to simulate the position when hand held for cross cutting wood. The stand was normally used in the laboratory in a permanent setup.

The test stand features a space frame construction, welded from solid round steel bars 2.5 to 5 cm in diameter, with total weight of approximately 260 kg. It provided a work space around the saw relatively free of acoustically reflecting or radiating surfaces. Vibration input from the saw was attenuated by the mass and stiffness of the stand. The test stand provided attachment points for the principal components: the saw, dynamometer, microphones, and accessories. An external gas tank was mounted on the test stand to provide gravity-fed fuel through 0.6 cm O.D. neoprene tubing. This prevented the handle mass from varying as the gas was depleted, and simplified refueling the saw. Provision was made to allow wood to be fed into the chain during testing, and the necessity to cut wood was eliminated by using a dynamometer.

3.2.1.1. Dynamometer speed and load control

Early cutting tests with the chain saw showed that engine speed had the most significant effect on noise, and cutting wood was not necessary. Consequently, a dynamometer was used to load the saw. The dynamometer allowed unlimited test duration, precise control of load and speed, and eliminated the need for the bar and chain. The dynamometer was mounted directly under the saw and attached to the test stand by vibration isolators (fig. 3.4).

A belt drive connected the chain saw to the dynamometer; this allowed the dynamometer to be mounted remotely on the test stand, well away from the chain saw measurements. The belt drive required several modifications to the chain saw itself, and did not significantly increase the stiffness of the mounting. By mounting the dynamometer directly under the saw, the belt tension force on the chain saw isolators is vertical. When the pulpwood saw operates at full load the total belt tension is approximately 100 N. This is similar to the force applied when the chain is cross cutting wood, from the bottom up, (opposite to cutting direction used in field testing).

The dynamometer was driven with a 1.2 cm wide V-belt from a clutch-mounted pulley. The drive ratio was approximately 1.17:1 using a 7.6 cm diameter pulley on the chain saw and an 8.9 cm diameter pulley on the dynamometer. There was 56 cm between pulley centers, and idler wheels were mounted on the test stand to reduce play in the belt. As seen in figure 3.4, an expanded steel mesh enclosed the idler wheels, to prevent measurement cables from becoming entangled in the chain saw.

The dynamometer was instrumented for control of speed and load. Speed was indirectly sensed using a coil encircling the spark plug cable, which was used to trigger a digital one shot. Visual display of speed was by a Hewlett Packard counter model 5314A, and could be monitored continuously by a data acquisition system (fig. 3.5). Energy absorbed by the dynamometer was dissipated by a resistive load, consisting of a heater bank mounted at 2 meter height near the room exhaust fan outlet (fig. 3.8).

3.2.1.2. Hand models

The chain saw mounting attachment was comparable to the hands of a human operator. The mounts, (fig. 3.6), consisted of three-stage dynamical models of the mechanical input impedance of the hand-arm system, representing the human hand-arm system from the

hand to the elbow. The masses and compliances used, were based on values found in the literature^{103,104}.

In view of the spatial requirements of the chain saw support, an anatomical model of the bones and flesh of the human hand arm system was considered most appropriate for test purposes. This would preserve similarity between field and laboratory testing in terms of reflections, absorption, and positioning of measuring microphones. Conversely, a mechanical model constructed from appropriate springs, dashpots and masses would simplify construction, and allow the use of parameters taken from the literature. For this reason, a hybrid model that preserved the spatial relations of the human hand arm system was constructed by using masses and compliances loosely based on measurements in the literature. The model used is shown in figure 3.6.

The surface of the hand model support could have been wrapped to simulate the acoustical properties of a human operator. However, selection of the surface properties is difficult. In practice, the surface properties could vary from bare flesh to a down filled jacket. The relative position of human operator and saw is not constant, adding yet another variable. Thus, in keeping with the space frame construction of the test stand, the hand models were designed without covering in order to reduce the surface area.

As detailed measurements were not made of the impedance of the human hand arm system, the hand models were designed to allow flexibility in the selection of masses and compliances. The wrist and elbow joints employed commercial vibration isolators to allow the compliance to be changed with ease. Although not used in testing, the compliance of the palm of the hand model could be varied by using a moldable silicone rubber compound with a stiffness that could be changed using an additive thinner, or through appropriate shaping of the mold. Similarly, the masses used could be varied by drilling holes, or adding extra material.

The first stage of the handle mount was a model of the flesh of the hand (thick black annulus at top right of fig. 3.6). It consisted of a 12 cm length of 1.2 cm wall high pressure rubber hose, split lengthwise to surround the handle in two sections. The mass of the second stage was 0.45 kg. (comparable to that of the hand), and consisted of an aluminum tube loosely clamping the rubber of the first stage. The second stage was attached by four Barry brand commercial vibration isolators, (each with stiffness 5×10^4 Nm^{-1} , shown in black, or thick crosshatch in fig. 3.6), to the third and final stage, this provided a compliant suspension similar to that of the wrist. Continuing the analogy to

the human arm, the third stage used a 2.5×2.0 cm solid aluminum bar of approximately 20 cm length, with attachments for vibration isolators at each end.

The initial two stage assembly was found sufficient for experimental purposes. The elbow joint was not used; it was bolted directly to the test stand. No further attempt was made to optimize the dynamic response of the mounts, as the first two stages were found to represent adequately the load impedance experienced by the powerhead at frequencies above 250 Hz. As measurements were made in the laboratory, and there is little noise radiated by the powerhead below 500 Hz, the use of only two stages was appropriate. This increased safety as the chain saw was held more rigidly.

3.2.1.3. Test stand performance

Use of the test stand allowed significant improvements in chain saw performance in terms of safety, repeatability, and consistency of test conditions. The saw could be warmed up and operated indefinitely without tiring the operator. Under operation without load and with the dynamometer disconnected, the test stand support allowed precise throttle control unattainable with a single human operator. This allowed the speed to be controlled to better than ± 150 rpm. Additional improvements in speed control were possible by using the dynamometer, which allowed the chain saw speed to be governed within ± 100 rpm using a custom made controller¹⁰⁵. The power absorbed was calculated from the torque, which was monitored using a piston with a fluid coupling to a Sensym hydraulic load cell, and calibrated by adding weights to an arm attached to the dynamometer. Calibration and repeatability of the load measurement were better than ± 75 W (for operation at 7500 rpm, with the error directly proportional to engine speed).

The influence of the test stand and dynamometer belt drive on saw vibration can be inferred from Fig. 3.7a. Differences between three orthogonal component accelerations recorded with the saw first hand held, and then operated on the test stand, are shown in the upper part of the diagram, measured at an arbitrary position on the powerhead, (close to a handle isolator). Measurements were made with an arbitrary vibration source (idling engine), so that the comparison could be between the mounting methods (human or model hands), independent of loading condition. It can be seen from Fig. 3.7a that the attachments to the saw when it is mounted on the dynamometer (the hand models, the belt drive and exhaust and intake pipes) together appear to introduce errors of +1 to -2 dB in the frequency range 250 to 4 000 Hz. As these differences are within the range of test to test variation, it is not possible to deduce changes attributable to the belt drive and hand

models. At higher frequencies, a systematic reduction in vibration occurs when the saw is suspended from the test stand. The high frequency change may show the improvement due to laboratory measurements, as small changes in engine idle speed would cause the clutch to engage momentarily.

In practice the grip of a human operator can change significantly, in the extreme case, the operator can actually release the front handle when cross cutting with the saw. Besides anatomical differences between operators, any variety of padded gloves could be used.

An example of the effect of the operator's grip was found in tests that determined the effect of the accelerometer mass on handle acceleration¹⁰⁶. These tests allowed for verification of the effectiveness of the hand models, and resulted in a proposal to change ISO 7505⁵³. In theory, a 50 g accelerometer mounted on the chain saw handle should affect its vibration. In testing for this effect, the operator attempted to maintain a constant grip on a chain saw and did not change either his grip or position during the time the accelerometer was changed. These tests with the human operator and similar tests on the test stand showed that accelerometer mass could affect the 1/3 octave band handle vibration by up to 6 dB at frequencies around 500 Hz. Despite the magnitude of this effect, it had never been demonstrated before, likely due to the variable effect of the operator's grip.

Thus the changes in vibration in figure 3.7a comparing test stand and human support are negligible. Insufficient damping of the lower modes of handle vibration¹⁰⁶ was the most notable difference found between the hand models and the human hand arm system. Note that due to the vibration isolation between the handles and the powerhead, small differences induced by the hand models should have a negligible effect on the noise or vibration of the powerhead.

3.2.2. Laboratory environment

Laboratory measurements were performed in a large, semi-reverberant room (4.33 x 3.89 x 4.0 m), with the saw suspended from the test stand, (fig. 3.8). These measurements differ from field measurements due to additional reflections from room surfaces, and lack of reflections from the operator, or wood. The environment will influence the absolute levels, but will not significantly affect the source ranking. To make the measurements more similar to the hemi-anechoic field tests, absorptive material was placed in the room to reduce reverberation and direct reflections from the walls. Two acoustically absorbing, felt covered pyramids lined with 5 cm thick fiberglass, each of 2.4 square meter area, were

suspended between the chain saw and the ceiling. Three additional pyramid absorbers were placed on the walls. A 5 cm thick acoustical foam sheet with dimensions of 1.8 x 3.5 m was hung over the remaining wall. In addition, an unused roll of acoustical foam was also standing on end in the room. The acoustical absorbers reduced direct reflections from room surfaces by up to 20 dB, and reduced the level of reverberation by 10 dB.

An indication of the effectiveness of these changes in terms of the deviation from free field propagation is shown in fig. 3.9. To obtain these results, the saw was removed from its mounts and replaced by a point source consisting of a loudspeaker attached to a 0.59 m tube of diameter 2.5 cm. This had two effects: to reduce reflections from the body of the speaker driver, and to make the point source frequency response similar to that of the chain saw above 160 Hz. The sound pressure level relative to that measured at 38 cm from the source was then plotted for several different microphone positions. For comparison the predicted decrease in sound pressure level for free field propagation, is shown by straight lines. Increased sound levels recorded at 53 cm from the point source (at 1, 2, and 8 kHz) are likely due to the proximity of the microphone to the saw support structure (test stand). The increased levels at 106 cm are expected, as reverberation in the room and reflections from the ceiling were beginning to affect the measurement. The reference microphone position used for measurements is marked 75 cm (or ear), representing the operator ear position.

At the operator's ear position, these measurements suggest that the deviation from free field propagation will be no more than ± 1.5 dB for any 1/3 octave band from 125 Hz to 12.5 kHz. The reduction of background noise and direct reflections in the room permitted intensity measurements to be made above 200 Hz for a saw to microphone separation of up to 50 cm, and above 500 Hz for a separation of 50 to 130 cm. The difference in level between sound pressure and intensity, an indicator of the errors associated with reverberant sound fields, was less than 8 dB at most frequencies in the range 315 to 10000 Hz.^{107,108}

The effect of the test stand support and laboratory acoustical environment is summarized by figure 3.10. Differences exist between field and laboratory measurements, but they are acceptable. The laboratory environment is preferable to field measurements for source identification due to increased stability and convenience.

The acoustical environment in the laboratory increases the overall A-weighted sound pressure levels an average of 2 dB compared to field measurements. The shaded area in

figure 3.10 represents the upper and lower limits of 6 measurements of chain saw mechanical noise in the field versus a measurement in the laboratory on the test stand. Both tests were made with the saw in the laboratory reference condition, except the bar and cutting chain were reattached. Positive levels in this figure indicate field measurements exceed those taken in the laboratory. In general, the shaded area indicates field measurements are somewhat lower in level than laboratory measurements. Due to the large variation in field measurements this difference is not very significant. However, this change is similar to the ± 1.5 dB variation between laboratory measurements and free field propagation seen in figure 3.9.

Although variations of up to 7 dB are seen in individual 1/3 octave bands the corresponding changes in the overall A-weighted levels in the field were ± 1.6 dB. The variation in overall level is slightly larger than found with the stock saw. Causes for differences noted in the field measurements are likely due to positioning of the operator, saw, and microphone. These differences would also be affected by the source characteristics.

A significant cause of differences between laboratory and field measurements is the chain saw operator. Figure 3.10 (∇) shows increases in sound pressure levels solely due to the operator. To obtain this curve, the microphone and chain saw were kept in the same position in the laboratory. Support was provided by first the human operator, then by the test stand. The resulting level differences found are similar to those when comparing field measurements to laboratory measurements (shaded area). Reflections from the operator are the most likely cause of the level differences. Note that the only frequencies where these measurements (∇) differ significantly from field measurements (shaded area) occur between 500 and 2000 Hz. This was also the region where the laboratory differed most from a free field propagation in figure 3.9.

As demonstrated by the shaded area in figure 3.10, (see also section 3.2.1.2 on hand model design), the presence of the operator can cause significant variability in the measured chain saw noise. A mannequin with appropriate acoustical properties was considered for attachment to the test stand. However, the difference between laboratory and field tests was small enough for the purposes of source separation and ranking, and additional modifications to laboratory testing were deemed unnecessary.

One additional factor that will affect differences between laboratory and field measurements, is the directivity and frequency response of the microphones used. This

could enhance differences in the acoustical environment. For example, when switching between microphones in the laboratory, at frequencies above 5 kHz the microphone normally used in the laboratory produced about 3 dB less output than the microphone normally used for field testing.

Noxious fumes that might escape from the chain saw required the installation of a large air moving system for safe use of the laboratory measurement facility. Fresh air supply to the room was obtained from the building roof, and could be heated to maintain a constant temperature. At audible frequencies, the noise from the fan system was under 50 dB, which was comparable to field testing, and below measurable levels during chain saw operation.

3.3. Chain saw modifications

To provide a consistent baseline measurement, the same pulpwood chain saw was used for almost all laboratory testing. This saw was modified as required for each test (see fig. 3.11). If replacement of parts of the chain saw was required, the number of parts changed was kept to a minimum. When these changes were no longer required, the chain saw was returned to its reference condition using the original parts.

To enable the use of selective source enhancement (or reduction) in the laboratory, some components were removed from the saw. A "reference condition" was defined that did not involve components used to cut wood. Thus, parts associated with the cutting chain were removed to reduce the number of variable factors in the measurements. The centrifugal clutch, oil pump, and oil pump drive gear, were replaced with dummy parts matching in size, and weight (fig. 3.11).

The reference condition also differed from the stock condition as supplied by the manufacturer, because of several minor modifications to reduce the noise from non essential components. The clutch cover was removed to reduce rattling and allow direct radiation of sound from drive clutch, crankcase, and bearings. The starter spring rattling was reduced by injecting a thick grease into the spring housing. Rubber pads were attached to the starter pawls to prevent metal to metal impact with the flywheel after disengaging from the starter. As already noted, gas was fed directly to the carburetor from an external reservoir, allowing the gas tank in the handle to be filled and kept at constant mass (maintaining a constant mechanical impedance at the saw vibration isolators).

In the reference condition an additional aluminum gasket was inserted between cylinder and chain saw. To allow testing of the saw with vibration isolation of the cylinder, sufficient metal was removed to insert a rubber material between the cylinder and crankcase. As vibration isolation was not desirable in the reference condition, the rubber material was normally replaced by a shaped aluminum gasket.

The effectiveness of the modifications in reduction of the reference condition mechanical noise is seen in figure 3.10 (◆). This figure shows the difference between the mechanical noise of the stock condition while cutting with intake and exhaust ducted, versus the noise in the reference condition (also cutting). The reference condition is approximately 7 dB quieter at all frequencies. Much of this difference can be attributed to modifications to the intake system, discussed later in section 3.3.2.

There was no doubt that the 7 dB change in noise levels between stock and reference condition had a significant effect on the source ranking of the chain saw noise. However, the reference condition was the appropriate condition to study for a noise source ranking. This condition approximated the way the saw would have been made, if sources such as the exhaust were quieter. The modifications were of a sort that could have been incorporated in the original design without significant changes in design, materials, or components. To use a common analogy of a chain that is only as strong as its weakest link, the intent of changes to the stock saw was only to replace the weak links, not the chain itself. The only exceptions to this rule, were modifications that were intended to eliminate sources that were accounted for separately (i.e., removal of the chain, or use of a dynamometer). Detailed descriptions of changes to the saw are discussed in the following subsections.

3.3.1. Chain saw belt-drive attachments

Loading the chain saw with the dynamometer belt drive required some modifications to the chain saw itself; the principal change being, the addition of a pulley on the clutch. Two separate drive systems were provided for the dynamometer. Normally, the chain was removed, and the chain drive sprocket replaced with a 7.6 cm diameter steel pulley (fig. 3.11). If necessary, the belt could exit through an existing hole in the clutch cover of the chain saw. This system was used when measurements were made without the saw chain. When operation with a chain was required, the pulley was mounted on the other side of the clutch drum. For safety, the cutting teeth of the chain were filed off. The clutch cover

panel, which was not used in reference measurements, could not be installed when chain and pulley were used together.

When the dynamometer was not required, the belt and pulley attachments were removed and replaced with a dummy clutch made of brass that matched the size and weight of the original clutch. The dummy clutch was rigidly attached to the crankshaft, and eliminated noise associated with the clutch shoes and clutch bearings. This was the reference mode of operation, with no engine load, as the dummy clutch increased safety by eliminating the possibility of the clutch shedding components during high-speed operation.

3.3.2. Intake and exhaust modifications

It is common for the intake and exhaust to make significant contributions to the overall sound power of an engine. It was necessary to eliminate both the intake and exhaust noise to identify other sources.

In field experiments, the intake and exhaust noise was reduced by using flexible (spiral wound) automotive exhaust tubing (fig. 3.12). The tubing was at least 6 m long and placed well away from the measuring microphones. The tubing for the exhaust terminated in an automotive muffler. Laboratory experiments, indicated that the tubing could provide 30 dB to 45 dB of attenuation at the frequencies above 500 Hz. The effectiveness of the tubing under operating conditions is apparent from figure 5.2b (—). In this curve, intake and exhaust noise was silenced using flexible tubing. This reduced the A-weighted noise of the exhaust and intake below 70 dB in all frequency bands, sufficient for testing purposes.

In laboratory measurements, additional measures were taken in order to quiet the intake and exhaust. For reference measurements, the intake was quieted to allow for measurement of the sound radiated from adjacent engine surfaces. Intake air was allowed to flow through the original intake system. This allowed the use of an intake air filter, and eased operation of the saw compared with field measurements. The intake inlet was sealed and replaced with a 15 cm long brass pipe with 12 cm internal diameter. The brass pipe was connected to 4.2m length of flexible automotive exhaust tubing that drew fresh air from the building exterior.

Even with these modifications, it was possible by using acoustic intensity (fig. 3.13), to detect slight changes in noise level from the crankcase and intake system. Great care was required to find and seal any small leaks, as the acoustic energy emitted could interfere

with the measurement and identification of other sources. Any small holes, or potential leaks were sealed by using hot melt glue, silicone rubber or epoxy. The noise emitted from leaks or holes was thereby reduced below the level transmitted through the intake surfaces.

Further modifications were made to improve the transmission loss through the surfaces of the intake system. The plastic intake system walls were covered with a 0.3 cm thickness of a lead-like material (Cerro bend) cast to match the machine contours (see gray metal visible in fig. 4.3, 4.4). The throttle and choke controls were extended from the carburetor directly through the side of the saw in order to simplify the operation and sealing of the intake wall. The carburetor was originally connected to the crankcase with a thin rubber tube. This was changed to three concentric brass tubes compliantly connected using o-rings to increase the transmission loss, while allowing 6 degrees of freedom of movement.

The effectiveness of these measures was checked by acoustical excitation of the crankcase by an external acoustical horn driver. The piston was removed and the source attached to the cylinder exhaust port. In this fashion, any leaks could be detected and sealed.

Using the external horn driver, the radiated noise was measured with an open intake, fully modified intake (no duct on the intake), and stock saw, respectively. The results are plotted as a level difference re open intake in fig. 3.14. Although the speaker used to drive the chain saw produced sound levels over 140 dB in the crankcase, the noise was strongly attenuated before reaching the intake, and tended to be obscured by background noise in the room. Thus the higher insertion loss values show the maximum dynamic range of the measurement, and underestimate the actual intake insertion loss. It can be seen that the intake system modifications (✚ fig. 3.14) significantly improve the insertion loss provided by the stock intake muffler ✚. These modifications eliminated all evidence of intake noise during saw operation.

During most laboratory measurements, the exhaust was ducted through the building wall through the use of brass tubing, which was vibration isolated from the saw, wall and floor. The tubing terminated in an automobile muffler which was located outside the building. The connection to the chain saw used a separate brass tube compliantly connected at each end with o-rings, which allowed rotation and translation in 3 axes. To prevent o-ring failure due to overheating an external "muffin" fan was used to cool the tube.

The attachment of flexible exhaust and intake pipes to the saw, which allowed for the measurement of sound radiated from adjacent engine surfaces, appears to influence the vibration of the powerhead at frequencies between 250 and 4 000 Hz (Fig. 3.7b). Differences between three orthogonal component accelerations recorded with flexible exhaust and intake pipes attached and then removed are shown in this diagram. It can be seen from Fig. 3.7b that the exhaust and intake pipes typically introduce errors of +1 to -2 dB in the vibration of the crankcase (between 63 and 12500 Hz).

The effect on engine power of the ducting of exhaust and intake was found to be a loss of 2% and 5% respectively (relative to the stock saw). In a separate test using sound intensity, with and without intake ducting, no identifiable difference was found in sound energy radiated by saw surfaces above 500 Hz.

The modifications described above were necessary to permit measurement and diagnostic tests of the other sources of chain saw noise.

3.3.3. Bearings

Noise source identification by selective source enhancement or reduction requires a controlled change in system parameters. Changes in clearance in the bearings may indicate the relative level of bearing noise. Thus, the effect of clearance in the gudgeon pin needle bearings on chain saw noise, was determined by changing the gudgeon pin diameter (figure 3.11). The gudgeon pin needle bearings have a nominal clearance of 0.036 mm in diameter between the bearings and the hardened gudgeon pin. Four additional bearing clearances were tested by modifying a piston to accept a series of gudgeon pins. For ease in fabrication, each new gudgeon pin was 0.025 mm larger than standard, (fig. 3.11), and made of stainless steel, (instead of hardened steel). The bearing surface (center section) of each gudgeon pin was reduced in diameter to produce the desired bearing clearance. Five pins, which provided nominal clearances of 0.023 mm, original clearance (=0.036 mm), 0.086 mm, 0.137 mm, and 0.29 mm were produced. The bearing clearance could be varied, with the only modification being the installation of a new gudgeon pin.

In operation, the (new) modified gudgeon pins would lose their machining marks and become very smooth (as if polished), after 5 to 10 minutes of saw running. If the chain saw was in operation longer than 5 to 10 consecutive minutes, the bearing surface of the gudgeon pin would darken. Slight scoring indicated that the gudgeon pins had rotated inside the piston. Apart from an initial settling in period immediately after a gudgeon pin

was changed, the change of material from the original hardened steel to stainless steel did not noticeably affect the chain saw noise.

Modifications to the main bearings on the crankshaft were made to determine their effect on chain saw noise. A separate crankcase and crankshaft were assembled with non standard main bearings, and then this crankcase was switched with the crankcase on the test saw. The main bearings are 6203 C3 deep groove ball bearings, that are given an axial preload when the saw halves are assembled. Thus the bearings should have no free play in the axial or radial direction. The effect of bearing clearance was tested by running the saw with standard, C3, and C4 clearance group bearings. The specified bearing clearance for each of these groups respectively, is 3 to 18, 11 to 25, or 18 to 33 microns. Thus the effect of changing from a standard to a C4 bearing clearance, approximately doubles the clearance, although due to the preloading, the running clearance should still be essentially zero.

3.3.4. Fan

Fan noise, as will be shown later, is one of the principal sources remaining after the intake and exhaust noise have been reduced.

Suppression of the noise associated with the fan from that radiated by the surfaces of the machine, was achieved by replacing the fan with a "dummy" fan (fig. 3.11 and 3.15). The fan, flywheel, and magneto magnet are combined in one unit. In constructing the dummy fan, the fan blades were removed from the flywheel, and the surface filed smooth. The magnet and counterweight remained as two large blade like protrusions about 4 cm wide and rising 1 cm above the flywheel surface. Their effect on the airflow was reduced by attaching to them a smooth 1.6 mm thick aluminum disk, the same diameter as the flywheel (fig. 3.11). An external quiet "muffin" fan was provided in order to maintain operational temperatures in the saw.

Separation of the noise associated with the fan from that of the surfaces of the powerhead, was accomplished by externally motoring the fan, (fig. 3.16). In order to maintain similarity with other tests, the saw was mounted as usual on the test stand, but the piston and crankshaft were removed. The crankshaft was replaced with a straight shaft, stepped to fit the main bearings with the usual axial preload (fig. 3.11). Four small pulleys about 5 cm in diameter provided a range of fan speeds from 5600 rpm to 9000 rpm. The diameter of one pulley was reduced until it provided 7500 rpm (measured shaft rpm). The pulleys were driven by a 1.2 cm wide v-belt from a 20 cm diameter pulley attached to a 250 W

electric motor. The motor was mounted on the test stand directly under the chain saw, (fig. 3.16). To confirm that modifying the fan, or motoring the fan, did not increase the noise, the sound pressure levels of the dummy fan and normal fan were compared when driven by the electric motor.

The sound level reduction of the dummy fan is plotted in figure 3.17. In this figure the zero dB line represents 1/3 octave band levels equal to those produced by the normal fan, while levels above this line indicate a level reduction compared to the normal fan. The results (*) revealed that the dummy fan was on average 2 to 12 dB quieter than the normal fan, (above 200 Hz). The increased noise was next shown to be associated with air flowing through the fan. The inlet to the normal fan was blocked with a single layer of thin plastic film (Glad food wrap), and the fan was driven by the electric motor at 7500 rpm. This reduced the noise above 500 Hz, by at least 9 dB (20 fig. 3.17), showing that the motor drive and bearings are generally quieter than the dummy fan, (below 2000 Hz).

3.4. Vibration-isolated, high-temperature cylinder pressure transducer

Any measurement of combustion noise, including reciprocity, requires measurement of the cylinder pressure spectrum. Early in the stage of measurements, it became apparent that there were no pressure transducers available that could provide valid measurements while exposed to the high heat, pressure, and vibration present within the cylinder.

As recommended in the literature⁵, a Kistler 6121A2 quartz element transducer (with integral heat shield) which was exposed directly to the combustion chamber (see appendix G.3.2) was originally used for cylinder pressure measurements. The transducer worked well for low frequency measurements, but above 1 kHz, the acceleration sensitivity was too high to allow measurements of pressure in the no load condition (see fig. 2.1). Inspection of the available literature indicated similar difficulties in more recent measurements on another petrol engine⁶⁸ (note that diesel engines show less effect due to increased cylinder pressure at higher frequencies).

It was thus decided to vibration isolate the pressure transducer. Experience with Viton o-rings had shown them to be extremely durable and reliable under extremes of temperature and pressure. Room temperature testing on a shaker table indicated they would also be effective for vibration isolation. The o-rings produced attenuation above resonance of over 12 dB per octave, extending upwards to 8 kHz. They also had the unusual property of fairly high damping at resonance (with vibration amplitude increasing only about 3 dB). It was decided to use these o-rings both to seal, and vibration isolate the pressure

transducer within the cylinder (see fig. 3.18). The vibration isolated mass was selected based on the shaker table testing. However, the effect of heat on the o-ring properties was not known, and thus provision was made to increase or reduce the mass.

The pressure transducer was mounted in a 0.3 kg mass, and compliantly suspended by Viton o-rings on three axes (fig. 3.18). The outer diameter of the lower o-ring was minimized to reduce transducer movement due to cylinder pressure (appendix B). Severe space restrictions made it difficult to secure the assembly to the cylinder head, and to fit the transducer in between the handle and cylinder.

To reduce thermal effects on the transducer and lower o-ring, which were directly exposed to combustion, clearances were exactly sized to match the maximum calculated transducer movement. This reduced the quantity of combustion reactants directly in contact with the transducer. A brass heat shield was also placed at the lower end of the o-ring to protect it from the combustion flame. Although these measures could not stop charring and eventual degradation of the o-rings, they allowed hours of operation before replacement of the o-rings became necessary. An external fan was provided for additional cooling of the transducer and o-rings.

The behavior under operating conditions on the chain saw was similar to the room temperature measurements, except that the amplitude at resonance was slightly increased. This caused the transducer holder to impact against its end stops. This behavior increased high frequency vibration requiring the end stops to be moved back, until impacts no longer occurred. Evidence of these impacts was obtained by painting the end stops, and visually inspecting the paint for chipping, or by monitoring the transducer acceleration with a fast Fourier transform analyzer. Examination of higher derivatives of the input signals, showed small spikes occurring at the limits of transducer displacement, which is consistent with the occurrence of grazing impacts with the end stops. When these occurred, a corresponding increase in high frequencies was noted, producing a spectrum relatively flat up to 3 kHz. These high frequencies were reduced by enlarging clearances.

In the final design, small impacts occasionally occurred, which resulted in momentary increases in the high frequency output from the pressure transducer. Experience suggested the occurrence of these impacts could have been reduced by enlarging the clearances. However, this would have caused additional problems with the measurement in terms of increased cylinder volume, transducer movement, and exposure to the flame front from combustion.

Reduction of acceleration normal to the diaphragm is seen in fig. 3.19, (taken under actual full load operating conditions on the pulpwood saw). Vibration reduction to the transducer begins at 500 Hz and drops an average of 14 dB per octave to a maximum reduction of 56 dB at 8000 Hz. The vibration isolation effectively reduces the high frequency acceleration sensitivity of the transducer by up to 56 dB.

To find the magnitude of measurement errors due to pressure changes (i.e., cylinder volume changes), induced by transducer movement, a comparison between pressure and acceleration sensitivity is appropriate. The acceleration and pressure sensitivities are determined by the same first order system, and are related by a constant. It is shown in appendix B that the induced sound pressure level in dB (re 2×10^{-5} Pa) will always be at least 6 dB below transducer acceleration in dB (re 10^{-6} ms⁻²). Thus the induced pressure can be ignored, as the transducer acceleration sensitivity (appendix B) will produce a false pressure reading that exceeds the induced pressure by 11 dB.

The charge amplifier matched to the pressure transducer (Kistler type 504E) had insufficient dynamic range and sensitivity to allow either measurement at high frequencies, or calibration with a pistonphone (Brüel & Kjær type 4220). A Brüel & Kjær type 2635 charge amplifier was found to operate equally well, and yet provided additional sensitivity and dynamic range. In operation, the charge amplifier output was split into two paths, low frequencies that were recorded directly, and higher frequencies, which were high pass filtered at 1.8 kHz to avoid overloading the measurement and recording instruments.

The functionality of the vibration isolation was checked by a comparison of similar configurations by using different equipment. For example, changing the high pass filtering frequency that was used showed little difference. Comparable results were obtained from the 1/3 octave, and FFT analyzer. At low frequencies where the required dynamic range was not as large, use of the unisolated transducer, isolated transducer, Brüel & Kjær charge amplifier, or Kistler charge amplifier, showed no discernible difference at frequencies unaffected by vibration, (below 800 Hz). It is believed that the vibration isolation of the pressure transducer has allowed the first valid measurements of cylinder pressure on a petrol engine at frequencies above 1 kHz.

The crankcase temperature approximated room temperature. This allowed use of a conventional pressure transducer (Kistler type 601B), with acceleration sensitivity almost 30 dB better than the cylinder pressure transducer. Useful measurements were possible

2

PM-1 3 1/2" x 4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

without vibration isolation. A solid steel dummy transducer was used to replace the pressure transducer for tests that did not require the measurement of crankcase pressure.

3.5. Computer based data acquisition system

The computer data acquisition system could control all aspects of data acquisition and storage (fig. 3.5). It consisted of a Hewlett Packard model 9816 computer interfaced through a Hewlett Packard model 3421A data acquisition system. The system controlled operation of the Brüel & Kjær type 3360 sound intensity, (sound pressure) measurement analyzer, Racal Store 4 FM tape recorder, and monitored the dynamometer load, and speed. During field testing the computer monitored speed and could reject data based on 1/3 octave sound pressure spectra. Two way communication between operator and computer was established with a remote hand unit. During measurements all input signals were normally monitored by oscilloscope.

In conjunction with other measurements, the computer data acquisition system also recorded narrow band spectra using an FFT dual channel signal analyzer (Brüel & Kjær type 2032) with 801 lines resolution. Measurement was obtained simultaneously with other measurements, but real time power measurements could not be obtained over 800 Hz because 75% overlap of the Hanning window could not be obtained at high frequency. The problem could be eliminated by slow playback of the tape recorded data.

Spectra from the data acquisition system, 1/3 octave and FFT analyzer obtained during measurements was recorded to floppy disk. Output from the FFT analyzer included a time history with 2048 points resolution. These data were input into an IBM compatible computer for further analysis, such as plotting, statistical analysis, or Wigner distributions (appendix E).

Chapter IV Experimental Procedures

Measurements were made both in the field and in the laboratory. The saw was further analyzed in the laboratory to obtain a more detailed analysis of the sources involved. For reference, laboratory sound pressure measurements were made under conditions similar to actual use in the field.

4.1. Field measurements

Sound pressure levels were recorded at the outdoor field test site, with the saw operated at full throttle and best cutting speed, (7500 rpm for the pulpwood saw, 9600 rpm for the large-timber saw). During most experiments, 2 cm thick discs were cut from the end of a 35 cm diameter hardwood log, (fig. 4.1). The log was supported horizontally, 0.8 m above the ground and was 45 to 120 cm in length. In one experiment designed to isolate the noise associated with the cutting components, the saw was operated at best cutting speed but with no load. Data acquisition was controlled by the computer, and measurements began after the saw had cut to a depth of 5 cm. The data were discarded by the computer if the average saw speed was not within best cutting speed ± 300 rpm. During cutting, the engine speed was monitored by the sawyer using a matchbook sized tachometer with a digital readout.

A condenser microphone (Brüel & Kjær type 4134) with windscreen, was mounted at the operator's ear, 8 cm from the side of the head, (fig. 4.2). A similar microphone was located in the far field, 5 m from the saw, 1.0 m above the ground, in a direction defined by the cylindrical axis of the log (fig. 3.1.4.1). The operator's ear microphone signal was linearly averaged for 4 seconds using a one-third octave band real-time analyzer (Brüel & Kjær type 3360, see also fig. 3.5). Both microphone signals were simultaneously recorded for subsequent analysis by an FM tape recorder (Racal, Store 4), with flat frequency response from DC to 10 kHz (± 1 dB). The repeatability of sound pressure measurements was ± 2 dB from 800 to 10 000 Hz, with field calibration at 1000 Hz both before and after each experiment (using Brüel & Kjær type 4230 sound level calibrator), careful tuning of the engine and maintenance of the cutting components. Comparable repeatability could be obtained at lower frequencies associated with the engine firing and its harmonics by computer control of data acquisition keyed to engine speed. This technique was employed to obtain the data presented in Figs. 5.1 and 5.2. The measurement procedure is similar to that specified by the International Organization for Standardization for determining the

noise experienced by operators of chain saws,¹⁰⁹ with the main exception that the surface of the test site was acoustically reflecting, rather than sound absorbing.

The most intense sources were rank ordered by first identifying and then removing the dominant source. This was achieved by piping away and silencing the engine exhaust and intake. Separation of the noise associated with the cutting process from that radiated by the surfaces of the machine was achieved by placing the saw in an acoustic enclosure.

4.2. Laboratory measurements

Detailed diagnostic measurements were performed indoors in the laboratory. The speed control and relatively easy access to the saw, which was provided by the test stand, permitted the use of a wide variety of tests.

Laboratory measurements were performed with the pulpwood saw suspended from the test stand. The engine speed was maintained at 7500 rpm, (equivalent to best cutting speed). For full load testing, the engine power was absorbed by an electric dynamometer that controlled speed to within ± 100 rpm. During reference measurements, the saw was operated with no load, the dynamometer disconnected, and the speed controlled by the throttle to within ± 150 rpm.

Exhaust and intake systems were piped away and silenced (as in field measurements of sound pressure). As already noted, the clutch cover, guide bar, and chain were removed, in order to eliminate additional noise.

4.2.1. Measurement of sound pressure

A condenser microphone (Brüel & Kjær type 4181) was mounted on the test stand at a position equivalent to that of the operator's ear. The operator's ear position is defined by a point 0.75 m vertically above the crankshaft, as specified in ISO 7182 for determining the noise experienced by operators of chain saws¹⁰⁹.

The microphone signal was linearly averaged for 8 seconds, with the same real time analyzer and tape recorder used for the field measurements of sound pressure. With calibration at 250 Hz both before and after each experiment (using Brüel & Kjær type 4220 pistonphone calibrator), and careful tuning of the engine, the long term repeatability of sound pressure measurements in individual 1/3 octave bands was typically ± 1 dB from 630 to 12 500 Hz. Improved repeatability of ± 0.6 dB was obtained for measurements

taken during a single day. The variability of overall A-weighted level was ± 0.6 dB for a single day, as well as for long term measurements.

4.2.2. Measurement of sound intensity

The component acoustic intensity normal to the surface of the saw was measured in the laboratory by a one-third octave band real-time intensity analyzer (Brüel & Kjær type 3360). A single channel sound pressure measurement was simultaneously tape recorded as in the field measurements. The average intensities of individual panels were determined in the frequency range 250 to 10 000 Hz at a distance of 5 cm (to the acoustic center of the two microphone probe), by scanning each surface of interest for 16 s. The scanning pattern was either vertical bands, horizontal bands, or a spiral: the pattern used was not found to affect the results. Each pass of the microphone was separated from the previous pass by 2 to 4 cm. Care was taken to maintain uniform coverage of the surface. This required a consistent separation between scanning passes, and appropriate speed of scanning at reversals of direction. Single point measurements were taken either at 5 cm from the saw surface or at the operator's ear position¹⁰⁹.

The microphones employed in these measurements were selected by the manufacturer for minimal differences in sensitivity and phase (Brüel & Kjær probe type 3519, with 1/2 inch microphones type 4181, and 4165 with adapter UA 0808, and 1/4 inch microphone type 4135). The phase difference was confirmed to be within manufacturer's specifications ($< 0.05^\circ$) using an intensity calibrator (Brüel & Kjær type 3541). In some experiments when the phase difference was not measured directly, the microphones were checked at 1000 Hz before and after each experiment using the sound level calibrator. The influence of the phase on acoustic intensity was inferred by comparing measurements employing different matched microphone sets (Brüel & Kjær types 4165 and 4135).

Under these conditions and with a microphone separation of 6 mm, the repeatability of measurements was within ± 2 dB from 400 to 10 000 Hz. Similar repeatability could be obtained using a microphone separation of 12 mm, but above 5 kHz a correction for the microphone spacing (calculated for a plane wave, for simplicity) was required to match the results found using the 6 mm spacer. This range is believed to reflect the precision of the measurements as the difference in level between sound pressure and intensity, an indicator of the errors associated with reverberant sound fields, was less than 8 dB at most frequencies^{107,108}. Data were rejected when the pressure-intensity index exceeded 12 dB.

Intensity measurements could be made above 200 Hz for saw to microphone separation of 50 cm and above 500 Hz for a separation of 50 to 130 cm.

4.2.3. Measurement of acceleration

Acceleration was measured normal to the measurement surface using piezoelectric accelerometers (Brüel & Kjær types 4393 and 4344) attached with steel screws, following the manufacturers specifications, and ISO 5348. The weight of the accelerometers was 3 g. To reduce triboelectric noise, the accelerometer cables were glued to the accelerometer and saw in order to minimize flexing and impacts against the saw surface¹¹⁰. The accelerometer signals were conditioned using a charge amplifier (Brüel & Kjær type 2635) and processed by the same real time analyzer and tape recorder used for field measurements. Calibration at 80 Hz was checked at regular intervals using a Brüel & Kjær type 4291 calibrator. The frequency response of each accelerometer was verified on a shaker table (Brüel & Kjær type 4808) using a reference accelerometer, (Brüel & Kjær type 4374), which was fastened base to base with the accelerometer under test. The repeatability of vibration measurements was ± 5 dB at all significant frequencies up to 10 000 Hz.

In most measurements, the acceleration of the main bearings, cylinder, and the side panel of the crankcase, was monitored. Bearing acceleration was monitored by mounting three orthogonal accelerometers on the oil pump, (or dummy oil pump). The oil pump provides an axial preload on the main bearings, and so was in close contact with them.

The piston and cylinder acceleration was monitored by placing an accelerometer on the outside of the cylinder wall. For most measurements, the accelerometer was attached to the end of the plug at a location even with the ends of the cylinder fins, and oriented perpendicular to the cylinder axis. The plug was located near the upper limit of piston travel, opposite the piston thrust side of the cylinder. The cylindrical aluminum plug was pressed into a hole reamed out between two of the cylinder fins to a depth almost touching the cylinder wall. As the exhaust port was near the accelerometer, the accelerometer was mounted on a mica washer, and a small stamped brass heat sink was placed between the washer and the accelerometer to reduce its temperature.

The remaining accelerometer locations were used to estimate the reaction of the saw panels. One accelerometer was located on the crankcase side panel at the position of maximum acoustic intensity. The other accelerometer was located on the same crankcase half, at the position of minimum recorded acoustic intensity.

4.3. Source-specific measurements

4.3.1. Measurement of cylinder pressure

Engine cylinder pressure was measured with the vibration-isolated pressure transducer mounted flush to the cylinder wall (fig. 3.18). The saw was operated on the test stand, with intake and exhaust silenced, and bar and chain removed. The engine was warmed up under full load, and engine speed was controlled within ± 100 rpm by use of the dynamometer. Measurements were then taken with full load. Next, the dynamometer connection was removed, and the saw temperature was allowed to stabilize. Then measurements were made of the no load condition, with the engine speed controlled within ± 150 rpm using the throttle. This procedure improved the repeatability of the no load measurements, by reducing fouling of the transducer and shortening the warm up time required.

Signals from the transducer were conditioned by using the same charge amplifier, analyzer, and tape recorder as in acceleration measurements. One channel of the FM tape recorder was directly connected to the charge amplifier, while another channel was connected through a Krohn Hite 3343R filter (1.8 kHz high pass setting) in order to increase the dynamic range that could be recorded. Charge amplifier sensitivity was set according to manufacturer's specifications and was checked at regular intervals by use of a piston phone (Brüel & Kjær type 4220), fitted with a custom made mounting for the transducer. Calibration results were within manufacturer specifications for linearity and hysteresis, and gave a sensitivity typically -4 dB re the pistonphone output (124 dB re 2×10^{-5} Pa). The repeatability of cylinder pressure measurements and calibration was within ± 3 dB (note in appendix G.3.2, manufacturer's specifications are expressed as a percentage of full scale, suggesting the repeatability in decibels should improve at higher pressures).

4.3.2. Measurement of crankcase pressure

Crankcase pressure was measured with a quartz pressure transducer, (Kistler type 601B), mounted flush to the crankcase wall. Measurements were performed using the procedure and equipment used for the cylinder pressure measurements. Calibration was set according to manufacturer's specifications, and was checked by the same method used for the cylinder pressure transducer. The repeatability of crankcase pressure measurements and calibration results was within ± 3 dB.

4.3.3. Measurement of structural attenuation

The structural attenuation, as defined here, was taken to be the level difference (in dB) between the sound pressure at the operator's ear, and the gas pressure inside the saw. Structural attenuation was used to estimate the noise contribution at the operator's ear from the gas pressure fluctuations within the crankcase and cylinder. Because of the high levels of structural attenuation from the cylinder, different techniques were used to estimate structural attenuation from the crankcase, and for the cylinder.

4.3.3.1. Crankcase and intake system

The structural attenuation of the crankcase is limited by airborne transmission of noise through the intake. This was the easiest path for noise to escape from the crankcase. The transmission of noise from the crankcase through the intake is modified by the intermittent coupling between crankcase and intake resulting from the operation of the reed valve. Structural attenuation due to the walls can be obtained on a motionless saw, but, to account for the effect of the reed valve, the saw must be running. Thus, two methods were used to calculate the noise from the intake.

In both cases, the crankcase excitation was provided by an enclosed Adamson horn driver connected through the cylinder exhaust port (fig. 4.3). In order to allow the sound to propagate from the exhaust port to the crankcase, the piston was positioned at bottom dead center, to open fully the exhaust and transfer ports. The signal source for the horn driver was white noise produced by a Brüel & Kjær type 2032 analyzer, fed through a Bryston 2B, 50 watt amplifier.

In one method, the level difference between crankcase pressure and received pressure at the operator's ear was found by direct measurement on a stationary saw. Acoustical excitation of the crankcase was provided through the exhaust port, and the sound pressure level transmission was measured at the operator's ear position. Crankcase and cylinder sound pressure levels were monitored by using Kistler 6121A2 and 601B pressure transducers. All three pressure signals were recorded with the FM tape recorder and analyzer used for sound pressure measurements. The structural attenuation of the crankcase and intake system was obtained by subtracting the operator's ear pressure from that of the crankcase. This estimate accounted for leaks from the crankcase, but did not account well for the noise escaping through the intake. Accuracy was expected to be lowest near the frequency corresponding to the engine firing frequency, because the reed

valve closes when the crankcase pressure exceeds the intake pressure, thus isolating the crankcase from the intake.

The second method accounted for the reed valve opening and closing, and for actual operating conditions. Open intake noise (with the carburetor inlet exposed to the atmosphere) was obtained on the test stand at the operator's ear position from a saw running at full load. The unducted intake system insertion loss relative to the open intake was next obtained on a motionless saw. The effect at the operator's ear due to noise escaping from the intake, was found by subtracting the intake insertion loss from the open intake noise. This method is least accurate above 1000 Hz, where the open intake noise is masked by other sources associated with the running saw. It also does not account well for any noise that may escape through the walls of the crankcase. In addition, the calculated levels will overestimate actual levels, since the normal reference condition used a duct on the intake, which should help to reduce the noise.

Of the two methods discussed above, one is a poor predictor of the effect of the reed valve, and the other is a poor predictor of effects of transmission through the crankcase walls. A combination of the expected noise levels from both methods provided a conservative estimate of the noise escaping from the crankcase. The modifications made to the intake system should make the noise emitted insignificant.

4.3.3.2. Combustion noise from cylinder

The structural attenuation of the cylinder could not be determined by a direct measurement. Noise inside the cylinder was attenuated by more than 90 dB at the operator's ear position, reducing the levels below those of background noise. Thus to increase the signal to noise level, the reciprocal measurement was used. A point source was located at the operator's ear position, and the receiving microphone was placed in the cylinder, (fig. 4.4). Measurement of the volume velocity of the point source, assuming spherical spreading, and the pressure in the cylinder, allowed for the reciprocal calculation of pressure at the operator's ear due to a source in the cylinder.

Measurements were taken with the saw located in the laboratory and not running. The piston was moved by turning the crankshaft, thus, no structural modifications were required to perform the tests (with the exception of a small amount of Plasticine used to block the exhaust port). Even with the improved dynamic range of the reciprocity measurement, the signal to noise level still appeared to be negative at some frequencies below 1250 Hz.

The point source consisted of the Adamson M200 horn driver, located 0.75 m vertically above the crankshaft in the reference position for the operator's ear. Point source volume velocity (appendix A.1) was calculated from the pressure recorded using a Brüel & Kjær type 4181 microphone, mounted at 45° to the speaker axis, at a distance of 25 cm. Cylinder sound pressure levels were monitored using Panasonic WM063T electret condenser microphone and a custom made preamplifier. The microphone signals were recorded with the same FM tape recorder and analyzer used for sound pressure measurements.

Chapter V Results and Discussion

5.1. Field measurements

5.1.1. Exhaust, intake and residual noise

Sound pressure levels at the operator's ear and in the far field when sawing wood at best cutting speed are shown in Fig. 5.1. These one-third octave band spectra were recorded with open exhaust and intake (X) (i.e., exhaust port and carburetor inlet exposed to the atmosphere), with open intake and the exhaust piped away (+), and with both exhaust and intake piped away (◆). The overall A-weighted sound levels corresponding to these conditions are listed in table 1. As expected, the open exhaust is significantly more intense than the other sources, though the relative source strengths recorded at the operator's ear and in the far field appear to differ somewhat. The reason for this is unclear. The variation may be associated with the proximity of the near-field microphone to the operator and saw, and with the low-frequency directionality of internal combustion engine exhaust noise.¹¹¹ It also should be noted that the exhaust discharge of the large-timber saw is directed away from the far-field microphone.

The extent to which the aeroacoustic sources are reduced by production exhaust and intake mufflers, is also shown for the same operating conditions in fig. 5.1 and table 1. Figure 5.1 (*) shows the noise at the operator's ear and the far field, respectively, of the saws as supplied by the manufacturer, and after the saw has been "run in," (from now on called the "stock" condition). Evidently in this condition, noise is dominated by the intake and exhaust at frequencies below 1250 Hz, and by the residual mechanical sources (remaining after the exhaust and intake are piped away) at higher frequencies. Table 1 reveals that, the potential for further reduction in the noise of these saws by treating the exhaust and intake is limited (3-4 dB), before the overall A-weighted sound level is determined solely by the residual sources.

Due to the use of measured values, and typical variations in levels, table 1 is rounded to the nearest decibel. For this reason the values may not sum exactly (i.e., the value of Stock Exhaust summed with the Stock Intake overestimates the result given for the Stock Saw). Results such as these can be confusing and potentially misleading, so in general, results will be shown as variations in spectra, rather than overall A-weighted sound levels.

Table 1: A-weighted sound levels for various source conditions at the operator's ear and in the far field when sawing wood at best cutting speed. Symbols in brackets match those used in figures.

Source Conditions	Operator's Ear (dB)		5 meter Far Field (dB)	
	Pulpwood Saw (7500 rpm)	Large-timber Saw (9600 rpm)	Pulpwood Saw (7500 rpm)	Large-timber Saw (9600 rpm)
Open Exhaust (X)	116	129	110	110
Open Intake, Silenced Exhaust (+)	108	110	91	95
Mechanical noise (Silenced Exhaust and Intake) (◆)	102	107	88	93
Stock Exhaust, silenced intake	106	114(109*)	93	95
Stock Intake, silenced Exhaust	104	109	88	94
Stock Saw (*)	106	114(109*)	92	96
Chain and cutting (Powerhead in acoustic enclosure) (■)	91	97	83	86

*using improved stock muffler supplied by manufacturer

5.1.2. Residual noise

To establish whether the residual mechanical noise is radiated primarily by the cutting components and wood, or by the machine surfaces and engine cooling fan, the saws were mounted in acoustic enclosures. The A-weighted results which are shown in Table 1, reveal a significant reduction in level from the residual noise (10-11 dB). In view of the similarity between source characteristics of the two saws, particularly as evidenced by the far-field data, results of this and other experiments are presented here only for the pulpwood saw.

Figure 5.2 shows the sound pressure levels observed when cutting wood with the pulpwood saw mounted in the enclosure with intake and exhaust piped away. The residual mechanical noise recorded without the enclosure is also shown (◆) for comparison. The maximum noise reduction provided by the enclosure is shown by ■, (obtained with the chain running, but not cutting). Evidently, the frequency range in which the largest contributions occur to the A-weighted sound level of the residual sources (800-8 000 Hz) corresponds closely to the frequencies at which this noise is most reduced when the saw engine is enclosed (■). Hence, the residual noise must be radiated primarily by surfaces or sources within the enclosure, which together comprise the powerhead.

5.1.3. Noise radiated by the cutting components

With the powerhead enclosed and the engine operating at best cutting speed (but not cutting wood), a 20 dB reduction in residual mechanical noise was observed in the far field at frequencies of 4 kHz and above, when the cutting chain and guide bar were removed (— Fig. 5.2). Reattaching the cutting components to the powerhead, while continuing to operate the saw under the same conditions (■), (i.e., best cutting speed but no load), resulted in a large increase in sound pressure at high frequencies.

A continuation of the experiment with the powerhead enclosed resulted in curves ■, and ■, when the saw was used to cut two types of wood, (old damp slightly rotten wood, or dry wood respectively). When compared to the chain noise (■), cutting significantly increases the noise radiated at mid frequencies. These differences are most notable at the operator's ear, where the spectra are clearly dependent on wood type. Note, that the level changes due to cutting remain at least 6-8 dB below the residual noise of the saw in stock condition (◆) at frequencies above 1250 Hz, and so may be inaudible. The dependence of sound pressure on wood type with the powerhead enclosed, suggests an interaction between the bar chain and wood.

In the far-field data (Fig. 5.2a), evidently cutting produces significant noise only at mid frequencies (1-4 kHz). At higher frequencies the similarity between curves ■, ■ and ■ implies that the bar-chain interaction dominates the sources outside the acoustic enclosure. Also, the zero level difference in the far field at high frequencies, suggests that this source determines the residual mechanical noise of the saw in stock condition, at frequencies above 8 kHz.

The inconsistent differences in sound pressure between those recorded in the far field and those at the operator's ear (Figs. 5.2a and b), suggest the bar and chain noise is directional, and preferentially radiates normal to the cutting plane (or surface of the bar), toward the far field microphone, rather than from within the cutting plane (from the edge of the bar), toward the operator's ear. The mechanical noise in the stock condition (◆), does not show significant directivity when comparing results at the operator's ear and far field (changes in level are consistent with the difference in path length between source and receiver).

The apparent reduction of the high frequency chain noise when the saw is enclosed, suggests the chain noise reaching the operator's ear is dominated by sources within the acoustical enclosure. This is consistent with the expectation that the chain noise is

strongly influenced by sprocket impacts due to polygonal action (section 2.8). As the sprocket is within the enclosure and attached to the crankshaft, the vibration excitation must travel through the powerhead before being transmitted to the guide bar.

Results when using an acoustical enclosure showed a 6 dB difference in the noise levels of the large timber, and pulpwood saws (table 1). Separate measurements under similar conditions but reduced speed (6000 rpm for the pulpwood saw, and 7500 rpm for the large timber saw) decreased noise levels by about 2 dB at most frequencies, (comparable to normal test to test variations). Thus the level difference between the two saws does not appear to be due to speed, and may relate to the cutting rate, which was approximately double in the large timber saw (compared with the pulpwood saw).

Of the 4 types of hard wood tested, (old, fresh, dry and frozen), only frozen wood had a measurable effect on the mechanical noise (2 to 3 dB increase at most frequencies above 800 Hz). The cutting tests have shown that although wood type can influence radiated noise, the changes in level are generally masked by other sources. The chain, however, was a significant noise source, and further investigation of this source was left to measurements with the saw in the laboratory.

5.2. Laboratory measurements

Laboratory tests differed from field tests, in that the saw was supported by the test stand, and unless otherwise noted, was operated without load, or chain. In addition, some modifications were made to the saw to reduce noise (section 3.3).

Repeated measurement in the laboratory of the sound pressure levels at the operator's ear position, permitted an estimate of the variation in spectra during reference chain saw operation. The average levels for all tests are shown in figure 5.3 by diamonds (◆). Values for plus and minus one or two standard deviations of decibel levels are shown by the shaded areas in figure 5.3. These data were derived from 18 data samples obtained in 8 days of testing, representing normal measurements spanning a period of 3 months.

The average level and standard deviation shown in figure 5.3 were obtained by both geometric and arithmetic means of the sound pressure. The original data in each of the 18 measurements (— figure 5.3), were obtained by an energy (mean square pressure) average over the time of the measurement. A linear average of the dB levels reduces the average power of multiple tests compared to a long duration single test. To avoid changing the total power, the measurement average (◆) was also based on energy.

However, the variation of levels about this average appears normally distributed in terms of decibels. As seen in figure 5.3, all measurements (—) are bounded by ± 2 standard deviations of an arithmetic dB average (shaded area in fig. 5.3). Note that this figure also shows ± 1 standard deviation, but as could be expected, this area is obscured by the measurements (see for example fig. 5.4). Thus the average levels ($\diamond$) in figure 5.3 are an arithmetic average of pressure squared, and the standard deviation was obtained by a geometric mean of the same values.

Variations in 1/3 octave band levels obtained in a single day's testing were approximately half those shown in figure 5.3. The cause of the increased variation over the longer term is unclear. Possible factors are, repeated disassembly of the saw, small changes in experimental setup, or location of equipment in the room. The range below 630 Hz is composed of well-defined harmonics of the firing frequency, and may reflect changes in the speed of the saw, which was controlled by the tuning of the carburetor.

The standard deviation of the overall A-weighted noise was 0.6 dB for all 18 measurements. The deviation in overall A-weighted noise could not be further reduced, even when restricted to measurements taken in a single day.

The most effective method of source identification has been to modify individual source strengths. The effect of changing one of the source strengths on noise can best be established by comparison with the shaded area showing plus and minus two standard deviations in figure 5.3. Any measurement that falls outside of the shaded area would suggest a significant deviation from the reference condition (as defined in section 3.3, or appendix G.1.2.2).

5.2.1. Comparison of laboratory versus field measurements

For ease of comparison, figure 5.4 summarizes the differences between laboratory and field measurements. These differences are discussed in detail in sections 3.2, and 3.3. With the saw similarly configured, the laboratory reference condition (shaded area in figure 5.4) shows less noise at high frequencies than field measurements ($\diamond$). This is due to the absence of the cutting chain. At other frequencies, statistically significant differences exist due to the change in measurement conditions, however, these differences are small and may be ignored when comparing sources.

Figure 5.4 also shows that in field measurements when cutting dry wood, sound pressure levels at the operator's ear with the modified saw, ($\diamond$), were about 6 dB lower in level

than the original field measurements ($\blacklozenge$). Much of this difference can be attributed to modifications to the intake system. This reduction in level allowed the effect of cutting to be seen, as demonstrated by the level reduction with the saw hand held with the chain running, but not cutting (∇).

5.2.2. Noise from the powerhead

To investigate further the sound radiated by the powerhead, the average acoustic intensity of each surface was determined with the saw mounted on the test stand. It was found from an examination of intensity spectra that they may be classified into two types, depending on whether the radiating surface is directly coupled to, or vibration-isolated from the crankcase. Examples are shown in figure 5.5. The fan (⚙) and clutch covers (⚙), which together form over 25% of the side surfaces of the saw, are examples of panels rigidly attached (by screws) to the crankcase. In contrast, the base plate (⌞) and rear structure containing the right hand guard (⌞) are die-cast parts of the handles, which in turn, are compliantly attached to the engine. These spectra show that the vibration isolators provide approximately 4-6 dB reduction in acoustic intensity at 800 Hz and above; frequencies at which residual sources dominate the noise of these saws. Although the handle isolators are currently designed to reduce the transmission of low-frequency vibration to the hands, in order to lessen the risk of vibration induced white finger,¹¹² they could, with appropriate powerhead design, also serve the purposes of noise control.

5.2.2.1. Impact excitation of the powerhead

To determine the effect of impacts within the powerhead, the pulpwood saw was externally impacted with a hammer. This provided insight into problems with digital signal processing, and allowed a check of the structural response of the engine.

Association of a source event with the radiated noise is complicated, since panel bending modes radiate energy most efficiently. These modes are affected by dispersion and reverberation within the structure. This was apparent when observing panel acceleration with an oscilloscope. In one case, after a hammer impact, a strong mode was seen to occur at a frequency of about 2 kHz. The amplitude of this mode did not reach a peak until 2.5 milliseconds after the impact, and the vibration at this frequency was visible on an oscilloscope for over 10 milliseconds. At best cutting speed, the engine cycle lasts 8 milliseconds, thus the noise radiated from this panel at 2 kHz would occur relatively long after the impact, and would be radiated in conjunction with the noise of previous impacts.

Dispersion and reverberation reduce the coherence between impact and detected event. The reduction in coherence increases the farther the vibration sensor is located from the impact point. In tests with hammer impacts on the front of the saw, coherence approached 1 only when the sensor was at the excitation point. The average coherence decreased with increasing distance from the impact location, with levels below 0.9 at a distance of about 4 cm.

Hammer impacts also allowed a check of the estimated structural response of the pulpwood saw. Results from impacts at 4 separate locations are plotted using thin solid lines in figure 5.6. The vibratory power spectrum was obtained from equation 23, and was substituted into equation 21 to obtain the noise spectrum due to the hammer impacts. The result is plotted in figure 5.6 as a thick solid line. The calculated levels in figure 5.6 assumed roughly sinusoidal impact force time histories, and force spectra as described in section 2.5.3. The vibratory power and force derivative (F') corner frequencies of 2500 and 3150 Hz were arbitrarily chosen.

In general, the agreement in shape of the measured and calculated impact spectrum noise in figure 5.6 is very good. As could be expected, the magnitude of the response spectrum is sensitive to the location of the impact on the saw. Of note, the slope at high and low frequencies match in the measured and calculated results. This was not expected at frequencies below 1250 Hz, as the calculated slope was expected to overestimate the measured levels. The variability of the measured curves was too large to establish the break points in the calculated spectrum.

5.2.3. Sources internal to the powerhead

5.2.3.1. Fan noise

The chain saw mechanical noise sounds like the droning hum of the cooling fan. Examination of the blower cover spectrum ($\otimes$) in Fig. 5.5, suggests that there may be an aerodynamic contribution to the noise at the blade passage frequency of the engine cooling fan (3150 Hz). These observations suggest that the fan contributes significantly to the noise of the chain saw.

A dummy bladeless fan was used to reduce the effect of the fan on the mechanical noise. The resulting spectrum is shown in figure 5.7, ($*$). For comparison, average mechanical noise levels, (taken a few days before, and a few days after), are also shown ($\diamond$). The change in mechanical noise only equals the reduction expected ($*$ fig. 3.17) from the

bladeless fan in the 1/3 octave bands at 630, 800 and 1000 Hz. The local spectrum peak at the blade passage frequency of 3150 Hz was also absent when the dummy fan was used. These observations suggest that the fan strongly influences the residual mechanical noise at these frequencies.

By replacing the crankshaft with a straight shaft, and turning the fan with an electric motor, the other sources of mechanical noise were eliminated so that the fan's contribution could be determined. The results plotted in fig. 5.8 (✕) show that the main contribution to the fan noise occurs between 500 and 3150 Hz. These results indicate the noise at 500 and 1000 Hz can account for the measured reference mechanical noise. There is also a strong contribution at the blade passage frequency of 3150 Hz. The motored fan produced levels of 91.7 dB (A-weighted), approximately one third of the total mechanical noise of the chain saw (96.6 dB A-weighted).

Although externally motoring the fan indicates that fan noise can be significant, it may not provide accurate noise levels. The frequencies from 500 to 1000 Hz are most likely related to fan noise. These frequencies also show the largest standard deviation in the reference condition (shaded area in fig. 5.8). This indicates some uncontrolled factor in the measurements. One example of a quantity that changes significantly when the saw is motored versus running, is the saw temperature (which can change by over 100° C). An interesting result was obtained when the saw was operated at full load versus no load (fig. 5.11), conditions under which the saw temperature also changes. Around 1000 Hz and 3150 Hz, the full load mechanical noise levels (◆, fig. 5.11) are seen to peak at the same frequencies found with motored fan noise (✕, fig. 5.8). These results illustrate a potential problem with externally motoring the saw. As the conditions are not the same when the saw is motored versus running, the noise produced cannot be arbitrarily assumed to be the same.

The best estimate for the noise of the fan is obtained from the motored fan results (✕, fig. 5.8) although this may underestimate lower frequencies. The fan noise for the pulpwood saw (✕ in fig. 5.23) was assumed equal to the motored fan noise. The fan noise levels are 92 dB (A-weighted). Previous results using the dummy fan (*, fig. 5.7) suggested a stronger contribution from fan noise at some frequencies below 1250 Hz than were found with the motored fan. To agree with the dummy fan results, the motored fan noise (✕ in fig. 5.8) should be increased by about 6 dB at 630 and 800 Hz.

Empirical relationships (equations 15 and 17) predict a 15 dB or 18 dB sound pressure increase in fan noise per doubling of rpm⁴³. The motored fan noise changed at a similar rate at speeds between 5600 and 7500 rpm, (at frequencies above 315 Hz). However, the range of speeds tested was not large enough to determine whether 15 dB or 18 dB per doubling of speed was more appropriate.

On a chain saw running at different speeds, (fig. 5.9), lines drawn at 15 dB per doubling of rpm give reasonable agreement with the spectra between 600 Hz and the blade passage frequency. At higher frequencies, there is a change in the slope of the spectra, and an interval of 23 dB per doubling of engine speed is more appropriate. This suggests that the fan affects the noise levels below the blade passage frequency, and another source is dominant at higher frequencies. At the lowest speeds (3800 and 4800 rpm), all frequencies are approaching the 15 dB per doubling of rpm expected from the fan.

5.2.3.2. Combustion noise

The contribution of the combustion process to the noise from the powerhead was explored by changing the combustion pressure spectrum. The cylinder pressure and radiated sound were measured both at full power and no load, conditions in which the gas forces are known to change substantially in gasoline engines^{40,113,114}. The gas forces associated with the combustion process (fig. 5.10) were seen to increase by 4 to 10 dB when changing from no load (⊞) to full load (⊗). Yet, a comparison of the radiated sound spectra recorded at the operator's ear (fig. 5.11), only showed a significant increase at 1000 Hz and 3150 Hz for a change from no load (◇) to full load (◆). These differences are due to peaks in the spectrum 5 dB greater than the adjacent 1/3 octave bands, yet, the corresponding combustion pressure spectrum shows no evidence of peaks. This suggests a resonance or a change in a tonal source, (such as the fan). Ignoring the peaks at 1000 Hz and 3150 Hz, the lack of effect of load above 500 Hz indicates that combustion noise does not exceed the noise of other sources (in reference condition).

A measure of structural attenuation, (at the operator's ear), was obtained from reciprocity measurements with a source outside the chain saw and a microphone located in the cylinder. The results are shown in fig. 5.12. Although reciprocity significantly improved the signal to noise levels, background noise still caused an underestimate of structural attenuation. The high levels of structural attenuation show that the transmission loss is due to structural stiffness, (i.e., the mass law³⁰ indicates the wall mass must increase by 3 orders of magnitude to obtain the structural attenuation observed at 1 kHz).

An estimate of the combustion noise level at the operator's ear (✱ fig. 5.13), was obtained by subtracting the estimated structural attenuation (fig. 5.12) from the full load cylinder pressure (✱ fig. 5.10). Calculated noise levels at the operator's ear position in fig. 5.13, (✱), show that the full load combustion noise is significantly lower than the reference condition (shaded) at all frequencies except 400 Hz. The results indicate that the changes observed due to change in load (fig. 5.11) at around 400 Hz could be related to combustion noise. However, they do not explain the peaks at 1000 Hz and 3150 Hz, which may be related to another source, one that is affected by the increased heat or load.

Combustion noise levels estimated in figure 5.13, (✱) may overestimate the full load levels of the saw. However, the levels are so low their effect on overall noise is minimal, with an A-weighted level of 68 dB. Consequently, they have been directly copied to figure 5.23 (✱).

5.2.3.3. Crankcase and intake system

The previous tests showed noise changes due to load (fig. 5.11), that could not be related to combustion noise. The crankcase pressure is also affected by the load conditions, with pressure changes of up to 8 dB (fig. 5.14). Thus the crankcase pressure could produce noise spectrum changes with a change in load.

The influence of crankcase noise could not be measured directly, the received noise had to be related to the structural attenuation, and the insertion loss of the intake system. A conservative estimate of the reference condition intake noise (♣ fig. 5.15) was obtained by subtracting the intake insertion loss, (✱ fig. 3.14), from the open intake noise of a running saw in the laboratory. This measurement is least accurate above 1000 Hz where the open intake noise is masked by other sources. It also does not account for any noise that may escape through the walls of the crankcase. As an alternate measurement, the reference condition intake noise (✱ fig. 5.15) was also estimated by subtracting the transmission loss (similar to structural attenuation) of the intake from the full load crankcase pressure (♣ fig. 5.14). This estimate will account for leaks from the crankcase, but, does not account well for the noise escaping through the intake. It is least accurate at the engine firing frequency, and its harmonics, because the reed valve closes as the crankcase pressure increases.

The two predicted intake noise curves (♣, ✱ fig. 5.15) show fair agreement above 500 Hz. At most frequencies the noise from the crankcase and intake system is 20 dB below that of the mechanical noise. Thus the noise from the crankcase and intake cannot

account for any change in noise due to load in figure 5.11. The exception is at 125 Hz, where leaks from the crankcase could be significant.

The lack of influence of crankcase and intake system noise is due to the laboratory modifications made to the intake system of the saw. The noise in the stock condition can be strongly influenced by crankcase noise at frequencies below 6.3 kHz. Figure 3.14 (♣), shows that in the stock condition at most frequencies below 6.3 kHz, the insertion loss is at least 20 dB less than in the laboratory reference condition.

The total estimated A-weighted noise level for the intake system and crankcase noise on the modified saw is 81 dB at full load, with a spectrum shown by ♣ in figure 5.24. This estimate was obtained from open intake noise and insertion loss (♣ fig. 5.15), at frequencies below 500 Hz. At higher frequencies, the noise was assumed equal to the maximum levels in figure 5.15.

5.2.3.4. Shaking forces from piston and crankshaft

The effect of the considerable shaking forces due to piston and crankshaft movement was considered. Typically, only 30 % to 50 % of the dynamic unbalance of the crank and piston are balanced on two stroke single cylinder engines. The main shaking force exists at the fundamental rotation frequency of the crankshaft, and a secondary force of comparable magnitude is present at the first harmonic. These frequencies are below the lowest bending mode of the chain saw, and so, can only exist as rigid body modes. Measurements of acceleration and stroboscopic illumination were made along the cylinder axis (oriented towards the operator in up and down direction). They showed acceleration levels of 165 dB (re $1 \mu\text{m/s}^2$), at the fundamental, and up to 150 dB at the first harmonic. Previous calculations (section 2.4) have shown that this acceleration can be based solely on the effect of piston and crankshaft movement on the powerhead. Using these acceleration levels and assuming dipole radiation from the powerhead, the maximum expected sound pressure levels at the operator's ear were: 90 dB for the fundamental and 95 dB at the first harmonic. A comparison with figure 5.3 shows that the shaking forces more than account for the noise at the firing frequency and first harmonic. As the pressure radiated from a dipole source will be extremely dependent on microphone position, this could account in part for the difficulties experienced in field measurements and laboratory measurement of acoustic intensity at low frequency.

5.2.3.5. Piston slap

Quantitative information on the noise of piston slap was determined by running the chain saw with normal and undersize pistons; conditions that are known to affect piston slap noise. The chain saw manufacturer originally provided cylindrical pistons in 3 sizes with 0.013 mm difference between each size; these were to be matched with a cylinder in order to obtain a running clearance of 0.025 mm. To extend the range of sizes that could be tested, the diameter of one piston was reduced by using a metal lathe to provide a 0.12 cm running clearance. As indicated by equation 43, this should shift the piston impact spectra vertically upwards 9 dB^{36,67}.

Piston slap is not as significant as other sources at the pulpwood saw's best cutting speed. The results using the undersize piston (□ fig. 5.16), only show the expected 9 dB change, at frequencies around 8 kHz when compared to the reference condition (◇). The effect of piston slap is masked by other sources below 4 kHz. It must therefore be concluded, that in the reference condition (◇), piston slap is only significant above 4 kHz. Previous results have shown that this region is dominated by noise due to the chain, or cutting.

The frequencies and noise levels predicted for piston slap are shown by the solid line (—) in figure 5.16 (see section 2.6). The calculated normal piston slap noise levels are within 2 dB of the undersize piston slap levels. It can be concluded that normal piston slap noise will be approximately 12 dB below the solid line (—) in figure 5.16. This allows for a 1 dB contribution to the undersize piston noise from other mechanical sources (□, fig. 5.16), and the prediction that normal piston slap noise should be about 9 dB lower than the undersize piston levels. These differences were expected due to the original derivation of the piston impact energy.

The calculated spectrum for the normal piston clearance with the 12 dB correction, is plotted in figure 5.24 (□). The resulting A-weighted level for piston slap noise is around 85 dB. While piston slap is not significant on the pulpwood saw, the levels are sufficiently high that they may be important on other chain saws.

The piston side force should produce less noise energy than piston slap (section 2.6.5). On the basis of the measured piston slap noise frequencies, the frequencies most affected by side force should be below 8 kHz. The increase in piston cylinder clearance used would increase the side force noise level by 7 dB, compared to the reference condition (eqn. 53). This is only slightly less than the effect on piston slap. If the noise produced by the side force were significant it would be apparent in figure 5.16 (□). All spectrum level

changes can be attributed to piston slap, thus it appears the effect of piston side force is less than that of piston slap.

The effect of speed on piston slap noise was next considered in order to verify these results. Figure 5.17 shows previous results at 7500 rpm (shaded curve labeled 7500) along with results at 4 other speeds (at 1/3 octave speed intervals). The uppermost shaded curve shows results at 9600 rpm, while the lowermost curve shows results at 3800 rpm. The thickness of each curve represents the change in sound level due to the change in piston size. The lower edge of each curve represents the normal piston noise level, while the upper edge of the curve represents the undersize piston noise.

The predicted changes in piston slap noise energy are good at higher speeds, but not at the lowest speeds. The upper edge of each band should be used for comparison of piston slap noise, as this measurement is least affected by other sources. On the basis of figure 2.6, which predicts piston impact energy with speed: The sound pressure levels at 9600 rpm should be about 2 dB higher than those at 7500 rpm, while the levels at 6000 rpm should be about 7 dB lower. The noise levels for each speed below 6000 rpm should fall off at about 2 dB per 1/3 octave reduction in speed. The measured change with speed in figure 5.17, matches the expected values above 4800 rpm. However, at 3800 and 4800 rpm, the estimates from figure 2.6 do not correspond to the measured values.

The cause of the discrepancy at lower speeds is not known. It was observed that the chain saw would not run as smoothly at low speeds, thereby increasing the variability of the results. The vibratory power estimates in figure 2.6 were based on a cylinder pressure time history, measured at 7500 rpm, and thus may not be accurate at other speeds.

As expected, the effect of piston clearance in figure 5.17 is most pronounced above 4 kHz at all speeds. At 9600 and 3800 rpm, the change in level due to piston clearance (shown by the thickness of the curve), is significantly smaller than that found at the other speeds. This suggests a contribution to the noise from another source. The levels at 3800 rpm may be associated with the fan, which could exceed all other sources at this speed. The reason for the smaller than expected increase in noise at 9600 rpm is unknown.

Note that slight changes (.006 mm) in piston clearance, produced results opposite to those expected. A reduction in piston and cylinder clearance caused a slight noise increase, while an increase in piston and cylinder clearance tended to reduce the noise (although the difference was not enough to be significant). Similar behavior has been noted in other engines with tight fitting pistons and has been called stick slip noise⁷⁸. The noise has been

attributed to the piston sticking and slipping in the cylinder bore. These sticking forces that affect piston movement will directly affect the bearing forces. A large jerk in the piston could cause impulsive movement at the bearings, which also could account for the increased noise.

The previous tests showed that piston slap noise only had a significant effect on the reference condition at high frequencies. This made piston slap impacts difficult to identify. Two experimental analysis procedures were found to aid in identification of impact events in the accelerometer signal.

The previous results have shown that a continuous band of frequencies extending to over 8 kHz (see fig. 5.17, or section 2.6.4) are excited by the piston slap events. High-pass filtering of the signal at very high frequency produced a narrow pass band at the highest measured frequencies, which is similar in concept to providing an n th order derivative of the input signal. This provides two advantages: first, the restricted frequency range reduces the effects of dispersion, and second, the response to an impact decays more quickly at high frequencies (assuming structural damping doesn't change with frequency). Figure 5.18 shows an example of the results; the upper trace is the normal output of the transducer, where it is evident that impacts are difficult to identify. The next lower trace shows the signal high-pass filtered with an 8 pole Butterworth filter (appendix G.5.1), effectively giving a band limited 8th derivative of the upper signal. The high frequency bursts in the lower trace are associated with small spikes in the input signal, which are most likely due to impacts.

The Wigner distribution (appendix E) also provided similar information in a different format. Figure 5.19 shows a Wigner distribution for an accelerometer signal recorded over two cycles of the chain saw. The vertical axis shows time with a resolution of about 1° of crank rotation, and the horizontal axis shows frequency with a resolution of about 500 Hz. The darkness of each point is proportional to the logarithm of the amplitude. The dynamic range shown in the diagram is 40 dB, below which, results are shown in white (including negative amplitude results). An impact event excites a continuous range of frequencies within a short time, and thus appears as a black horizontal band. Excitation at a single frequency would produce a vertical line, although the poor (500 Hz) frequency resolution tends to accentuate impacts more than single frequencies. A few lines that likely represent impacts are indicated by arrows. Note that the frequencies affected extend over 12.5 kHz. While many additional features are also apparent, it is not known whether they represent real events or simply artifacts of the measurement.

The analysis of impact acceleration time histories did not account for impacts at locations away from the measurement position. Both the Wigner distribution and high pass filtering showed what appeared to be impacts just after top dead center. This was expected from the results in figure 2.5, however, the piston slap that must occur near bottom dead center was not apparent. As the sensing accelerometer was located near the top of the cylinder, this suggests the identification of piston slap impact events was sensitive to the location of the accelerometer. The piston stroke is 36 mm, which is about 1/4 of the circumference of the cylinder. Thus a number of accelerometers located near each expected point of impact would be required to identify positively piston slap events. This has implications for the measurement of other impact sources, for which it is not as easy to position accelerometers close to the impact event.

5.2.3.6. Bearing noise

The estimated sound radiation from all sources to this point does not fully account for the measured noise radiation in the reference condition between 1200 Hz and 4 kHz. The process of elimination leaves only one remaining source of sound energy within the powerhead, that of bearing impacts.

A common indicator of bearing noise is a symmetric amplitude modulation of the time domain signal (also indicated by sidebands in the spectra). Such a modulation was noted in the sound pressure measured at the operator's ear position. High resolution zoom analysis identified sidebands 30 dB down at 18 ± 5 Hz to either side of the fundamental and its harmonics. The modulation of the time history was anti symmetric about the zero, (DC), axis, and did not appear to vary in frequency when the chain saw speed changed. The frequency did not correspond to any rotational speeds in the bearings. The 18 Hz frequency only seemed to correspond to the difference in speed of the dynamometer and crankshaft, and a vibration isolator/powerhead resonance. Thus the existence of bearing noise was not immediately obvious from chain saw measurements.

As with piston slap noise, bearing noise is strongly affected by the impacts across bearing clearances. Thus changes in the bearing clearances were made to identify the sources. The main bearings in a chain saw are ball bearings, which, due to a radial preload, can run at close to zero clearance. As could be expected, no significant increase in noise was noted when the main bearing clearance was doubled, (changing from standard to C4 tolerance). The only estimate of main bearing noise came from the motored fan results, which used the main bearings mounted on a straight shaft (fig. 3.17). These results show

that at 7500 rpm, the bearings must be at least 10 dB quieter than the fan at most frequencies above 400 Hz. These results suggest that the main bearing noise should be less than 72 dB at all frequencies, with an associated overall A-weighted level of 79.8 dB. However, the results may not be representative, as the shaft was externally motored, the bearings were not running under normal operating conditions, and other noise sources were present. While the main bearings would not likely produce significant noise, the accuracy of these results would not be very high.

The bearings on the connecting rod are needle roller bearings that require¹¹⁵ approximately 0.036 mm, and 0.053 mm clearance at the little and big end respectively. These clearances are comparable to the piston and cylinder clearance, but the forces involved are much greater. Thus the impact energy and radiated sound energy could be comparable to those of piston slap, and may have similar factors affecting the noise levels.

The effect of changing the bearing clearance on the little end of the connecting rod is shown in figure 5.20. Three clearances were used, the normal clearance for the reference condition (shown by $\diamond$), four times the reference clearance ($\circ$), and eight times the reference clearance ($\bigcirc$). The results show a clear and significant effect due to the bearings, at and above 1250 Hz.

The largest effect of bearing noise is noted in the range 1250 to 3150 Hz; this is precisely the range that cannot be accounted for by noise from the other sources considered so far. Comparison of the spectral shape produced by increased bearing clearance ($\circ$, $\bigcirc$), and that of the reference condition, shows striking similarity above 1250 Hz. A similar match in spectral shape has not been noted for any other source considered. The spectral shape with increased bearing clearances is similar to that expected for piston slap impact noise (fig. 5.16, ---), suggesting that the noise may be due to impacts.

Measurements with increased bearing clearance were not made at speeds above 7500 rpm for fear of damage to the saw. Other engines have shown up to 86 dB noise increase per decade increase in speed³. It is thus reasonable to conclude that bearing noise should dominate at high engine speed. On the basis of previous measurements of piston slap (fig. 5.17), at higher speeds, the reduced effect of piston slap on the saw noise suggests that the bearing noise dominates. The shape of the spectrum (fig. 5.17) at 9600 rpm is also similar to that of the bearings in figure 5.20 ($\bigcirc$). It is also noteworthy that at frequencies where fan noise is expected to dominate the spectrum (below 1250 Hz) there is only a slight increase in noise when the saw speed is changed from 7500 to 9600 rpm. The increase in

spectrum levels is on average 7 dB, when the speed is changed from 7500 to 9600 rpm, suggesting that the bearing noise increases at least 70 dB per decade increase in speed. This change is lower than that found in other studies of petrol engines³, and may only indicate the presence of other sources with lower noise versus speed dependencies. As is possible with a source that changes significantly with engine speed, an increased little end bearing clearance affected the noise levels by less than 2 dB when the chain saw was slowed to 6000 rpm (or 4800 rpm). Other sources that are less affected by speed, such as fan noise and piston slap, should become the dominant sources at lower speeds; these conclusions are consistent with previous results for other sources. Thus by the process of elimination, it appears the bearings strongly influence the noise in the reference condition between 1250 to 3150 Hz.

There is no simple method to predict the noise produced by the bearings. Thus an attempt was made to estimate the bearing noise in the reference condition by curve fitting the data (fig. 5.20) using two different models to account for the effect of the little end bearing clearances.

$$p_j^2 = p_{other}^2 + (a(f_j)d^b)^2 \quad 65$$

$$p_j^2 = p_{other}^2 + (a(f_j)b^d)^2 \quad 66$$

where $a(f_j)$ represents the calculated spectral shape of the noise due to the bearings alone, p_{other} represents the calculated noise of all other sources, and p represents the measured total noise (giving 22 known values). The term d is the known bearing clearance, and b is an unknown constant to be determined. The subscript j indicates that each of the three preceding quantities are a function of frequency (and j), with values calculated for 22 frequency bands. Bold face is used in equation 66 as a reminder that the equations represent the same variables, but due to the model used, the calculated values differ.

The above models are used only to illustrate the possible range of normal bearing noise. Equation 65 is based on the model used by Ungar and Ross for piston slap noise. Equation 66 is based on the model used by Earles and Wu for bearing noise. Accuracy of the equations is limited, since piston slap is much simpler than bearing impacts, and the clearances used may be outside the range of applicability for equation 66.

On the basis of the models in equations 65 and 66, and the similarity of the noise produced due to the two increased bearing clearances, the bearing noise estimate is straightforward.

The assumption was made that $a(f)$, p_{other} , and b are independent of bearing clearance. The large bearing clearance determines the shape of the bearing noise spectrum. The difference between the two enlarged bearing clearances essentially determines the value of b . The lack of an effect of clearance at frequencies below 1250 Hz indicates that bearing noise is not significant at these frequencies. As a result, the levels of the other sources (p_{other}) could be inferred from simple subtraction.

Curve fitting the 3 bearing clearances at 22 frequencies produces an overdetermined system with $22+22+1=45$ unknown values and $3 \times 22 + 3 \times 1 = 69$ known values. The solution was obtained by minimizing the least square error in dB using a commercial numerical routine¹⁶. Values for b , and b were respectively 2.7, and 4.26×10^{36} (a very large number, since d was in meters). The calculated values of p are shown in figure 5.20 as thin straight lines (—) that appear to connect the measured values ($\circ$, $\circ$). The closeness of the fit is seen by noting that measured and calculated values differ only at 630 Hz and 125 Hz. The differences are less than those found in consecutive tests in the reference condition.

The variability of results in the reference condition added a minor complication to the calculation. The noise of p_{other} was assumed constant during a given day's testing. But, the bearing noise measurements were taken on different days, the reference levels were somewhat lower than average on the day when the smaller bearing clearance was tested (shown in fig. 5.20 $\diamond$). On the day the larger bearing clearance was tested the reference levels were somewhat greater than average. The differences in measured levels were assumed to be related to a difference between p_{other} and p_{other} . Allowance was made by adding the difference to (p_{other}) in the equations involving the larger bearing clearance. This did not increase the number of unknown values calculated, and as evidenced from figure 5.20, did not degrade the accuracy of the curve fit to the measured reference levels.

The calculated noise from the bearings is shown in figure 5.20. The result using equation 65 is shown as a lightly shaded line. Corresponding results for equation 66 are shown by the dark shaded line. The levels from equation 66 (dark shaded line), added to the levels calculated for the other sources discussed above, will produce a spectrum that approximates the reference spectrum. Levels shown by the lightly shaded line require an additional source with a similarly shaped spectrum, to reproduce the reference condition spectrum. In this case the other source would most likely be the big end bearings.

Unexpected systematic errors in the data could substantially effect the results. For example, if systematic errors reduced curves ○ and ○, by 2 dB (comparable to the level of uncertainty in an individual measurement), and also increased curve ◇ by 2 dB, the results of equation 65 could increase by up to 15 dB. Although systematic errors of this magnitude are unlikely, the effect must be considered in interpretation of the results.

Note that in the above curve fitting exercise, the data are not accurate enough to prove or disprove either model. Some benefits would be obtained by measurements on the big-end bearings of the connecting rod. However, the results would not improve with measurements of additional clearances, as normal variations in level would have more effect than the clearance in the bearings. An increased bearing clearance could also affect results. Dubowsky indicates that impacts are more likely with increased clearance, more impacts could cause an increase in the bearing noise. Without a theory to explain the noise, it is more significant to note that the bearings can produce noise, rather than trying to determine which bearing is responsible.

The spectrum shape determined for the bearing noise differs from that obtained from hammer hits on the saw (fig. 5.6), most notably below 1250 Hz. This is most likely an effect of the location of the impacts. A lack of response below 1250 Hz indicates that no significant vibrational modes are excited below that frequency. Note that there is evidence of a single vibrational mode occurring within the 1250 Hz 1/3 octave band. As the vibrational modes become more numerous at higher frequencies, the spectrum could be expected to become smoother, as is observed. The bearing noise frequency spectrum does not extend as high as that found for piston slap, which suggests a smaller impact area, or larger masses involved in the bearing impacts.

On the basis of the spectrum shape, and the lack of other sources to account for the noise between 1250 and 4 kHz, the bearings appear to be a dominant noise source. In figure 5.24 (○), the estimated bearing noise spectrum for normal pulpwood saw operation is plotted. The levels are estimates of gudgeon pin bearing noise, calculated from equation 66 (, fig. 5.20) plus one dB. The resulting A-weighted level produced by the bearings is estimated to be 92 dB. The contribution of the big end bearings to this curve is not known.

5.2.4. Bar chain interaction

A common feature of the residual mechanical noise radiated by all panels is a gradual decrease in radiated noise at frequencies above 2.5 kHz, by approximately 6 dB per

octave, (fig. 5.21, $\diamond$; or 5.5). This is characteristic of the sound radiating from the surfaces of internal combustion engines.¹¹⁷ Reference to the data of figure 5.4 reveals that the high frequency roll off of residual noise commences at a much higher frequency when using the chain, or cutting wood. These observations suggest that there are one or more additional sources of high frequency noise external to the powerhead.

The contribution to this noise from the only attachment to the powerhead, the cutting chain and guide bar, is shown in figure 5.21. Comparison with the residual mechanical noise ($\diamond$) shows that when the cutting chain was installed and driven by the engine (though not cutting wood), a significant increase in the sound radiated by the chain saw occurred at frequencies above 4 kHz, (∇).

In normal operation, use of the chain can result in impacts with the wood (cutting), the guide bar, or the sprocket (polygonal action section 2.8). Consequently, the energy from these impacts will be radiated as noise by the surfaces of the wood, guide bar, or powerhead. The influence of each of these components is described below.

Field testing had originally indicated that cutting had little effect on the total noise of the saw. This conclusion changed due to the reduction of powerhead noise in the reference condition. As seen in figure 5.4 the noise while cutting ($\diamond$), exceeded that found with the chain running (∇), by about 4 dB above 4 kHz.

The surfaces that radiate this additional cutting noise can be inferred by comparison with field results using the acoustical enclosure. While cutting, surfaces outside the enclosure produced levels around 80 dB ($\blacksquare$ fig. 5.2). This is almost enough to account for the difference between cutting and not cutting in figure 5.4 ($\diamond$ versus ∇). Thus, while much of the cutting vibration is radiated as noise by the guide bar and wood, additional noise related to cutting may also be radiated by the powerhead.

Without cutting, the interaction between chain and guide bar also produces noise. This source, was explored further by using scanning acoustic intensity to determine the sound radiated from the bar. The lower spectrum () figure 5.22, which is close to the threshold for reliable acoustic intensity measurements at frequencies below 2.5 kHz, was obtained without a cutting chain, and shows the noise due to powerhead vibration that is radiated by the bar. When the cutting chain was installed and driven by the engine (though not cutting wood), a significant increase in the sound radiated by the guide bar occurred at frequencies above 2 kHz. This source, (" Fig. 5.22), is shown to increase in intensity with frequency, and exceeds that of the powerhead panels (fig. 5.5), at frequencies above

5 kHz. Reference to figure 5.2, (comparing —, with ▣), confirms that this change in intensity is similar to that recorded in field measurements with the saw in an acoustic enclosure. It would therefore appear that operation of the chain can significantly affect the noise of the chain saw at frequencies above 5 kHz.

The acoustic intensity results also show that impacts, sliding and scraping between the bar and chain can produce noise comparable to that radiated by the powerhead. It is apparent from figure 5.21 (▽) that the addition of the chain increases the reference mechanical noise by up to 10 dB at frequencies above 5 kHz. An assumption that this high frequency noise is due to vibration originating in the powerhead implies that the noise from the bar also should increase by 10 dB. Yet, under the same conditions, the intensity radiated from the bar (fig. 5.22) changes by up to 25 dB. A similarly large change was found with the noise radiated from the bar and chain using an acoustic enclosure (—, versus ▣ in fig. 5.2). This indicates that the chain interacts directly with the bar to produce noise. This additional noise from the bar and chain interaction is less significant than the noise from the powerhead at the operator's ear. However, as seen in figure 5.2 (▣), the opposite result is found at 5 m, where the noise from the bar chain interaction can equal the noise from the powerhead (and cutting) at high frequencies.

Much of the chain noise at the operator's ear is radiated by the powerhead, even while cutting. The noise levels at the operator's ear above 5 kHz are about 15 dB larger when the powerhead is exposed (◆ fig. 5.4), when compared to results with the powerhead in an acoustical enclosure (▣). This conclusion is not changed, even when considering the reduced noise produced by the modified saw (◇ fig. 5.4).

The chain interacts with the powerhead through sprocket impacts due to polygonal action (section 2.8 on chain noise). Sprocket impact vibration is coupled to the powerhead through the crankshaft. The noise from this vibration is radiated more efficiently by the powerhead panels than by the bar. Tests without the chain show the powerhead panels (i.e., ▣ fig. 5.5) radiate 10 dB more acoustic intensity than the bar (◇ fig. 5.22) under similar conditions. The same effect could be expected for noise from sprocket vibration. Before the vibration reaches the guide bar, it must first be transmitted through the powerhead.

As suggested in section 2.8 on the prediction of chain noise, it appears that the coupling of sprocket vibration to the powerhead is weak. The predicted noise due to polygonal action causing chain and sprocket impacts was 95 dB for each frequency above 1.2 kHz. This

exceeds the measured results in figure 5.21 (▽) by about 12 dB at frequencies above 5 kHz. Below 5 kHz, figure 5.21 shows no significant change in sound pressure level when the chain was introduced. This indicates that the noise due to chain and sprocket impacts must decrease at frequencies below 5 kHz, just as found in field measurements using an acoustical enclosure.

A better estimate for the noise due to polygonal action is shown in figure 5.24 (▽). This figure was obtained using the noise measured with the chain running in the laboratory (▽ fig. 5.21). Next, the residual mechanical noise (◇ in the same figure) was subtracted from these levels, leaving the noise due to chain impacts. Finally, the effect of chain impacts with the bar was removed, by subtracting the chain noise taken field measurements with the saw in the acoustical enclosure (■), (this is reasonable, as no modifications were made to the chain for laboratory testing). These results were then plotted in figure 5.24 (▽) for frequencies above 5 kHz. There was not enough data to estimate chain noise levels below 5 kHz, although the effect appears minimal.

Figure 5.24 allows a comparison of noise due to chain-sprocket impacts (▽) with that of the bar, chain, and cutting (■ from fig. 5.2). The two sources are comparable, and their sum (--- including a contribution from residual mechanical sources) is similar to that found when cutting in figure 5.4 (◇). Cutting vibration that may be radiated by the powerhead is the only other noise contribution that was not included in figure 5.24.

These results show that the noise from the chain and sprocket interaction, the bar and chain interaction, and cutting are similar in magnitude, with the dominant source determined by the measurement conditions. The surfaces of the powerhead, guide bar, and wood all contribute to the radiation of this noise. Cutting noise affects frequencies above 400 Hz. The chain impacts mainly affect frequencies above 4 kHz, and will be influenced by polygonal action producing impact excitation at the sprocket and bar.

5.3. Source summary

A graphical summary of the sources discussed above is shown in figure 5.24. The corresponding A-weighted results are listed in table 2, and figure 5.23 which show the noise levels estimated for each source on the pulpwood saw as configured for laboratory measurements, and the stock condition.

Figure 5.24 illustrates spectra calculated for the modified saw. Not all entries in table 2 are included in this figure. The exhaust and intake noise were not included as they depend

on muffler design. As an example, table 2 includes levels produced by the stock mufflers and mufflers developed in other work in the laboratory¹¹⁸. The shaking force was omitted, it should dominate the sound pressure levels in figure 5.24 at 125 and 250 Hz, but, should also have a strong dependence on the receiver. The noise of the main bearings was also omitted, as the level was uncertain.

Table 2: Estimated A-weighted sound levels for various sources at the operator's ear cutting wood at best cutting speed. Symbols match those used in figure 5.24.

Source	Symbol	Calculated Modified saw A-weighted level (dB)
fan	✕	92
con-rod bearings	○	92
main bearings		N/A*
piston slap	□	85
crankcase	⊕	81
combustion	⊗	68
Sub Total Reference Condition	—	95
cutting, bar and chain	▣	91 (field)
chain and sprocket	▽	87
Sub Total Chain and Cutting		94
Running Total	---	97
exhaust		87
intake		85
shaking force		81
TOTAL		98

* best estimate for main bearing noise is 80 dB (using motored fan)

The agreement between the measured and calculated results is quite good. The reference condition sub total, and final total in table 2 are respectively 95 and 98 dB (A-weighted) compared to 97, and 97 dB (A-weighted) levels measured in field testing. Inspection of figure 5.24 shows agreement between the calculated total for the laboratory reference condition (—), and actual measurements within ± 1 standard deviation at all frequencies except those dominated by fan noise. When cutting is included, the calculated spectrum in figure 5.24 (---) is about 2 dB lower than that found in field testing (◇ fig. 5.4), at frequencies above 5 kHz. This was the approximate difference found between laboratory and field measurements (fig. 3.10). Other differences between the calculated cutting noise in figure 5.24 (---), and measured levels (◇ fig. 5.4), can be attributed to

differences between laboratory and field testing (which are likely due to the human operator).

The most significant differences found in the summation of sources in figure 5.24 (—) occur at frequencies where the fan noise appears to be the dominant source, most notably at 630 and 800 Hz. Previous results using the dummy fan (section 5.2.3.1 fig. 5.7) suggested the fan was wholly responsible for noise at these frequencies, although this was not supported by measurements when using the motored fan (fig. 5.8). It thus appears that the motored fan results underestimate the noise below 630 and 800 Hz by about 6 dB. The other frequency where the level summation and residual mechanical noise differ, is at 3150 Hz, (the fan blade passage frequency). In this case, the motored fan appears to have produced a 2 or 3 dB overestimate of the noise level. It is thus reasonable to conclude that the motored fan results underestimate the low frequency levels and overestimate the high frequency levels.

Figure 5.23 summarizes the A-weighted noise sources in the pulpwood chain saw. The noise of the saw depends on the relative strengths of seven major sources. In the stock condition (grayed area) the total of these sources is 106 dB A-weighted, requiring the use of hearing protection. Improved intake and exhaust mufflers have been demonstrated (shown by vertical stripes), but without treatment of the mechanical noise sources, the potential reduction is only a few dB. At this point the noise would be determined by the gas forces, cutting chain, fan and bearings.

The prospects for reducing the remaining sources responsible for engine noise within the required design constraints appear to be limited. Only modest reductions in the sources of mechanical noise, presumed to result from piston slap and bearing impacts, have been achieved so far in conventional engine designs. It is probable that redesign of the surface panels, first to enclose the powerhead, and then to reduce the sound radiated will prove more effective (as shown by horizontal stripes in figure 5.23). However, there will be unavoidable penalties in size and weight. The potential reduction in A-weighted sound level at the operator's ear could, however, be in excess of 10 dB. With this reduction in engine noise, the level at the operator's ear would then be determined by the cutting process itself (labeled chain), which is dependent on wood type.

Chapter VI Conclusions

The noise of chain saws appears to depend on the relative strengths of seven major sources, four associated with the engine and three with cutting wood. They are the exhaust, intake, fan, bearing impacts, the interaction between the cutting chain, guide bar and sprocket, as well as the cutting process itself. When cutting wood in the stock condition, the far-field noise is determined by the exhaust at frequencies below 500 Hz, by the fan between 500 and 1250 Hz, by bearing noise radiated primarily by surfaces rigidly coupled to the crankcase at frequencies between 1250 and 3125 Hz, and finally by the chain, and cutting at higher frequencies.

Field testing showed the relative noise source strengths on the two saws to be comparable, with noise levels from the large timber saw averaging about 5 dB greater than the pulpwood saw (table 1). Theory indicates that this difference can be related to the speed difference between the two saws. The main exception is the exhaust noise, which was 13 dB higher at the operator's ear, but equal in level in the far field. This suggests that the exhaust directivity can be used to advantage in the reduction of exhaust noise experienced by the operator.

As supplied by the manufacturer, the exhaust noise of the pulpwood saw was approximately 106 dB (A-weighted) at the operator's ear. Removal of the muffler increased this noise source to 116 dB. The open exhaust noise levels measured in this thesis (between 116 and 129 dB A-weighted at the operator's ear) easily exceeded the noise from other sources. Thus measurement of this noise source could be undertaken without modifications to the rest of the saw.

The intake noise of the pulpwood saw as supplied by the manufacturer, was approximately 104 dB (A-weighted) at the operator's ear. The unmuffled intake noise level was 108 dB (A-weighted), and could be determined with the exhaust noise piped away.

Fan noise was determined by externally motoring the saw, which resulted in levels of 92 dB (A-weighted) at the operator's ear. There appeared to be some difficulty in determining the fan noise levels at 630 and 800 Hz on the pulpwood saw. The difficulty could be a result of the inability reproduce the running conditions of the saw by motoring the fan. The difficulty could be related to improper positioning of the fan in the motored condition, thermal expansion, or applied loads affecting the positioning of components.

The fan noise can be attributed to turbulence and aeolian tones. Handbook calculations⁴² suggest that the noise for a properly designed fan should be lower than those measured. Theory based on aeolian tones can account for the noise of the fan; however, accurate noise level predictions require similar accuracy in the estimation of flow velocity through the fan.

A complete theory for predicting bearing noise impacts was beyond the scope of this thesis. On the basis of spectral shape, and consideration of other sources, an overall level of 92 dB (A-weighted) at the operator's ear was deduced. A demonstration of the frequencies affecting bearing noise was obtained by using increased bearing clearance. This method must be used with caution, as Dubowsky predicts that increased bearing clearance can affect the likelihood of impacts (eqn. 59). Since bearing noise appears to be a significant source it may be advantageous to use a more comprehensive analysis to determine bearing noise (i.e., see reference 82). Conversely, if bearing impacts can be substantially reduced, there is no need to predict the noise levels. To this end, the impact predictors of Earles (eqn. 57) or Dubowsky (eqn. 59), could be used to attempt to reduce the impacts. This would likely involve force and form closure (using springs) as suggested by Fawcett¹²⁹, and Perera⁸⁶.

The noise from cutting and impacts between chain and guide bar is important since the chain must be used to cut the wood, and may ultimately determine the noise of the saw. The noise of this source was 91 dB (A-weighted) at the operator's ear when measured using an acoustical enclosure to eliminate the noise radiated by the powerhead. While no theory is available for cutting noise, torque and speed fluctuations due to polygonal action could affect the intensity of impacts between chain and guide bar. Of note, the effect of speed on cutting noise was not as significant as it was for the noise from other sources. This suggests factors such as cutting rate, or saw power may be more significant to this noise source.

Direct measurement at high frequencies indicated the noise level from chain and sprocket impacts was 87 dB (A-weighted) at the operator's ear. These impacts are likely due to chain movement caused by polygonal action. Noise calculations based on approximate impact power and mobility overestimated measured levels by 12 dB. Accounting for friction may improve this noise estimate. It would also help to measure the mobility of the sprocket impact areas. This is an area of active study and better methods of estimation for this source may be forthcoming.

Piston slap produced noise levels of 85 dB (A-weighted) at the operator's ear. This noise is due to piston impacts with the cylinder wall. The noise levels were estimated based on direct measurement with increased piston running clearance, and appropriate theory. Calculated levels were 12 dB higher than the measured values. Measurement of the piston and cylinder wall mobility could be expected to improve this estimate, although similar differences were found elsewhere between experimental and measured results⁶⁴. This suggests improvements are needed in the basic theory for the piston energy, to account for additional factors such as friction.

The crankcase (or intake system), on the stock saw produced noise levels around 102 dB (A-weighted) at the operator's ear. This masked the level of mechanical noise from other components. The noise was due to the transmission of sound through the walls of the intake system. Careful elimination of leakage through the intake system reduced noise levels to 81 dB (A-weighted).

The noise levels from the crankcase were estimated from crankcase pressure and intake insertion loss. The crankcase pressure was measured with a commercial pressure transducer, and the insertion loss (or transmission loss), was obtained by acoustical excitation of the crankcase. This method has the advantage of simplifying noise reduction. With acoustical excitation of the crankcase, simply putting one's hand on an area of high sound radiation can introduce sufficient damping to reduce the noise, and so identify the an area that needs treatment.

The rigid body vibration of the chain saw could be expected to produce 81 dB (A-weighted) noise at the operator's ear. This noise is due to dipole radiation from vibration which is caused by the reciprocating mass of the piston, connecting rod, and crankshaft. This noise was not exceptionally loud, and thus was not studied in depth. Estimates were based on acceleration, and the noise at the operator's ear position was overestimated as calculations were made for noise in the direction of maximum sound radiation.

Combustion noise produced an almost unmeasurable 68 dB (A-weighted) at the operator's ear. This is surprising, as this source has been significant in other engines. Combustion noise is due to the cylinder pressure and stiffness controlled transmission loss of the engine structure. The pulpwood saw presented a special problem in measurement of this source, since the excitation from combustion pressure drops off significantly with increasing frequency and, without vibration isolation, this became unmeasurable above 1 kHz. However, the cylinder pressure level at this frequency was still about 160 dB. A structural

attenuation of over 60 dB is required to reduce combustion noise to the levels of the loudest source. Although this level of structural attenuation was far exceeded by the pulpwood saw, the same could not be arbitrarily assumed for another saw. Measurement of this source required design of a vibration isolation system for the pressure transducer, and measurement equipment of the highest quality.

As may be deduced from the above, a reduction in A-weighted sound level of only 3-4 dB can be expected by improving exhaust and intake mufflers, before noise from the powerhead will determine the overall noise of these saws. Additional modifications to the powerhead show potential for a further 10 dB of noise reduction. Under these conditions the A-weighted sound level at the operator's ear is due to the cutting process, which is influenced by wood type, and produces a sound level of, typically 91-96 dB (A-weighted).

Appendix A. Measurement of Source Characteristics Using Reciprocity

Reciprocity allows acoustical measurements to be made with the source of excitation placed at the operator's ear position instead of inside the chain saw. A sound source outside the chain saw can be larger and more powerful than a source inside the saw.

A good application for reciprocity is combustion noise. Increased dynamic range and added flexibility using reciprocity are especially important for combustion noise measurement because of the high transmission loss of sound passing through the cylinder wall. Under normal circumstances an explosion is required within the cylinder to allow sound pressure levels at the operator's ear position to exceed the laboratory background noise. The reciprocal measurement on a non running chain saw, places the sound source in the laboratory, making it relatively simple to exceed the background noise.

The principle of acoustical reciprocity states that for a given volume velocity source and receiver pair, the sound pressure at the receiver is unchanged if the source and receiver positions are exchanged¹¹⁹. The principle does not require the sound transmission medium to be homogeneous¹¹⁹, but does require that the system is passive, linear, and time invariant^{120,121}.

In a passive system, all components only transmit sound energy, there are no sound energy sources. The presence of additional sources would alter the pressure the receiver as well as the original source.

A linear system allows superposition¹²⁰, the output acoustical energy is a weighted sum of the energy in the input signals. A rattle, or buzz in the transmission path is an example of a non-linear effect.

For the system to be time invariant, the output response of the system will always be the same for the same input. A system with moving parts may change significantly during the period of excitation, and will not be strictly time invariant. In large moving machinery, an assumption of time invariance will be further degraded by propagation delays. However, for a compact machine, if the excitation frequency is high compared to the cyclic frequency, any two contacting parts of the machine can be approximated as stationary for a short time during the engine cycle¹²⁰.

From Pierce¹¹⁹ we obtain a general formula for reciprocity:

$$\frac{\hat{p}_a(\mathbf{R}_2)}{\hat{Q}_a} = \frac{\hat{p}_b(\mathbf{R}_1)}{\hat{Q}_b}$$

67

where $\mathbf{R}_1$ and $\mathbf{R}_2$ are position vectors relative to the origin; $\hat{Q}_a$ is the complex source strength amplitude of point source 'a' such that $\Re\{\hat{Q}_a e^{-i2\pi t}\}$ represents time rate of volume efflux from the source; $\hat{Q}_b$ is as above for point source 'b' (at $\mathbf{R}_2$); $\hat{p}_a(\mathbf{R}_2)$ is the complex pressure field amplitude at point $\mathbf{R}_2$ due to source $\hat{Q}_a$ at $\mathbf{R}_1$; and $\hat{p}_b(\mathbf{R}_1)$ is the complex pressure field amplitude at point $\mathbf{R}_1$ due to source $\hat{Q}_b$ at $\mathbf{R}_2$.

Equation 67 can be applied to many situations, complex quantities are used to account explicitly for the phase of the pressure and volume velocity. It is not immediately apparent how to account for the directivity of the noise source. For example, if a directional source were moved to the receiver position, the orientation would significantly affect the results obtained from equation 67. However, the equation can be applied to point sources, which radiate noise equally in all directions. As shown by Lord Rayleigh¹²², a directional source can be constructed from some combination of simple point sources, the equation can be applied to each source, and the results combined through superposition. This allows equation 67 to be used for a variety of sources.

The advantages of reciprocity are that the receiving transducers tend to be very small, and there can be fewer restrictions on the source transducer characteristics. Since reciprocity deals with two different quantities, the volume velocity of the source and the pressure at the receiver, interchanging the two can frequently increase the signal to noise ratio at the receiver. As employed in this thesis, reciprocity measurements are used to define the attenuation of combustion noise by the machine structure. Thus rather than putting a high volume velocity source inside the cylinder, and measuring a small pressure signal at the receiving position, a comparatively larger pressure can be measured inside the cylinder with a smaller volume velocity source at the normal receiver position.

Although reciprocity can be applied to chain saw noise, caution must be used. The system is not passive due to the presence of multiple noise sources, thus the saw must be considered to be composed of independent sources that can operate alone. Oil films, contact stiffness, rattles and machine components are frequently nonlinear¹²¹. Due to the presence of moving parts, the system will not likely be time invariant. One way to

eliminate other sound sources in the chain saw and make the system time invariant, is to take measurements with the chain saw crankshaft fixed in position.

Thus as mentioned earlier, combustion noise is a good application for reciprocity. The peak pressure excitation occurs at a well-defined point (top dead center). Measurements can be made with the crankshaft fixed in position at top dead center. This makes the system time invariant and eliminates other noise sources present when a chain saw is running. Combustion noise is linearly related to the cylinder pressure, so non-linear propagation paths are not a part of the measurement.

A.1. Volume velocity in a small cavity

For simplicity, measurement of reciprocity requires a point source at the receiver position. Although pressure measurement is straightforward, the volume velocities of the original and reciprocal noise sources are required. This is straightforward in free space, in a small enclosed area, calculation of the volume velocity can be obtained from pressure measurement if the boundary conditions are known. For example, for free field propagation far from a point source, the volume velocity normal to the enclosing surface is⁵⁷:

$$\hat{Q} = UA \quad 68$$

where $\hat{Q}$ is the volume velocity; U is the particle velocity normal to surface; and A is the surface area.

Near a point source in a free field measurement⁵⁷:

$$\hat{Q} = p \frac{4\pi R (c + i2\pi fR)}{\rho_0 c i2\pi f} \quad 69$$

where p is sound pressure; ρ_0 is the air density; c the speed of sound in air; f is frequency; and R is the distance to the measurement position.

Using transfer matrices, the required volume velocity of a source within the chain saw cylinder can be calculated from the measured pressure. Assume plane wave propagation, and a cavity of the form shown above. L is the characteristic dimension of the cavity ($= \sqrt[3]{\text{volume}}$), x is the insertion point of extra gases. The cavity is separated in two sections by an imaginary vertical plane positioned at the flush with the end of the inlet tube, section 2 contains the inlet tube, and section 1 is the remainder of the cavity. Section 3 is the inlet tube with dimensions length $= \lim_{L_3 \rightarrow 0} L_3$, diameter $= \lim_{D \rightarrow 0} D$. Position 0 is at the (right hand) wall of the cavity.

$$\begin{bmatrix} p_3 \\ V_3 \rho A_3 \end{bmatrix} = \begin{bmatrix} \cos\left(\frac{2\pi f}{c} L_3\right) & iZ_{s3} \sin\left(\frac{2\pi f}{c} L_3\right) \\ \frac{i}{Z_{s3}} \sin\left(\frac{2\pi f}{c} L_3\right) & \cos\left(\frac{2\pi f}{c} L_3\right) \end{bmatrix} \begin{bmatrix} \frac{1}{Z_{s2}} \tan\left(\frac{2\pi f}{c} L_2\right) & 0 \\ 1 & 1 \end{bmatrix} \quad (70)$$

$$\times \begin{bmatrix} \cos\left(\frac{2\pi f}{c} L_1\right) & iZ_{s1} \sin\left(\frac{2\pi f}{c} L_1\right) \\ \frac{i}{Z_{s1}} \sin\left(\frac{2\pi f}{c} L_1\right) & \cos\left(\frac{2\pi f}{c} L_1\right) \end{bmatrix} \begin{bmatrix} 1 & Z_0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} p_0 \\ V_0 \rho A_0 \end{bmatrix}$$

where V is the velocity in each section, c is the speed of sound in air, ρ is the air density, A is the cross sectional area of the section, Z_0 is the acoustical impedance of the end wall, and Z_s is the specific acoustic impedance of the section, defined by c/A .

assume:

$$V_0=0$$

$$L_3=0$$

Simplifying equation 70:

$$\begin{bmatrix} p_3 \\ V_3 \rho A_3 \end{bmatrix} = p_0 \begin{bmatrix} \cos\left(\frac{2\pi f}{c} L_1\right) \\ \frac{i}{Z_{s2}} \cos\left(\frac{2\pi f}{c} L_1\right) \tan\left(\frac{2\pi f}{c} L_2\right) + \frac{i}{Z_{s1}} \sin\left(\frac{2\pi f}{c} L_1\right) \end{bmatrix} \quad 71$$

Noting that the cross sectional area of the cavity is $A_1=A_2=L^2$, thus:

$$Z_{s1} = Z_{s2} = \frac{c}{L^2} \text{ and } L = L_1 + L_2 \quad 72$$

and then the matrix simplifies to:

$$\begin{bmatrix} p_3 \\ V_3 \rho A_3 \end{bmatrix} = p_0 \begin{bmatrix} \cos\left(\frac{2\pi f}{c} L_1\right) \\ \frac{iL^2}{c} \frac{\sin\left(\frac{2\pi f}{c} (L_1 + L_2)\right)}{\cos\left(\frac{2\pi f}{c} L_2\right)} \end{bmatrix} \quad 73$$

Thus the solution for V_3 is:

$$V_3 = p_0 \frac{iL^2}{A_3 \rho c} \frac{\sin\left(\frac{2\pi f}{c} L\right)}{\cos\left(\frac{2\pi f}{c} L_2\right)} \quad 74$$

Note the dependence on L_2 : The pressure response on the wall varies depending on the location of the source within the cavity. If $L_2 = 0$, or if the impedance at section 3 becomes the lumped impedance for a cavity and the volume velocity of the source is given by:

$$\hat{Q} = p_0 \frac{iL^2}{\rho_3 c} \sin(2\pi f L/c) \quad 75$$

This equation is to be applied to the combustion chamber of the pulpwood chain saw, which, at top dead center has typical dimensions of 2 cm for L . Thus the equation becomes infinite at a frequency between 2 and 4 kHz at room temperature (depending on L). Since the speed of sound is proportional to the square root of the temperature, the corresponding frequency during saw operation will be much higher. Thus the small angle approximation can be used for the sine and cosine, and the results are then dependent on the cavity volume. For a point source in a small volume with equal dimensions¹²³:

$$\hat{Q} = V_3 A_3 = p 2\pi f \frac{iL^3}{\rho_o c^2} \quad 76$$

where p is the pressure at the wall, and L is the characteristic dimension of the cavity.

Thus all required parameters for reciprocity calculations (eqn. 67) can be obtained from simple pressure measurements.

Appendix B. Effect of Transducer Movement on Pressure Measured with a Vibration Isolated Pressure Transducer

Vibration isolation allows the pressure transducer to move, which changes the cylinder volume, and affects the cylinder pressure. It was necessary to show that the pressure changes induced by transducer movement had less effect on measurements than the acceleration sensitivity of the transducer.

The following derivation is used to determine the magnitude of measurement errors due to pressure changes induced by the cylinder pressure transducer movement. The acceleration and pressure sensitivities are determined by the same first order system, and are related by a constant. Thus a comparison with acceleration sensitivity is appropriate.

For adiabatic volume changes, the pressure is determined by:

$$p = \frac{p_o v_o^\gamma}{v^\gamma} \quad 77$$

where v is the specific volume, p is pressure, γ is the specific heat ratio, and subscript o indicates some reference condition. Taking the derivative with respect to the specific volume:

$$\frac{\partial p}{\partial v} = \frac{-\gamma p_o v_o^\gamma}{v^{\gamma+1}} \quad 78$$

for a small volume change $v \approx v_o$, and the equation simplifies to:

$$\Delta p = -\Delta v \frac{\gamma p_o}{v_o} \quad 79$$

This can be expressed in terms of the transducer displacement by first noting that the change in volume is related to the cross sectional area of the transducer (A) and transducer displacement (Δx), where:

$$\Delta x A = m_g \Delta v \quad 80$$

where m_g is the mass of gas in the cylinder.

Thus substituting for Δv the change in pressure becomes:

$$\Delta p = -\Delta x A \frac{\gamma p_o}{m_g v_o} \quad 81$$

The mass of the transducer assembly (m) can be related to the mass in a single degree of freedom system with base excitation. Solving for the transducer displacement Δx in terms of the base (i.e., cylinder) displacement Δz , and substituting for Δx ¹²⁴:

$$\Delta p = - \left(\frac{m(2\pi f)^2 \Delta z}{\sqrt{(K - m(2\pi f)^2)^2 + (C(2\pi f))^2}} \right) A \frac{\gamma p_o}{m_g v_o} \quad 82$$

And replacing second derivative of the base displacement ($\Delta z 4\pi^2 f^2$) with the symbol (acc) for the cylinder acceleration (as a function of frequency). Below the isolator resonance, the transducer displacement approximates that of the chain saw, and the induced pressure is related to the acceleration by a constant, i.e.:

$$\Delta p = -acc \frac{\gamma A p_o m}{m_g v_o K} \quad 83$$

Converting both sides to decibels and solving at cylinder top dead center, by substituting:

$m = 0.3 \text{ kg}$ (vibration isolated mass)

$f_o = 292 \text{ Hz}$ (natural frequency of vibration isolated transducer)

$K = m(2\pi f_o)^2 = 1 \times 10^6 \text{ N/m}$

$A = 7 \times 10^{-5} \text{ m}^2$ (transducer area)

$\gamma = 1.4$

$p_o < 3000 \text{ kPa}$ (estimate of top dead center pressure from measurements)

$m_g v_o = 67/8 \text{ cc}$ (volume at top dead center)

the result is:

$$SPL < AL - 6 \text{ dB}$$

where AL is the acceleration level in dB re 10^{-6} m s^{-2} , and SPL is the sound pressure level in dB re $2 \times 10^{-5} \text{ Pa}$

Above the isolator resonance, the induced pressure varies as f^2 .

$$\Delta p = -acc \frac{\gamma A p_o}{m_i v_o (2\pi f)^2} \quad 84$$

Both the transmissibility of acceleration and the induced pressure are due to the same first order system, thus the induced sound pressure level in dB (re $2 \times 10^{-5} \text{ Pa}$) will always be at least 6 dB below transducer acceleration in dB (re 10^{-6} m s^{-2}). The induced pressure can be ignored as the transducer acceleration sensitivity (appendix G.3.2) will produce a false pressure reading that exceeds the induced pressure by 11 dB.

Appendix C. Other Analysis Techniques Explored

In this initial source separation, noise sources found on the chain saw were most easily identified with conventional techniques. While more recent techniques could be used to advantage once the sources are known and understood, only a few techniques showed potential usefulness for the present analysis. Narrow band analysis provided more detail in the displayed spectra. The Wigner distribution (related to short term FFT analysis) showed impacts as well as frequency variations that might assist in source identification. Higher order derivatives of the acceleration time history made identification of individual impact events easier.

Averaging in the time domain was also attempted. Triggering was based on high pass filtering of the cylinder or oil pump acceleration, which should exemplify the piston slap events. The acceleration time history was played back slowly, and each trigger event was manually checked before including it in the average. The recovered cylinder acceleration waveforms had a dominant spike associated with the trigger event. Approximately 0.16 msec later a few cycles of a sinusoidal signal (approximately 5.5 kHz) was observed at the oil pump. This behavior was observed regardless of the location of the triggering accelerometer, and suggested that the highest frequencies were related to the piston. Since previous results associated piston slap with high frequencies (section 5.2.3.5), this behavior was not surprising.

A measure of coherent output power was attempted. By using the cylinder pressure transducer, the coherence between the cylinder pressure and the radiated sound was measured. Much of the noise radiated was coherent with cylinder pressure when the saw speed was held constant. The coherence was also found to vary depending on operating conditions. This suggested other sources could also be coherent with the cylinder pressure, artificially increasing its apparent contribution to noise. As a check, the saw speed was varied continuously to reduce coherence between cylinder pressure and other sources. As expected, the coherence between radiated noise and cylinder pressure dropped significantly. This variable speed test indicated a negligible contribution due to combustion noise, the same conclusion as was found using reciprocity. However, the validity of measurements taken with varying speed could be criticized. The signal was not stationary during the measurement period. The analyzer was not fast enough to obtain enough overlap in the time domain for a true measurement of power. Other sources could still be coherent with the cylinder pressure over the range of speeds tested. For these reasons neither measurement was used.

Coherent output power measurement was also attempted for piston slap impact noise. These measurements were likely meaningless. The same problems discussed in the previous paragraph on cylinder pressure coherence would be experienced. In addition, separate measurements of hammer impacts to the saw showed that the coherence of vibration decreased with distance from the impact site, even though the saw vibration was due solely to the impact.

Cepstrum measurements were equally unsuccessful. This technique was expected to show effects due to time delays, reverberation, or different rotational frequencies of components such as the bearings. Unfortunately, the speed variations in the saw reduced the quality of the output so that the only apparent feature was the repetition rate of the saw firing frequency.

A technique similar to the Wigner distribution is a gated spectrum measurement. Gating the input to the 1/3 octave analyzer keyed to the engine rotation speed allows a measure of the change in spectrum versus crankshaft position. This technique is subject to many of the same limitations of FFT analysis. As the gating time must be less than the period of one engine revolution, the lowest possible valid frequency is the engine firing frequency. If each revolution was split into 8 spectra, the lowest possible frequency would be 8 times the firing frequency or 1000 Hz for the pulpwood saw. There were also effects at higher frequencies, the gating process chopped the input signal so that the input filters received a discontinuous signal at the turn on and turn off of the gate. Objections aside, the technique produced a plot of spectra versus time. The frequency limitations, and poor resolution in the time and frequency domain did not easily allow identification of a specific sources from the output graph. The uncertainty in time relationship between source event and radiated sound made the plots difficult to interpret.

Vibration intensity (similar to sound intensity) was considered, and then abandoned due to the unavailability of appropriate equipment. Four separate sensors would have been required to describe the surface vibration in one dimension¹²⁵, as well as a description of the surface. Currently available equipment to measure sound intensity can be used with accelerometers instead of microphones. But this allows only two inputs, limiting the usefulness to simple structures, where the limited information provided can be used to trace the source of vibration energy¹²⁶. Unfortunately, the chain saw surface has many curves, ribs, and changes in thickness, which without detailed computations, could easily confuse the analysis.

Appendix D. Prospects for Noise Reduction

Current standards for the noise of chain saws are 106 dB (A-weighted) at the operator's ear during cutting operations. These levels exceed 85 dB (A-weighted), at which point, an increased risk of hearing loss is measurable (90 dB is recommended for an 8 hour day)¹²¹.

Some success in noise reduction was demonstrated during the investigation of chain saw noise sources. In order to perform the source separation on the pulpwood saw, small modifications were made to reduce the noise levels of unnecessary sources. These measures in conjunction with the improved exhaust and intake mufflers reduced the cutting noise at the operator's ear to 98 dB (A-weighted). Acoustical enclosure of the powerhead was shown to reduce the noise further to 91 dB (A-weighted) at the operator's ear. As a first attempt, a reasonable target for noise reduction would be to bring the powerhead noise sources below the cutting noise (91 dB A-weighted). At this sound pressure level, the cutting process need not be disturbed, and cutting would be possible during the much of the day without requiring hearing protection.

Although chain saws may seem unusually loud, comparison with other mechanisms in chapter II allowed reasonable predictions of chain saw noise levels. A comparison of machines based on mechanical output power shows chain saws are neither exceptionally loud, nor exceptionally quiet¹²⁷. If the sound output of a loud machines (i.e., some aircraft) were scaled to that of a 3 kW chain saw, they would produce noise levels as high as 115 dB (A-weighted) at the operator's ear. Under the same conditions quiet machines (i.e., dishwashers)¹²⁷ would produce the equivalent of 75 dB (A-weighted) at the operator's ear.

D.1. Exhaust

Any effective noise reduction of the pulpwood and large timber chain saws will first require reduction of the exhaust noise. Reduction of the exhaust noise below that of the chain requires a 30 dB total insertion loss. Identification of other noise sources was pointless unless a high performance exhaust muffler was possible.

A high insertion loss muffler was constructed¹¹⁸. As the insertion loss was increased, wave motion effects in the muffler limited the insertion loss at frequencies corresponding to the length and volume resonances. Control of these effects was obtained through resistive perforated metal screens in the muffler and an additional cavity to dampen and absorb the resonances. The result was a muffler producing peak insertion loss of 40 dB a:

1 kHz (30 dB higher than stock muffler), with power loss comparable to the stock muffler, and twice the stock muffler volume. The tailpipe diameter of the new muffler was generous (area exceeded that of the exhaust port), so possible improvements in power or size are possible.

The potential for exhaust noise reduction was demonstrated, so identification and reduction of other sources became practical.

D.2. Intake

During this study, it was necessary to seal leaks in the intake system, and provide an attachment for ducting the inlet. With the addition of a 15 cm long inlet pipe, these simple measures improved the intake insertion loss to 30 dB around 800 Hz, and 20 dB at higher frequencies. This was sufficient to drop the A-weighted intake system noise from over 100 dB to 85 dB. The resulting intake muffler was power neutral, however, use of the inlet air filter reduced power a few percent.

D.3. Powerhead noise

The powerhead contains a number of sources that produce similar noise levels. This study has shown two methods that can reduce most sources of noise: speed and enclosure. Within the speed range tested, (3800 to 12000 rpm), the noise decreased approximately 15 dB each time the speed was halved. A common characteristic of quieter machines appears to be partial enclosure of the noise producing mechanisms. Regardless of other design modifications this may be necessary to achieve maximum quieting of chain saws. Without modification of the chain, or cutting process, the measurements in this study have shown that a noise reduction to 91 dB (A-weighted) is possible with the pulpwood saw enclosed, (note, intake and exhaust mufflers also must be provided).

An acoustical enclosure need not be bulky, heavy, or complicated. The aluminum enclosure used for testing purposes was unworkably bulky, and heavy. Other tests have shown that (with intake and exhaust ducted) a simple cardboard box lined with 2.5 cm thick fiberglass reduced cutting noise to levels comparable to those using the aluminum enclosure.

Additional modifications could make an enclosure smaller. Not all surfaces of the saw need to be enclosed. Acoustic intensity measurements, (fig. 5.5), showed the existing vibration isolation of the handles reduced the radiation from the handle surfaces by at least 10 dB. These surfaces need not be enclosed, and an enclosure for the remainder of the

powerhead surfaces could be directly attached to the handles. This would eliminate the need for additional vibration isolation, and reduce the size of the enclosure. As an example, in tests using a cardboard box for an enclosure, the box was not vibration isolated from the chain saw, it simply rested against the handles.

Alternatives to speed reduction, or enclosure are to either reduce the total vibrational energy, or to reduce the amount converted to acoustical energy. Equation 21 indicates that each time the material density (ρ_m), damping (η), or thickness (h) is doubled results in a 3 dB reduction in the amount of vibrational energy converted to acoustical energy. Damping can be increased by external damping layers, or by increasing the amount of surface to surface contact between saw components. Increasing the damping of radiating panels, or modification of the stiffness, thickness, and shape have shown reductions of up to 10 dB in sound radiation^{2,76,128}.

Increased thickness and density could further reduce noise due to their effect on the structural impedance (eqn. 30 and 28). Equation 23 indicates the vibrational energy (P_v) could be reduced by increasing the structural impedance (Z). From equations 30 and 28, the structural impedance can be proportional to thickness squared (h^2) or root of density ($\sqrt{\rho_m}$). Allowing for both the structural impedance, and acoustical energy conversion; a doubling of thickness could result in a total of 9 dB noise reduction, a density doubling increase could result in 4.5 dB reduction. These both increase mass which is undesirable, but, an increase in thickness with a reduction in density could maintain the total mass, and reduce the noise radiated¹²⁸.

Equations 30 and 28 suggest that increasing the structural impedance is possible by increasing stiffness. Caution must be used when increasing bending stiffness by adding ribs on exposed panels. At higher frequencies adjacent areas on a panel can vibrate out of phase which effectively reduces the radiated noise⁹. Addition of ribs can interfere with the natural bending modes preventing this cancellation from occurring.

During this study effectiveness of damping materials was not seriously tested. When a small amount of damping was added between the cylinder fins, the vibration levels and surface area were too small to make a noticeable difference to the measured noise. However, one change that may have increased damping was the aluminum gasket added underneath the cylinder

D.4. Fan

The cooling fan produces almost half the noise radiated from the powerhead. It is a radial fan with forward curved blades (which is often noisier than other fan types⁴²). The frequencies where the fan noise was dominant also showed the most variation from test to test, suggesting a sensitivity to slight changes in the fan mounting. Removal of the inlet grill did not affect the motored fan noise, thus any effect of mounting is due to conditions downstream of the fan.

Comparison with predicted fan noise levels suggests large reductions (>10 dB) in fan noise may not be possible without significant changes in the fan design. A different fan might be quieter, but could reduce the flow velocity or volume flow, requiring increases in the area of the cylinder cooling fins, and size of the saw.

Improvements in fan efficiency should also reduce the noise, but these will likely increase the size of the passages leading to and from the fan, thus increasing the saw volume. Eck indicates that based on the specific speed of the pulpwood saw fan, a shorter blade with a longer chord is appropriate⁴⁴. Better matching of the fan blade entry angle to the streamline angle could reduce noise from aeolian tones. These changes could also increase the size or complexity of the fan.

As with other noise sources, this study has shown the existing fan noise can be reduced by slowing the fan rpm, or by enclosing it in an acoustically absorbing duct.

D.5. Bearings

The bearings are the next most significant noise source after the fan. The needle bearings employed in the chain saw use standard clearances. Reducing the bearing clearance may reduce noise, but will more likely cause additional problems due to the chain saw operating conditions, (i.e., thermal gradients, high speed, airborne lubrication, and particulate contaminants). For example, in one experiment, the gudgeon pin bearing clearance was reduced by half. In measurements, the modified clearance had little effect on the noise, except a possible level increase at around 4 kHz, (where piston slap is dominant), and a minor 0.5 dB decrease above 5 kHz. Due to the short length of the rolling elements in the bearings, the reduced clearance caused intermittent binding and stiffness in turning. Thus along with increased wear, the clearance reduction could produce a torque affecting the forces and movement of the piston and connecting rod.

A significant reduction in bearing free play, (and noise), requires different bearings. For example, the ball bearings on the crankshaft can effectively run without free play. An alternative to reduction of bearing free play has been suggested by Fawcett¹²⁹. His measurements have shown bearing impacts only occur when the force on the bearing becomes small and then changes direction. Expressed on a polar plot this is equivalent to the bearing force approaching the origin. To prevent a loss of contact and subsequent impact in the bearings, a preloading force could be applied with a spring to the bearing to shift the force polar plot away from the origin. Perera has demonstrated that bearing impacts can be also be eliminated using a torsion spring⁸⁶.

The bearing impacts occur within the chain saw, but the noise is radiated from surfaces of the saw. If the vibration was prevented from exciting the surfaces by using vibration isolation, the noise would not be radiated. As discussed previously, since noise from bearing impacts will be radiated by powerhead surfaces, any treatment of these surfaces will be of benefit. Enclosure of the saw, and vibration isolation of the external panels is one option that has been shown effective. In the case of the bearings, it may be possible to isolate the vibration from the radiating surfaces of the saw. Vibration isolation would only be required for frequencies where the radiation efficiency is highest, say above 630 Hz, so the isolators would be smaller and stiffer than those used to vibration isolate the chain saw handles.

An attempt was made to vibration isolate the pulpwood saw crankshaft bearings from the crankcase, (fig. 3.11) by compliantly mounting them in rubber. The vibration isolation was somewhat successful, but the overall saw noise was not reduced. The main crankshaft bearings were replaced with bearings with smaller external dimensions, (but same bore diameter). The smaller bearings were pressed into steel retaining cups. The outside of the retaining cups were then wrapped in 1.6 mm thick silicone rubber, and concentric 1.6 mm thick Viton o-rings were placed on the ends of the retaining cups. This assembly was then pressed into the existing bearing retainers on the crankcase.

The resonant frequency of the crankshaft and bearings was set at 630 Hz, to minimize crankshaft movement, and yet still provide attenuation at frequencies over 1 kHz, where the radiation efficiency of the chain saw is high. Due to the compliance of the rubber isolators, it was not possible to attain the original axial preload on the bearings. To allow for the additional crankshaft movement, the interior of the crankcase was made smooth and cylindrical so that irregularities in the crankcase surface could not contact the moving crankshaft. The oil seals on the crankshaft were also removed, and the crankcase seal at

the shaft was obtained by using sealed bearings, and by leaving the outboard seal intact. The inboard seal was removed to allow lubrication by the oil in the crankcase.

Vibration measurements of the isolated bearings on a running chain saw confirmed that the target resonant frequency had been obtained, but scuff marks in the crankcase, and at the oil seals showed the vibration amplitude was underestimated. Metal was removed from the locations where scuff marks were found, and the tests repeated until no more scuff marks were evident.

Vibration from the bearings could also be transmitted to the chain saw panels via the connecting rod, piston, and cylinder. To eliminate this additional flanking path to the radiating surfaces of the saw, the cylinder was also vibration isolated ((fig. 3.11)). The resonant frequency was set approximately to that of the crankshaft vibration isolators. All surfaces of the cylinder that contacted the crankcase were machined to allow 0.8 mm of silicone rubber to be placed between them. The cylinder head bolt holes were enlarged to allow the insertion of stacked o-rings to surround the bolts. To isolate the cylinder bolt heads, due to limited space, silicone rubber was glued between two steel washers, and the bolt head rested on the top washer.

The weakest point in the system was the silicone rubber and steel washers. Although silicone rubber is notorious for being difficult to bond, it was found that cyanoacrylate glue could form a bond stronger than the rubber itself. This required scrupulous cleaning, and preparation of the steel and silicone surfaces, by stripping them with an epoxy peeling. Yet, in operation, the high heat of the chain saw caused the bond to fail. This made constant inspection of the washers necessary during tests, and frequent replacement if more than one test was being performed.

Vibration isolation of the crankshaft reduced vibration in some parts of the saw, but, parallel to the cylinder axis, there was a small increase in vibration, and the received sound pressure at the operator's ear increased slightly. Additional vibration isolation of the cylinder did not improve results. Whether this was due to an inadequacy in the vibration isolation, or an increase in vibration due to the change in mounting (failure to provide sufficient preload on the main bearings, misalignment or free play in parts) is uncertain.

The results showed slightly increased noise levels at 800 Hz and above 4 kHz. Piston slap noise and fan noise, are significant at these frequencies. The compliance of the crankshaft mounting allows movement of the crankshaft, which could affect either source. Although

vibration isolation of the bearings did not reduce the noise, the saw could still operate (without load).

Vibration isolation at other points in the saw could prove easier than on the main bearings. For example, some benefit may be obtained if the cylinder and crankcase were vibration isolated from the external panels. This could prevent the vibration source from exciting the external panels. This method could be less bulky than trying to stop airborne sound radiation with an acoustical enclosure as suggested previously. However, the cylinder vibration would likely increase, requiring additional measures to reduce noise radiated by the cylinder (i.e., an acoustical enclosure).

D.6. Bar chain and cutting

A significant source of noise at high frequencies is due to chain impacts. Cutting noise extends over a broad frequency range and also can be important, especially when cutting frozen wood. Tests with the pulpwood saw in an acoustic enclosure cutting thin discs from dry wood produced levels of 91 dB (A-weighted) at the operator's ear.

The use of a true silent chain (fig. 2.8) on the chain saw may reduce noise. Some noise reduction could be obtained at high frequencies by reducing the noise from chain impacts (for most wood types). These impacts are likely related to polygonal action that produces impacts directly with the sprocket, and indirectly affects impacts between chain and bar through torque and speed fluctuations. Polygonal action could be reduced through the use of a silent chain, although this may prove difficult to implement on a saw chain. Reduction of polygonal action may allow the noise levels of the chain saw to be reduced a few dB below 91 dB (A-weighted) at the operator's ear when cutting dry wood.

Other measures could also reduce noise due to chain impacts. For example, the rim drive sprocket used on the pulpwood likely reduces noise due to the large contact area between chain and sprocket. This increases the stiffness and friction between chain and sprocket. Increased stiffness would spread the impact energy across a broader range of frequencies, and thus reduce levels in the audible range. An increased contact area increases friction between chain and sprocket, which could reduce the impact velocity of the chain with the sprocket.

D.7. Piston

Piston slap is not as significant a problem as other sources. However, the noise can be annoying, and some reduction of piston slap noise might be beneficial if other sources

were also quieted. As piston slap was a common problem in automotive engines, there are many ideas for quieting the noise, some of which were mentioned previously (section 2.6.6 on limitations of piston slap analysis).

One way to control the noise is to ensure that clearances do not get too large. The chain saw piston is cylindrical, thus a barrel shaped, or thermal expansion controlled piston could minimize running clearance. However, note that slight changes in piston clearance produced measurements opposite to what was expected. Making the piston cylinder clearance .006 mm tighter caused a 2 to 6 dB noise increase, and making the piston 0.006 mm looser produced no significant change, (an apparent noise reduction of 1 dB). This behavior has been found in other engines with tight fitting pistons and has been called stick slip noise⁷⁶.

In other tests on the pulpwood saw, an attempt was made to reduce metal to metal contact between piston and cylinder by using plastic inserts, (fig. 3.11). Plastic inserts have reduced noise in other engines⁷⁶. This can soften the piston impact force, add damping, reduce friction, and reduce clearances.

The metal piston rings on the pulpwood saw were directly replaced by using similar rings made of Viton or Teflon plastic. Teflon plastic was difficult to use as it wore quickly during saw operation, and would expand due to the heat of the engine. At the conclusion of a test, the Teflon ring would be worn to the point were it did not extend beyond the edge of the piston. In a further test, pads made of Teflon were inserted at the bottom of the skirt of one piston. The pads were inserted in a shallow groove made using a metal lathe. They were held in place using small Teflon dowels. This test was aborted because thermal expansion of the Teflon bent the piston skirt inward, deforming the piston. Tests using Vespel plastic were more successful, although likely due to the low noise produced by the piston, no noise reduction was noted.

D.8. Combustion and intake system noise

As shown during testing, combustion noise need not affect the chain saw noise. At low frequencies, the stiffness of the structure, and the low radiation efficiency due to the small size helps reduce the effect of combustion noise. At higher frequencies the spectra rolls off about 15 dB per octave and so has little effect on noise. Thus combustion noise should not be a significant problem, unless the pulpwood saw speed was increased significantly (i.e., a factor of two).

As with combustion noise, the intake system and crankcase noise need not be a problem. However, the intake systems on the pulpwood saw, and large timber saw were not designed to reduce noise. Both saws used thin, lightweight plastic or rubber components that provided little transmission loss. Replacement of these components with stiffer materials would be sufficient to reduce noise from this source.

D.9. Unbalanced shaking forces

As the unbalanced shaking forces occur mainly at low frequencies, they do not significantly affect the A-weighted noise of the chain saw. If other sources on the chain saw can be reduced, the low frequency noise may be desirable as it could make the saw sound more powerful, and help mask any rattles.

Appendix E. Definition and explanation of functions, and equations

For reference a definition and explanation of the functions referred to in this thesis is provided below.

Fourier transform¹²¹:

$$\mathcal{F}\{f(t)\} = F(f) = \int_{-\infty}^{\infty} f(t) e^{-i2\pi ft} dt \quad 85$$

where $\mathcal{F}\{\}$ is the Fourier transform operator, $f(t)$ is a time domain function (with t as time), $F(f)$ is the Fourier transform of f , and f represents frequency in Hz.

Inverse Fourier transform¹²¹:

$$\mathcal{F}^{-1}\{F(f)\} = f(t) = \int_{-\infty}^{\infty} F(f) e^{i2\pi ft} df \quad 86$$

Parseval's identity simply states that the energy in both the time and frequency domain must be equal¹²¹, i.e.:

$$\int_{-\infty}^{\infty} |f(t)|^2 dt = \int_{-\infty}^{\infty} |F(f)|^2 df \quad 87$$

Time domain data is assumed to be real valued. It is generally related to a single event occurring in an engine cycle, thus power is obtained by multiplying the energy per event by the cyclic frequency of the engine.

Unless specified otherwise, equations in the frequency domain are obtained using a Fourier integral transform, which is multiplied by two at positive frequencies. This allows negative frequencies to be ignored in the analysis, and provides a direct comparison with output from fast Fourier transform analyzers. The spectrum is not required at zero frequency, so this value is not explicitly accounted for. For example, an equation describing the energy (E) produced by the pressure spectrum (p) is:

$$E = a \int_{-\infty}^{\infty} |p|^2 df = 2a \int_0^{\infty} |p|^2 df \quad 88$$

where f is frequency, and a is dependent on the medium properties.

Equation 88 evaluated over a finite bandwidth Δf , centered at frequency f_o is:

$$E(f_o) = 2a \int_{f_o - \frac{\Delta f}{2}}^{f_o + \frac{\Delta f}{2}} |p|^2 df = 2a |p(f_o)|^2 \Delta f \quad 89$$

The right hand side of equation 89 is representative of the output of a fast Fourier transform analyzer. For a sinusoidal signal, the result of equation 89 is sensibly independent of the analyzer bandwidth. However, if p is a broadband spectrum, the energy $E(f_o)$ varies in proportion to the bandwidth Δf .

A real analyzer must measure over a finite time period, and equation 89 could be expressed in terms of power for a continuous signal. For example, a 1/3 octave band analyzer generally measures power. This requires consideration of the measurement duration (t_m), and equation 89 for power (P) becomes:

$$P(f_o) = \frac{E(f_o)}{t_m} = \frac{2a}{t_m} \int_{f_o - \frac{\Delta f}{2}}^{f_o + \frac{\Delta f}{2}} |p|^2 df = \frac{2a}{t_m} |p(f_o)|^2 f_o \left(\frac{\Delta f}{f_o} \right) \quad 90$$

where an event that occurs once per engine cycle has t_m equal to $1/n$ (n the cyclic frequency of the engine).

When the slope of a constant percentage bandwidth spectrum is described (i.e., 1/3 octave), the decibel (dB) change per octave, 1/3 octave, or decade will be used exclusively (i.e., 3 dB per octave). Quantities in equations will be described in terms of the Fourier transform representation, as a function of frequency raised to some power (i.e., f^p). For example, the bracketed term $(\Delta f/f_o)$ is nominally 0.231 for any 1/3 octave band filter, thus equation 90 shows white noise would be described as having a Fourier transform spectrum varying proportional to f^0 , with the associated 1/3 octave band spectrum measurements increasing at 3 dB per octave.

Fourier transform analysis was performed using the Brüel & Kjær 2032 FFT analyzer (Appendix G.7). Following is a listing of definitions for the functions used in this thesis.

Forward discrete Fourier Transform as implemented by the Brüel & Kjær 2032 ¹³⁰:

$$F(k) = \frac{1}{N} \sum_{n=0}^{N-1} f(n) e^{-i2\pi kn/N} \quad 91$$

where $f(n)$ is the n th sample of time history data in a record of length N points, and k is the frequency index (k evaluated from 0 to $N-1$). In a continuous representation time is $n\Delta t$ and frequency is $k\Delta f$, where Δt is the sampling interval, and Δf is $1/2N\Delta t$.

Inverse discrete Fourier transform as implemented by the Brüel & Kjær 2032 ¹³⁰:

$$f(n) = \sum_{k=0}^{N-1} F(k) e^{i2\pi kn/N} \quad 92$$

Spectrum as implemented by the Brüel & Kjær 2032 ¹³⁰:

$$S_A(k) = \mathcal{F}\{w(n) f(n)\} \quad 93$$

S_A is the time windowed two sided spectrum of $f(n)$, $w(n)$ is the weighting function, n and k are evaluated from 0 to $N-1$

The one sided spectrum G_A (spectrum of the analytic signal) is defined as¹³⁰:

$$G_A(k) = \begin{cases} S_A(k) & \text{for } k = 0 \\ 2S_A(k) & \text{for } 1 \leq k \leq N/2 - 1 \\ 0 & \text{for } N/2 \leq k \leq N-1 \end{cases} \quad 94$$

Autospectrum G_{AA} as implemented by the Brüel & Kjær 2032 ¹³⁰:

$$\overline{G_{AA}(k)} = \overline{|G_A(k)|^2} \quad 95$$

where the over bar indicates averaged spectra.

Cross spectrum G_{AB} as implemented by the Brüel & Kjær 2032 ¹³⁰:

$$\overline{G_{AB}(k)} = \overline{G_A^*(k) G_B(k)} \quad 96$$

where G_A^* is the complex conjugate of G_A , and G_B is the one sided spectrum for another measurement channel.

Coherence γ_{AB}^2 as implemented by the Brüel & Kjær 2032¹³⁰:

$$\gamma_{AB}^2(k) = \frac{|\overline{G_{AB}(k)}|^2}{G_{AA}(k)G_{BB}(k)} \quad 97$$

Cepstrum C_{2a} as implemented by the Brüel & Kjær 2032¹³⁰:

$$C_{2a}(n) = \mathcal{F}^{-1} \left[10 \log_{10} (\overline{G_{AA}(k)}) \right] \quad 98$$

Wigner distributions were obtained using a 2048 point time history from the Brüel & Kjær 2032 input into an IBM compatible computer and analyzed using a modified BASIC program¹³¹. Modifications to the program were limited its graphical output. The discrete implementation of Wigner distribution (pseudo Wigner distribution or *PWD*) used in the program is¹³²:

$$PWD \left(n, k \frac{\pi}{N} \right) = 2 \sum_{m=-N/2+1}^{N/2-1} e^{-imk2\pi/N} w(m) w^*(-m) a(n+m) a^*(n-m) \quad 99$$

where n is the time index (evaluated from $-N/2$ to $N/2$), m is an offset from the time index n , and k is the frequency index (evaluated from $-N/2$ to $N/2$), and a is the analytic time signal.

The use of an analytic time signal allows the maximum frequency evaluated to equal that determined by Fourier transform (Nyquist frequency)¹³². The analytic signal $a(n)$ is given by¹³⁰:

$$a(n) = f(n) + i\mathcal{H}\{f(n)\} \quad 100$$

where $\mathcal{H}\{f(n)\}$ is the Hilbert transform of the sampled time signal $f(n)$.

The most efficient method of obtaining the analytic signal is by inverse Fourier transform of the one sided spectrum of the input signal¹³⁰.

Appendix F. Reference values for decibel levels

The decibel levels used in this thesis are based on the SI system of measurement¹³³ and are defined below.

Measured quantity	Defining equation for decibel level	Reference value
Power	$PL = 10 \log(P/P_{ref})$	$P_{ref} = 10^{-12} \text{ W}$
Acoustic Intensity	$IL = 10 \log(I/I_{ref})$	$I_{ref} = 10^{-12} \text{ W m}^{-2}$
Sound Pressure	$SPL = 10 \log(p^2/p_{ref}^2)$	$p_{ref} = 2 \times 10^{-5} \text{ Pa}$
Acceleration	$LA = 10 \log(acc^2/acc_{ref}^2)$	$acc_{ref} = 10^{-6} \text{ m s}^{-2} *$

*from ISO 1683¹³⁴

Note that SPL , PL (through a 1 m^2 surface), and IL are almost identical in a free field, far from the source, for a measurement surface oriented normal to the direction of sound propagation (i.e. plane wave propagation).

Appendix G. Equipment Specifications

G.1. Chain Saws

G.1.1. Large-timber saw

manufacturer	Pioneer
model	P-51
cycle	two-stroke, crankcase scavenged
fuel	16 parts gasoline to 1 part oil
weight	9 kg
operating speed	9600 rpm
displacement	82 cc
cylinder bore	52 mm
stroke	38 mm
bar length	40 cm to 90 cm, (40 cm used)
chain	9.5 mm pitch
chain drive	7 or 8 tooth sprocket (7 tooth rim drive sprocket used)
clutch	direct drive, centrifugal, spring loaded clutch shoes
ignition	magneto, sealed electronic ignition (no adjustments possible)
cooling	air cooled
fan	26 blade radial forward curved, direct drive
carburetor	Walbro (diaphragm carburetor driven by crankcase pressure)
crankcase inlet	reed valves
transfer ports	piston ported
exhaust port	piston ported
vibration isolation	rubber standoffs disconnect handles and inlet plenum from engine

G.1.2. Pulpwood Saw

G.1.2.1. General

manufacturer	Homelite
model	410
cycle	two-stroke, crankcase scavenged
fuel	16 parts gasoline to 1 part oil
weight	7.5 kg
operating speed	7500 rpm
displacement	67 cc
cylinder bore	4.9 cm
stroke	35 mm
bar length	36 cm to 50 cm, (40 cm used)
chain	9.5 mm pitch
chain drive	7 or 8 tooth sprocket (7 tooth rim drive sprocket used)
clutch	direct drive, centrifugal, spring loaded clutch shoes
ignition	magneto, sealed electronic ignition (no adjustments possible)
cooling	air cooled
fan	24 blade radial forward curved, direct drive
carburetor	Walbro (diaphragm carburetor driven by crankcase pressure)
crankcase inlet	reed valves
transfer ports	piston ported
exhaust port	piston ported
vibration isolation	rubber standoffs disconnect handles, inlet plenum, oil and fuel tanks from engine

G.1.2.2. Reference condition

- no load
- 7500 rpm
- speed controlled by throttle within ± 150 rpm
- test stand
- Exhaust and intake piped away and silenced
- clutch cover, bar and chain removed
- dummy clutch, dummy oil pump, heavy grease to damp starter spring
- Cerro Bend metal to increase thickness and mass of intake plenum walls
- 6 degree of freedom brass coupler connecting crankcase (in powerhead) to carburetor (in vibration isolated handles)

G.1.2.3. Component values for equations

A	0.17 m ²	powerhead surface area
A_p	18 cm ²	piston area
d	0.05 mm	piston cylinder clearance (in diameter)
	0.03 in	gudgeon pin bearing clearance (in diameter)
D_1	37 mm	fan rotor inlet diameter
D_2	56 mm	outer diameter of fan rotor
D_R	49 mm	cylinder bore diameter
h	5 mm	material thickness
l	0.4	ratio of L_R/L_C
L_h	31 mm	fan blade height
L_r	19 mm	large fan blade chord (12 blades)
	11-16 mm	tapered fan blade chord (12 blades)
L_C	67 mm	connecting rod length
L_n	9.5 mm	chain pitch
L_p	40 mm	length from center of gravity to piston end of connecting rod
L_R	27 mm	length from center of gravity to crankshaft end of connecting rod
m	5.5 kg	powerhead mass (excluding handles)
m_C	0.06 kg	connecting rod mass
m_p	0.11 kg	piston mass
n	125 rps	engine revolutions per second
r	29 mm	connecting rod radius of gyration
R_n	(18+3) mm	chain sprocket pitch radius
R_C	18 mm	crank radius
β_1	82°	fan blade inlet angle
β_2	121°	fan blade exit angle
γ	0.27	ratio of crankshaft radius to connecting rod length (R_C/L_C)
η	0.05	structural damping
v	-1.0	dimensionless constant $(l-2l^2-r^2/L_C)/l\gamma$
u	0.55	ratio of m_C/m_p (connecting rod mass)/(piston mass)

G.2. Adamson M200 midrange compression driver

In some tests a point source was needed, (reciprocity measurement), or an acoustical driver needed to be connected to an enclosed area. An Adamson M200 horn driver, was

typically used. The throat of the horn driver was too large and therefor a Bakelite insert was made so that the throat narrowed smoothly to a 3 cm diameter opening. Then a flanged tube was attached to the 3 cm opening so that attachments could be screwed on. To reduce acoustical output from the back of the speaker, a rubber enclosure was constructed. The enclosure was constructed from 2 layers of 3.2 mm thick neoprene rubber, and filled with loose polyester fibers. The exterior was covered with 13 mm thickness of felt to help reduce diffraction from the edges of the flange and the speaker. The driver was suspended inside the enclosure by a stainless steel wire, and the flanged tube at the driver throat. The wire that supported the driver was also used to suspend the enclosure from a steel tripod.

Power handling	300 watts RMS long term 450 watts program
Sensitivity (1 Watt 1 m)	111 dB
Maximum SPL	139 dB
Efficiency	20%
optimum frequency range	200 Hz to 2 kHz

G.3. Pressure measurement

G.3.1. Measurement microphones

Brüel & Kjær type 4134

Nominal diameter	1/2"
Frequency response characteristic	Random incidence & pressure
Open circuit frequency response (± 2 dB)	4 Hz to 20 kHz
Cartridge thermal noise (dB(A))	18
Long term stability @ 20°C	1000 yr./dB
Influence of static pressure	-0.0007 dB/mbar
Influence of 1 m/s ² Axial vibration	67 dB re 20 μ Pa
Influence of relative humidity	<0.1 dB

Brüel & Kjær type 4165

Nominal diameter	1/2"
Frequency response characteristic	Free field, 0° incidence
Free field frequency response (± 2 dB)	2.6 Hz to 20 kHz
Cartridge thermal noise (dB(A))	14.5
Upper limit dynamic range for 3% distortion	146 dB SPL

Temperature coefficient	-0.007 dB/°C
Influence of static pressure	-0.001 dB/mbar
Influence of 1 m/s ² Axial vibration	60 dB re 20 µPa
Influence of relative humidity	<0.004 dB/% RH
Phase response difference between paired microphones	
Amplitude response difference between paired microphones	

Brüel & Kjær type 4135

Nominal diameter	1/4"
Frequency response characteristic	Free field, & random incidence
Free field frequency response (± 2 dB)	4 Hz to 100 kHz
Cartridge thermal noise (dB(A))	29.5
Upper limit dynamic range for 3% distortion	164 dB SPL
Temperature coefficient	-0.01 dB/°C
Influence of static pressure	-0.0007 dB/mbar
Influence of 1 m/s ² Axial vibration	59 dB re 20 µPa
Influence of relative humidity	<0.1 dB
Phase response difference between paired microphones	
Amplitude response difference between paired microphones	

Brüel & Kjær type 4181

Nominal diameter	1/2"
Frequency response characteristic	Random incidence & pressure
Free field frequency response (± 2 dB)	0.3 Hz to 16 kHz
Cartridge thermal noise (dB(A))	20
Upper limit dynamic range for 3% distortion	160 dB SPL
Temperature coefficient	-0.002 dB/°C
Influence of static pressure	-0.0007 dB/mbar
Influence of 1 m/s ² Axial vibration	68 dB re 20 µPa
Influence of relative humidity	<0.1 dB
Phase response difference between paired microphones	<5°/5000
Amplitude response difference between paired microphones	20 Hz-5 kHz <0.4 dB

G.3.2. Cylinder pressure

Kistler type 6121A2

measuring range	0 to 3500 psi
maximum pressure	5000 psi
sensitivity	-0.95 pC/psi
resonant frequency	60 kHz
frequency response $\pm 1\%$	6 kHz
rise time (10 to 90%)	6 μ s
linearity	$<\pm 1\%$ FS
hysteresis	$<0.5\%$ FS
acceleration sensitivity	0.05 psi/g
shock and vibration	<2000 g
temperature sensitivity shift	$<\pm 0.01\%$ /°C
operating temperature range	-80 to 350 °C
maximum flash temperature	2500°C
weight	10 g

G.3.3. Miniature microphones

Panasonic omnidirectional electret condenser microphones type WM063T were used for measurements inside cylinder and crankcase, when chain saw was not running. These are inexpensive (Five dollar) microphones with small size. Electronic signal conditioning used a custom made op amp based differential¹³⁵ amplifier. Tests have shown the microphones to have a nearly omnidirectional frequency response, ± 2 dB from 20 Hz to 20 kHz¹³⁶. The microphones are stable over a wider range of temperature and humidity than found in the laboratory¹³⁷.

The microphones were mounted inside the cylinder and crankcase by attaching them to the ends of dummy crankcase and cylinder pressure transducers, which were then mounted as usual inside the cylinder and crankcase. To vibration isolate the microphones from the chain saw exterior, the wires connected to the electrical terminals were wound into spring like coils. The mechanical resonance of the wire spring and microphone system was about 10 Hz.

G.3.4. Dynamometer load cell

SenSym type LX1802GZ

measuring range	0 to 15 psig
maximum pressure	45 psig
resonant frequency	>50 kHz
linearity, hysteresis, repeatability	0.7% FS
temperature shift 0-80°C	$\pm 2.7\%$ FS
operating temperature range	-40 to 105 °C

G.4. Acceleration measurement

G.4.1. Accelerometers

Brüel & Kjær type 4393

weight	2.4 g
charge sensitivity	$0.316 \pm 2\%$ pC/ms ²
mounted resonance	55 kHz
frequency range $\pm 5\%$	0.2-12000 Hz
frequency range $\pm 10\%$	0.1-16500 Hz
capacitance	650 pF
transverse sensitivity	<4%
transverse resonance	18 kHz
acoustic sensitivity	0.04 ms^{-2} @ 154 dB SPL

Brüel & Kjær type 4344

weight	2.7 g
charge sensitivity	$0.316 \pm 2\%$ pC/ms ²
mounted resonance	120 kHz
frequency range $\pm 5\%$	near DC-40 kHz
capacitance	1000 pF
transverse sensitivity	<4%
acoustic sensitivity	0.1 ms^{-2} @ 154 dB SPL

G.4.2. Charge amplifiers

Brüel & Kjær type 2635

Frequency Response	0.2 Hz to 100 kHz
10%	
Inherent noise	5×10^{-3} pC @ max sens, 1 nF transducer

Kistler type 504e

Range accuracy	$\pm 5.0\%$
Sensitivity linearity	$\pm 0.2\%$ FS
Amplitude linearity	0.1%
error	
Frequency Response $\pm 5\%$	near DC to 100 kHz
Noise	$0.0005 \{ [1 + 0.3(\text{Transducer Sens Dial})](\text{Range}) + 0.3 \}$ pCb threshold + 0.007 pCb/nF source load
Capacitive source loading	$10 \text{ nF} \times \text{Range}$ for 1% attenuation

G.5. Miscellaneous

G.5.1. Filter

Krohn-Hite model 3343R

frequency range	0.001 Hz to 99.9 kHz
low pass filters	-maximally flat 8 pole Butterworth -8 pole damped response low Q
high pass filters	maximally flat 8 pole Butterworth
filter slope	48 dB per octave
maximum attenuation	>80 dB
output voltage	±7V peak
output noise	0.5 mV rms 100 kHz bandwidth

G.6. 1/3 octave analyzer

Brüel & Kjær type 3360

Sampling Frequency	66667 Hz
1/1 octave filters	Chebyshev, ANSI S1.11 Class II
1/3 octave filters	Chebyshev, ANSI S1.11 Class III
1/12 octave filters	6 pole
A-filter	IEC 651 Type 1
Frequency Response	±0.2 dB 0 to 30 dB below FSD ±2.0 dB 55 to 60 dB below FSD
Dynamic Range	63 dB
Averaging	exponential or linear
Frequency Accuracy	0.01%
Functions	sound pressure sound intensity

G.7. FFT analyzer

Brüel & Kjær type 2032

Sampling Frequency	65536 Hz
Antialiasing filters	
cut off frequency	25.6 kHz
pass band ripple	±0.3 dB
minimum aliasing	75 dB
attenuation	
Frequency Response	±0.4 dB
Amplitude Linearity	±0.1 dB
Dynamic Range	>75 dB
Frequency Accuracy	0.01%
Frequency Resolution	801 lines
Functions	Autospectrum Cross Spectra Frequency Response (Transfer function between two channels) Coherence Coherent Power Non coherent Power Autocorrelation Cross Correlation Impulse response Cepstrum

Liftered Spectrum
 Time history
 Signal Averaging (time history)
 Hilbert transform
 Probability Density
 Probability Distribution

Appendix H. Publications

Invited papers

S.E. KEITH, A.J. BRAMMER, "Prospects for low-noise, high-performance chain saws: An achievable goal?," 121 st meeting J. Acoust. Soc. Am., Baltimore Maryland, Apr. 29 to May 3, 1991

General

S.E. KEITH, A.J. BRAMMER, "Prospects for low-noise, high-performance chain saws: An achievable goal?," J. Acoust. Soc. Am., 89:4, pg. 1969, 1991

A.J. BRAMMER, S.E. KEITH, "Prospects for low-noise, high-performance chain saws", Noise-Con 87, pg. 171-176. 1987.

S.E. KEITH, A.J. BRAMMER, "Current research on Chainsaw noise", Pulp and Paper Canada, 87:4 pg. T153-T156. 1986

S.E. KEITH, A.J. BRAMMER, "Sources of noise in professional chainsaws", Inter-noise 84, Vol. 1, pg. 143-146. 1984.

S.E. KEITH, A.J. BRAMMER, "Current research on chain saw noise," proceedings Technical Section, Canadian Pulp and Paper Association, 1984 Noise Control Seminar, pg. 53-56, 1984.

Oral

A.J. BRAMMER, S.E. KEITH, "Prospects for low-noise, high-performance chain saws", Noise-Con 87, State College, Pennsylvania, June 8-10, 1987.

S.E. KEITH, A.J. BRAMMER, "Sources of noise in professional chainsaws", Inter noise 84, Honolulu, Hawaii, Dec. 3-5, 1984.

S.E. KEITH, A.J. BRAMMER, "Current research on Chainsaw noise", 1984 Noise Control Seminar, Technical Section, Canadian Pulp and Paper Assoc., Vancouver, British Columbia, Sept. 10-12, 1984

Publications on Related Topics

Invited papers

S.E. KEITH, A.J. BRAMMER, "Rock Drill handle vibration: Measurement and hazard assessment," Canadian Acoustics, Edmonton, Alberta, Oct. 9-10, 1991.

General

A.J. BRAMMER, S.E. KEITH, "Hazard to the hand posed by vibration at different frequencies: Assessment of two equi-noxious contours for vibration-induced white finger", in preparation

S.E. KEITH, A.J. BRAMMER, "Rock-drill handle vibration: Influence of operating parameters and high frequencies on the specification of hazardous exposures," submitted to the Journal of Sound and Vibration, 1991

S.E. KEITH, A.J. BRAMMER, "Rock Drill handle vibration: Measurement and hazard assessment," Canadian Acoustics, 19:4, pg. 49-50, 1991.

S.E. KEITH, A.J. BRAMMER, "Errors in the measurement of power tool handle vibration from excess accelerometer weight", J. Acoust. Soc. Am. Vol. 84, S1 pg. S4, Fall 1988.

W.G. RICHARZ, S.E. KEITH, "Acoustic power of jets", J. Acoust. Soc. Am. Vol. 69, S1 pg. S117, 1981.

S.E. KEITH, "Direct measurement of sound from large scale structures in jet flows", University of Toronto MSc Thesis. 1983.

S.E. KEITH, "Acoustic power of hot and cold jets", University of Toronto BSc Thesis. 1981.

S.E. KEITH, H.S.B. SCHOLAERT, "A study of the performance of an Olson type active noise controller and the possibility of reduction of cabin noise", University of Toronto Institute for Aerospace Studies TN 228. 1981.

Oral

S.E. KEITH, A.J. BRAMMER, "Errors in the measurement of power tool handle vibration from excess accelerometer weight", 116 Meeting J. Acoust. Soc. Am. Honolulu, Hawaii, Nov. 14-18, 1988.

W.G. RICHARZ, S.E. KEITH, "Direct measurement of sound from large scale structures in jet flows", American Institute of Aeronautics and Astronautics, 83-0662, Atlanta, Georgia, 1983.

W.G. RICHARZ, S.E. KEITH, "Acoustic power of jets", 101 st meeting J. Acoust. Soc. Am., Ottawa, Ontario, 1981.

Contract

S.E. KEITH, A.J. BRAMMER, "Handle vibration of the HD-30 and S-250 jack leg rock drills", NRC report for Boart Canada, March 1990.

S.E. KEITH, A.J. BRAMMER A.P. LORIMER, acoustical performance evaluation of muffler assembly, addendum to proposal by Stebro Canada, for Military operational support trucks branch, Department of Supply and Services. January 1989.

S.E. KEITH, A.J. BRAMMER, "Change in chain saw handle vibration with accelerometer mass", for Canadian Forestry Service Contract #OIK02-7-0256, March 1988.

Figure 1.1: Exploded diagram showing significant components of a typical chain saw.

Figure 1.2: Cross section of typical chain saw.

Figure 1.3: A-weighted 1/3 octave spectra for three equal amplitude square waves with different fundamental frequencies (labeled).

Figure 2.1: Typical internal combustion engine cylinder pressure spectrum (Reprinted by permission of the author¹¹⁴).

Figure 2.2: Fan blade vector diagram.

Figure 2.3: Piston and connecting rod geometry and forces.

Figure 2.4: Relative gas and inertial force contribution to piston side thrust, (for pulpwood saw operating at no load and best cutting speed): — average gas force values; - - - gas forces due to cylinder compression only; — cycle to cycle variation in gas forces; — inertial forces.

Figure 2.5: Non dimensional piston side thrust, (for pulpwood saw operating at no load and best cutting speed). - - - zero clearance estimate; — cylinder side force with clearance; — possible variation in side force due to non repetitive gas force.

Figure 2.6: Graph showing variation of piston slap vibratory power with speed (dB re $10^{-12}W$). — average values; - - - high or low speed asymptotes; possible variation in power level due to non repetitive gas force.

Figure 2.7: Non dimensional connecting rod bearing forces. Horizontal axis indicates force along cylinder axis. — little end of connecting rod; — big end of connecting rod. Dashed lines indicate sections where contact loss may occur. Arrows indicate progression through cycle. Forces at top and bottom dead center crank angle are noted.

Saw Chain

Silent Chain

Figure 2.8: Comparison of saw chain (top) and silent chain (bottom).

Figure 2.9: Illustration of polygonal action when chain meshes with sprocket. Sprocket is shown shaded, initial position of chain is shown by solid line, final position of chain by dashed line, chain pitch (L_p) and pitch radius (R_p) are also shown.

Figure 3.1: Overhead view of test site.

Figure 3.2: Photo of acoustic enclosure, (rear handle rubber boot removed). Chain saw is vibration isolated from enclosure using blue isolators located underneath saw. Note brass fixtures are for intake and exhaust silencing.

Figure 3.3: Level reduction provided by placing a point source in the chain saw acoustical enclosure. ○ point source level at operator's ear position; ◻ point source level at 5 m; ◼ levels at operator's ear with point source in acoustical enclosure; ● 5 m levels for enclosed point source; [shaded area] range of background noise levels.

Figure 3.4: Photo of chain saw mounted on test stand, showing dynamometer and belt drive

Figure 3.5: Block diagram of typical measurement setup.

Figure 3.6: Comparison of human arm and hand model support (compliant rubber elements shown in black).

Figure 3.7: Difference in component acceleration levels on the engine casing of the large timber saw: a - when first hand held and then suspended from the test stand by model hands with the belt drive and flexible exhaust and intake pipes attached (engine idling); b - when mounted on the test stand first with belt drive attached and then with belt drive and flexible exhaust and intake pipes attached (engine at best cutting speed). $\odot$ front to rear; $\updownarrow$ up and down; $\leftrightarrow$ side to side.

Figure 3.8: Overhead view of laboratory. Shaded areas indicate acoustically absorbing material.

Figure 3.9: Relative change in 1/3 octave band level plotted for different microphone positions when saw replaced by a point source; the 0 dB line was recorded 38 cm from the point source, — predicted decrease for free field propagation; 75 cm (EAR) represents the reference microphone position used with the saw.

Figure 3.10: Differences in 1/3 octave band sound pressure levels for measurements in the field versus the laboratory. ▲ range of increase in field measurement levels; ▼ increase in level due to human support. ◆ level reduction from modifications to saw used in laboratory.

Figure 3.11: Composite sketch combining most modifications made to stock saw. Not shown; cutting bar and chain removed, intake system walls thickened, intake ducting, exhaust ducting, crankcase pressure transducer, and gas feed from external tank.

Figure 3.12: Field measurements showing ducting of intake and exhaust

Figure 3.13: Contour plot showing change in acoustic intensity with change in chain saw load
Contour levels are marked, higher numbers indicate more noise in the full load condition

Figure 3.14: Insertion loss of intake system relative to open intake. + stock intake; ✕ modified silenced intake.

Figure 3.15: Photo showing modifications to dummy fan, on right. To show detail, the 2 mm thick, flat aluminum sheet normally used to block airflow to the dummy fan has been removed.

Figure 3.16: External motor drive for fan.

Figure 3.17: Sound level reduction due to fan modifications compared to normal fan (0 dB line): $\blacksquare$ inlet blocked with Glad kitchen wrap; $*$ dummy fan (blades removed). Fan was driven by electric motor at 7500 rpm.

Figure 3.18: Vibration isolated cylinder pressure transducer, shown actual size.

Figure 3.19: Effectiveness of pressure transducer vibration isolation. Measurement obtained on chain saw operated without load.

Figure 4.1: A typical cutting test.

Figure 4.2: Location of operator's car microphone during field testing.

Figure 4.3: Laboratory setup for crankcase structural attenuation. Speaker on left is connected to chain saw with rubber coated brass tube. Note operator's ear microphone located near top center.

Figure 4.4: Laboratory setup for reciprocity measurements. Point source is approximated by a small hole in the white, felt covered, cylinder at top center of picture. The monitoring microphone, with windscreen, is shown (black sphere) just below point source.

Figure 5.1: One-third octave-band spectra at the operator's ear (a and b) and in the far field (c and d) for: X open exhaust; + open intake; ◆ silenced exhaust and intake (residual noise); * stock. Data for the pulpwood saw are shown in a and c and for the large-timber saw in b and d.

Figure 5.2: a) Far field, and b) operator's ear sound pressure levels with exhaust and intake silenced and saw in acoustic enclosure; ◼, ◻ cutting two types of wood (mechanical noise suppressed by acoustical enclosure); ⊠ chain driven at best cutting speed (but not cutting, and no engine load); — without the saw chain (no engine load). For comparison with previous figures; ◆ shows no enclosure cutting wood (i.e., residual mechanical noise);

OF / DE

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

Figure 5.3: One-third octave-band reference sound pressure levels obtained in laboratory at operator's ear position: — single tests, $\diamond$ energy average of single tests, ± 2 standard deviations of linear dB average, ± 1 standard deviation of linear dB average (obscured by data).

Figure 5.4: Comparison laboratory versus field measurements of sound pressure levels at operator's ear position with saw in reference condition, and chain running at best cutting speed. $\diamond$ outdoor measurements cutting wood; ∇ outdoor measurements not cutting. For comparison: $\blacklozenge$ original field cutting measurements (see fig. 5.2), shaded area indicates normal level variations in reference condition.

Figure 5.5: Average one-third octave-band acoustic intensity levels scanned 5 cm from the selected engine covers or panels (dB re 10^{-12} W m $^{-2}$): ▽ Clutch cover; ⊗ Blower cover; ▲ Isolated Handle; ⊠ Isolated Bottom.

Figure 5.6: Graph showing acoustical response of chain saw to hammer hits. — calculated values (vertical position of curve arbitrarily selected); — (and —) measured (and range of) sound pressure levels due to impacts on saw exterior.

Figure 5.7: One-third octave-band sound pressure levels at operator's ear position: * mechanical noise with dummy fan (and external cooling); ◇ mechanical noise with normal fan, (obtained under similar conditions); shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.8: One-third octave-band sound pressure levels at operator's ear position (at 7500 rpm): x motored fan noise; shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.9: One-third octave-band mechanical noise sound pressure levels at operator's ear position (- - -). From top curve 9600, 7500, 6000, 4800, 3800 rpm. — are arbitrarily placed spectral estimates. For comparison shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.10: One-third octave-band cylinder pressure levels: ● full load; ○ no load.

Figure 5.11: One-third octave-band sound pressure levels at operator's car position:
 ◆ mechanical noise with full load; ◇ mechanical noise, (without load, obtained same day);
 shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.12: One-third octave-band attenuation of cylinder pressure predicted at operator's car position.

Figure 5.13: One-third octave-band calculated sound pressure levels at operator's ear position using cylinder pressure and reciprocity: **x** calculated noise from full load combustion; shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.14: One-third octave-band crankcase pressure levels: **●** full load; **+** no load.

Figure 5.15: One-third octave-band calculated sound pressure level contribution to mechanical noise from crankcase and intake system at operator's car position: $\blacksquare$ intake noise calculated from insertion loss, $\square$ intake noise calculated from transmission loss. Shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.16: One-third octave-band sound pressure levels at operator's car position: $\square$ mechanical noise with large clearance piston; $\diamond$ reference mechanical noise with normal piston, (obtained same day); — calculated noise due to piston slap with 8 kHz corner frequency; - - - calculated noise with 3 kHz corner frequency; shaded area represents ± 1 and ± 2 standard deviations of reference operation.

Figure 5.17: One-third octave-band sound pressure level range for normal piston (lower limit of colored stripe) versus undersize piston (upper limit of stripe) at operator's ear position, for five speeds from 3800 rpm to 9600 rpm (as labeled).

Figure 5.18: Impact identification by high pass filtering of acceleration time history. Top curve shows acceleration signal recorded at top of cylinder on pulpwood saw. The next lower curve shows the high pass filtered response to the same input. Remaining curves show similar information for consecutive engine cycles.

Figure 5.19: Wigner distribution of accelerometer signal recorded at top of cylinder for two consecutive cycles of the pulpwood saw. The relative darkness of each area is proportional to the logarithm of the acceleration amplitude. Black indicates maximum level, white areas are at least 40 dB below the maximum level. Arrows, $\leftrightarrow$, indicate horizontal lines (suggesting impacts).

Figure 5.20: One-third octave-band sound pressure levels at operator's ear position: ○ mechanical noise with 8 times normal gudgeon pin bearing clearance; ● mechanical noise with 4 times normal gudgeon pin bearing clearance; ◇ mechanical noise with normal gudgeon pin, (obtained same day); — estimate of bearing noise from eqn. 66; estimate of bearing noise from eqn. 65; shaded area represents ± 1 , and ± 2 standard deviations of reference operation.

Figure 5.21: One-third octave-band laboratory sound pressure levels at operator's ear position: ▽ reference mechanical noise with chain, ◇ reference mechanical noise (obtained same day), shaded area represents ± 1 , and ± 2 standard deviations of reference operation.

Figure 5.22: Average one-third octave-band laboratory acoustic intensity levels scanned over guide bar at 5 cm distance: guide bar (no chain); guide bar and saw chain driven at best cutting speed.

Figure 5.23: A-weighted sound pressure levels for pulpwood saw cutting wood. Stock (gray), levels with exhaust or intake mufflers (vertical stripes), noise produced using acoustical enclosure of powerhead (horizontal stripes).

Figure 5.24: Graph showing estimated sound pressure levels at operator's ear for most noise sources present when the pulpwood saw cuts wood. For comparison, ± 1 standard deviation of laboratory reference residual mechanical noise; ± 2 standard deviations of reference operation.

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