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THIS THESIS EMBODIES THE RESULTS OF
SUPERVISED PROJECT WORK MAKING UP
TWO-THIRDS OF THE WORK FOR THE DEGREE.

ON AERODYNAMIC NOISE GENERATED BY CIRCULAR SAWS

by

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ABSTRACT

A number of models have been proposed by various investigators to describe the characteristics of saw noise. However, none of the proposed models has been completely successful. Thus an acceptable solution to the problem of noise prediction requires greater understanding of the noise generation mechanisms.

The noise produced by a free running circular saw can be attributed to two sources. Resonant vibration of the saw blade excited by interaction with the turbulent flow over the teeth is one source of noise while interaction between the rigid non vibrating saw blade and the surrounding air stream is a second source of noise. The noise produced by the latter mechanism is called aerodynamic noise and this thesis is concerned with investigation of the latter noise source.

It is generally accepted that the source of the aerodynamic noise is the fluctuating side to side forces exerted by the teeth on the induced turbulent air stream and thus it may be modelled as a collection of point dipoles acting at each saw tooth. However, the experimental verification of the theory is not complete and this thesis is directed at further investigation of the characteristics of the point dipole theory.

The point dipole theory is supported by the results of experimental investigation of the dependence of the radiated sound power on tooth area. However, the observed noise directivity pattern deviates from that predicted by point dipole theory. The effect of carbide tipping on saw noise generation is investigated and found to be relatively insignificant. Gullet width is observed to be of limited importance to saw noise. It is also found that the incident air flow speed relative to the saw blade is proportional to the peripheral speed of the saw.

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STATEMENT OF ORIGINALITY

This thesis contains no material which has been accepted for the award of any other degree or diploma in any University. To the best of the author's knowledge and belief, this thesis contains no material previously published or written by another person, except where due reference is made in the text.

SANGARAPILLAI KANAPATHIPILLAI

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NOMENCLATURE

p	Acoustic pressure at the observation point $\underline{x}$ (N/m^2)
T_{ij}	Fluctuating fluid stress tensor
c	Speed of sound (m/s)
r	Distance between the source point $\underline{y}$ and the point of observation $\underline{x}$ (m)
F_i'	Force per unit area exerted on the fluid by the surface S' (N/m^2)
S'	Area of the boundary surface (m^2)
p_{ij}	Compressive stress tensor
x_i	Directional factor in the $\underline{i}$th direction
V_i	Fluid speed in the $\underline{i}$th direction (m/s)
v	Total volume of the wake fluid surrounding the boundary surface S' (m^3)
A	Total surface area of the saw teeth (m^2)
F_l	Fluctuating lift force acting at each saw tooth (N)
N	Total number of teeth on the saw
ω	Broad band centre frequency (Hz)
U	Peripheral speed of the saw (m/s)
V	Relative air speed over the saw tooth (m/s)
ρ	Density of air (Kg/m^3)

S	Surface area of each saw tooth (m^2)
SPL	Sound pressure level (dB) re 2×10^{-5} (N/m^2)
W	Sound power (Watts)
SPR	Sound power level (dB) re 10^{-12} watts
w	Gullet width (mm)
d	Gullet depth (mm)
b	Saw blade thickness (mm)
h	Maximum tooth thickness across the saw blade (mm)
E	Voltage across the hot wire (volts)
V_{aj}	Air jet speed over the hot wire (m/s)
ϕ	Angle of the relative air flow with the tangent (degrees)
f	Third octave band centre frequency (Hz)
rms	Root mean square

LIST OF FIGURES

1. INTRODUCTION

In industries where saws are used the loud noise generated by their use poses a health hazard to workmen exposed to the noise. Noise levels near the operator's position* often exceed 90 dB(A) and continued exposure to such high levels leads to noise induced hearing loss. In many countries hearing protection in industry is required by law. In most cases this protection is achieved by requiring workers to wear hearing protection devices, usually ear plugs or ear muffs. Unfortunately, these measures often fail to achieve results, because workers find the hearing protection devices too uncomfortable to wear for long periods. Alternative means are needed to reduce the noise associated with the use of saws to acceptable levels. Thus there is a need to understand the noise generating mechanisms associated with the use of saws and this thesis is concerned with one aspect of the problem. The idling noise of a circular saw is investigated in this thesis.

Saw noise is generally considered to fall into two categories, (1) Cutting noise and (2) Idling noise. Cutting noise is the noise generated as a saw is cutting a work piece, while Idling noise is the noise generated when a saw runs free. Cutting noise is usually louder than Idling noise although in certain cases a whistling noise generated by resonant vibration of the saw blade may be louder than cutting noise. Unfortunately,

* Terminology such as operator's position, guller area, tooth area, tooth set etc, are given in Appendix 1

cutting noise is very difficult to control because of its complex nature. However, there are important cases, for example, in a timber mill when a saw may cut only a small percentage (20%) of the time it is in use [1,2]. Thus in such cases a substantial reduction in the total noise exposure of a workman can be attained through the control of idling noise.

An obvious way of controlling idling noise would be by shutting off the saw between cuts. However, this is impractical for a number of reasons. Frequent on-off cycling of motors would lead to excessive overheating. After a saw shut-off, the saw would usually take a long time to coast down to a speed where the noise generation is substantially reduced. Most objectionably, the smooth flow of work would be interrupted by the on-off cycling and this would be unacceptable in nearly all work situations.

To be effective, any method of controlling idling noise must not interfere with normal work procedures. To achieve a reduction in the noise generated, the nature of the noise source and its important parameters must be understood. Thus a theoretical model must be developed.

Idling noise is usually divided into two types:

(1) Whistling noise, which is the noise produced by resonant vibration of the saw blade. Whistling noise is of tonal quality and can be very loud, in some cases producing noise levels over 100 dB(A) [3]. Much research has been performed on whistling noise and effective methods of controlling it has been developed [1-17].

(2) Aerodynamic noise is generated through interaction of the rigid saw with the surrounding air stream. Aerodynamic

noise frequently exceeds an acceptable noise level at the operator's position [3]. Thus the reduction of aerodynamic noise may result in a substantial reduction in noise dose incurred by workers.

Considerable effort has been devoted to identify and control the source of the aerodynamic noise generated by circular saws. Early investigations on saw noise were mainly directed at experimental observations. In recent years noise source models have been suggested which are based on theoretical considerations and empirical arguments [1,2,14].

Most researchers have concluded that the dominant source of the aerodynamic noise may be modelled as a collection of point dipoles located on the saw teeth [1-3,14,18,36]. However, the source identification remains a subject of controversy. This thesis is directed to further investigation of the validity of the point dipole model to describe the saw noise.

The sound power radiated by the rotating circular saw is observed [1-3,18,20,34,35,37] to be proportional to the tooth speed* raised to a power ranging between 4.9 and 5.6. Thus it is generally accepted that tooth speed is the most important parameter affecting saw noise. However, there is no agreement on what may be the second most important parameter or parameters [17,18,19]. Theoretical consideration of possible source models indicate that the radiated sound power from the circular saw is proportional to the square of the tooth area [1,2,14,36] and empirical evidence to be presented supports this conclusion. Alternatively, empirical evidence [19,35,37] suggests that the radiated sound power from the circular saw is proportional to the gullet

* Tooth speed is the peripheral speed of the saw

area to a power of 1.6.

Although most researchers agree on the point dipole model, the empirically determined radiated sound power dependence on tooth speed deviates from the expected value of six and this deviation has not been fully explained. In the current study, two characteristics of the aerodynamic saw noise which have not been previously examined and which should behave in a specific manner for a point dipole source are investigated. These characteristics are the directivity pattern of the generated noise and the dependence of the radiated sound power on tooth area.

To perform these investigations, tests were conducted with model saw blades. On the basis of the tests performed, evidence will be presented to show that the sound source of the aerodynamic noise is in part point dipoles.

2. REVIEW OF EARLY WORK ON AERODYNAMIC NOISE

Aerodynamic noise generated by the rotating circular saw has been recognised as the result of the interaction of the rigid saw teeth and the surrounding air stream [3,5]. It thus becomes necessary to understand and identify the mechanism responsible for noise generation by the flow - surface interaction. Previous investigations have related radiated sound power and the peripheral speed of the saw rather than the characteristic incident air speed relative to the teeth, thus it is of interest to investigate the relationship between the relative air stream speed, which is responsible for the noise generation and the peripheral speed of the saw, before considering flow - surface interaction.

2.1 Incident Air Flow

Early investigators made different predictions for the speed of the incident air stream relative to the saw. Pahlitzsch & Friebe [7] predicted that the incident air stream speed should be 25 percent of the tooth speed. By treating peripheral speed of the saw as the characteristic speed of the incident air stream, Slone & Robertson [20] observed Strouhal number of 0.12 for the broad band aerodynamic noise [3,14] generated from the circular saw. They suggested that the relative air stream speed is 60% of the peripheral speed of the saw. This was based on the expectation that the Strouhal number must be equal to a value in the range from .20 to .23 for the generation of the aerodynamic noise.

It is generally assumed by all other investigators that the tooth speed is approximately equal to the air stream speed relative to the saw. Alternatively the relative air stream speed is assumed proportional to the tooth speed. However, none of the previous investigators experimentally determined the flow speed except Cho [14], who measured the mean velocity of air flow by fixing a hot wire anemometer behind a tooth model fixed on a rotating disc. He observed that the mean relative air stream speed increases non linearly with the tooth height, and is on the order of the tooth speed.

Dugdale [5] measured the direction of flow of the air stream relative to the saw by fixing a probe on the saw blade inside the root of the gullet. He observed that the direction of flow varies from approximately 0 to 10 degrees with the tangent at the disc edge, depending upon the boundary layer thickness and the position where the flow measurement is made. Cho concluded that the air flow is in the tangential direction based on his observation on turbulence measurements, which indicated that the axial turbulence is negligible compared to the tangential turbulence, within the tooth height, except near the edge.

In the current study, an experiment was undertaken to determine the relative air stream speed and observe its relationship with the peripheral speed of the saw.

2.2 Flow-surface Interaction

Using dimensional analysis, Curle [21] extended Lighthill's theory [22] of aerodynamic noise generation by turbulent flow and showed that noise radiated from the unsteady flow over a solid boundary is attributable to the dipole field distribution on

the solid boundary. The resulting acoustic power is dependent on the characteristic flow speed raised to the sixth power. Curle's result was substantiated by Clark and Ribner [23], who performed measurements of the reaction force on the surface of an airfoil exposed to turbulent flow and correlated these with farfield acoustic pressure measurements. Siddon [24] supported Curle's dipole theory for turbulent flows over airfoils by cross correlation methods, and, Hersh and Meecham [25] performed a check on the predicted directional distribution of the noise radiated by a small air foil exposed to flows. Their investigation compared favourably with the point dipole theory of Curle for acoustically compact surfaces. In addition, Sharland [26] experimentally observed that a small flat plate exposed to turbulent flow produces a noise directivity pattern similar to that expected for a point dipole. For the case of circular saws, the teeth are found to be acoustically compact surfaces [3] (i.e. dimensions of the individual saw teeth are small compared with the wavelength of radiated sound). Thus the source of the noise can be regarded as point dipoles acting at each tooth.

The resulting dipole field on each saw tooth arises from surface pressure fluctuations associated with the turbulent boundary layer and the incident flow. Boundary layer separation leads to periodic vortex shedding at the trailing edge of each tooth [26]. In this respect, early investigations of saw noise failed to observe the vortex shedding phenomenon [14,27,28], however, recently Price and Mote [29] have observed periodic vortex shedding behind a single tooth on a rotating disc. They concluded that the vorticity shedding is maintained as the tooth number is increased but the strength of the vorticity is reduced. Thus the theory that point dipoles are located at each tooth has

been substantiated.

However, as the expected scaling value of six for the acoustic power dependence on tooth speed is not experimentally observed, rather a power more nearly five is observed, it is necessary to look for a model which explains the empirical observation. It has been shown by various researchers that the acoustic power radiation resulting from interaction between flow and acoustically non-compact surface is proportional to the characteristic flow speed raised to a fifth power [30,31,32]. Thus it might be possible to explain the acoustic power radiated by circular saws as arising from a combination of interaction between the incident air stream and both compact and non-compact surfaces. The former arising from the teeth and the latter arising from the surface of the saw blade depending upon the direction of relative air flow across the blade. As will be shown in Sections 5.1 and 3, the relative air flow is essentially tangential to the periphery of the blade so that interaction between the incident air stream and the saw surface leads to insignificant contribution to saw noise.

2.3 Source Prediction

In their experiment with circular saws Reiter and Keltie [3] observed that the radiated sound power is approximately proportional to the tooth speed raised to the power of 5.3 and that the noise directivity pattern was similar to that of a rotating disc as observed by Chanaud [33]. From these observations they proposed that the dominant noise sources are dipole in nature. Subsequently, based on dimensional arguments and the results of experiment, they found [34] that the sound power was proportional to the tooth speed raised to the power of 5.05.

Consequently, they postulated that the noise source is a combination of monopoles and dipoles.

Slone and Robertson [20] observed that the source of the saw noise is dominantly of dipole nature. Later on, Segal, Becker, Slone and Robertson [35] proposed that the aerodynamic noise generated by circular saws is due to amplified quadrupole sound sources. This conclusion was based on a concept proposed by Ffowcs Williams and Hall [30] for flows over non-compact surfaces. The consequent change in the opinions with regard to the source prediction by the above investigators were primarily based on the acoustic power dependence on tooth speed.

In support of the dominant dipole model, Cho and Mote [36] derived an expression for sound power radiation from the wave equation with the aid of empirical arguments. This relationship indicates that the radiated acoustic power from the saw is proportional to the tooth speed raised to a power which may be less than the characteristic exponent six, of a dipole sound field. However, they concluded that the source of the aerodynamic noise is dominantly of dipole nature and any deviation from the dipole characteristic is attributable to Reynolds number variations of the incident air flow and the tooth design.

Stewart [18] experimented with hardboard discs to model circular saws and he drew an analogy to fan noise theory to derive an empirical formula for the sound pressure level produced by the saw. This model indicates that the sound pressure level at the operator's position in free field is approximately proportional to tooth speed raised to a power of 5.6. It was concluded by Stewart that the source of noise is of simple dipole nature.

By treating each saw tooth as a small air foil, Bies [1] developed a mathematical model for the sound power radiated by the rigid saw blade and concluded that the primary source mechanism responsible for the saw noise is due to point dipoles acting at each tooth of the saw as proposed by other researchers who support the dominant dipole model.

Review of early work on saw noise indicates that consideration given to source models largely ignored the associated noise directivity patterns. Reiter and Keltie [3] and Stewart [18], experimentally investigated the directivity pattern of the noise produced by circular saws, however, in their investigations they failed to compare their experimental results with any appropriate models which might describe saw noise.

Although most researchers agree that the source is most likely a distribution of point dipoles located at the saw teeth, the identification of the source of aerodynamic noise radiated by the circular saw remains open to question because of

(i) the experimentally observed sound power dependence on tooth speed

and

(ii) the deviation in the noise directivity pattern from that of a point dipole source, as shown in this thesis.

2.4 Parameters Contributing to Noise Generation

The dipole models propossed by Cho and Mote [36] and Bies [1], indicated that the total radiated noise should be dependent of the square of the tooth area. Thus, verification of such a dependence will provide support for the dipole model.

In his experiments with hardboard discs, Stewart [18] observed that the noise level was maximum at a ratio of gullet width to blade thickness of approximately 2.5. His investigations suggest that the gullet width is an important parameter in saw noise generation.

In a design and application manual published by Wyle Laboratories, U.S.A. [19], an empirical equation for sound pressure level produced by circular saws as measured at the operator's position was presented. This equation includes gullet area as a substantial parameter in noise generation, which indicates that the sound pressure level is proportional to gullet area raised to the power of 1.6. The significance of the gullet area is not understood on theoretical grounds, however, this finding was based on experiments performed with over one hundred industrial circular saws.

From their experimentation with tooth models, Cho and Mote [36] concluded that the strength of the source of the aerodynamic noise generated is increased with tooth height, tooth back angle and carbide tipping on teeth. They observed that the effect of these parameters, however, lead to a variation in noise level of less than 5 dB. Zockel et all [17] experimentally observed that a doubling of the tooth set parameter seemed to increase the sound pressure level by 15 dB(A) at the operator's position.

Wyle Laboratories, U.S.A. [19,37] devoted considerable effort to developing design criteria for the suppression of the noise of circular saws. They adopted three different approaches; modification of the saw blade design, closure of the gullets and roughening the saw blade in a band at the root of the teeth.

Based on the notion that the intensity of vorticity shedding can be reduced by bevelling the trailing edges of each saw tooth in various patterns, they achieved a maximum noise reduction of the order of 3 dB(A). Maximum noise reduction was observed for the case in which consecutive teeth were bevelled in an alternating fashion at an angle of 30 degrees. This shows that the geometry of saw teeth is relatively insignificant in noise generation. It was observed by Wyle Laboratories that the elimination of the conventional gullet from the saw blade by using gullets 'scooped out' on only one side of the disc, results in a reduction of aerodynamic noise of the order of 10 to 14 dB(A). However, the viability of this design is in question, as the cutting efficiency is reduced.

By roughening one side of each tooth extending down to approximately 12 mm below the gullets [37], maximum noise reduction on the order of 3 dB(A) was achieved. It was postulated that the roughened surface reduces the vorticity shedding, based on the notion that such a surface creates more homogeneous vorticity shedding at the saw blade trailing edge. This view is not in agreement with the concept that roughening the surface will increase the strength of the shear dipoles [33] and thus increase noise levels. However, as roughening of the saw disc achieves at most a noise reduction of 3 dB(A), analysis of the mechanism involved is moot as this is not a significant reduction in noise levels.

Review of early work on the contribution of the physical parameters to saw noise suggests that gullet area, gullet width, tooth set and tooth area may be of importance. It will be shown in this thesis that tooth set is of minimal importance and that gullet width is of only limited importance. The predicted dependence upon tooth area is confirmed.

3. MECHANISM OF AERODYNAMIC NOISE GENERATION

When a saw rotates the surrounding air is carried with it because of the viscosity of the air and ejected outwards across the surface by the centrifugal forces [38]. These centrifugal forces on the air stream provides a pressure gradient similar to that in a centrifugal compressor. To replace the ejected air stream an air flow is induced in an axial direction as shown in figure 3.1. The induced air stream from the saw spirals outward in the direction of motion of the saw, which results in a boundary layer separation from both surfaces of the saw blade edge. These boundary layers mix so that the resulting air stream becomes highly turbulent at normal operating speeds of saws.

As the saw teeth pass through the turbulent air stream, they are subject to flow - surface interaction which radiate sound. Experimental evidence suggests that the characteristic wavelength of the noise radiated by the circular saw is considerably larger than the dimensions of a saw tooth, which means that the saw teeth are acoustically compact surfaces [3]. At subsonic flow speeds an acoustically compact surface generates noise, mainly due to the force fluctuations associated with the turbulent boundary layer developed on its surface, and of the incident flow from the upstream wake [39].

Figure 3.2 shows different types of possible sound sources which can be produced by rotating circular saws. While the saw is rotating, monopole sound sources result when each tooth

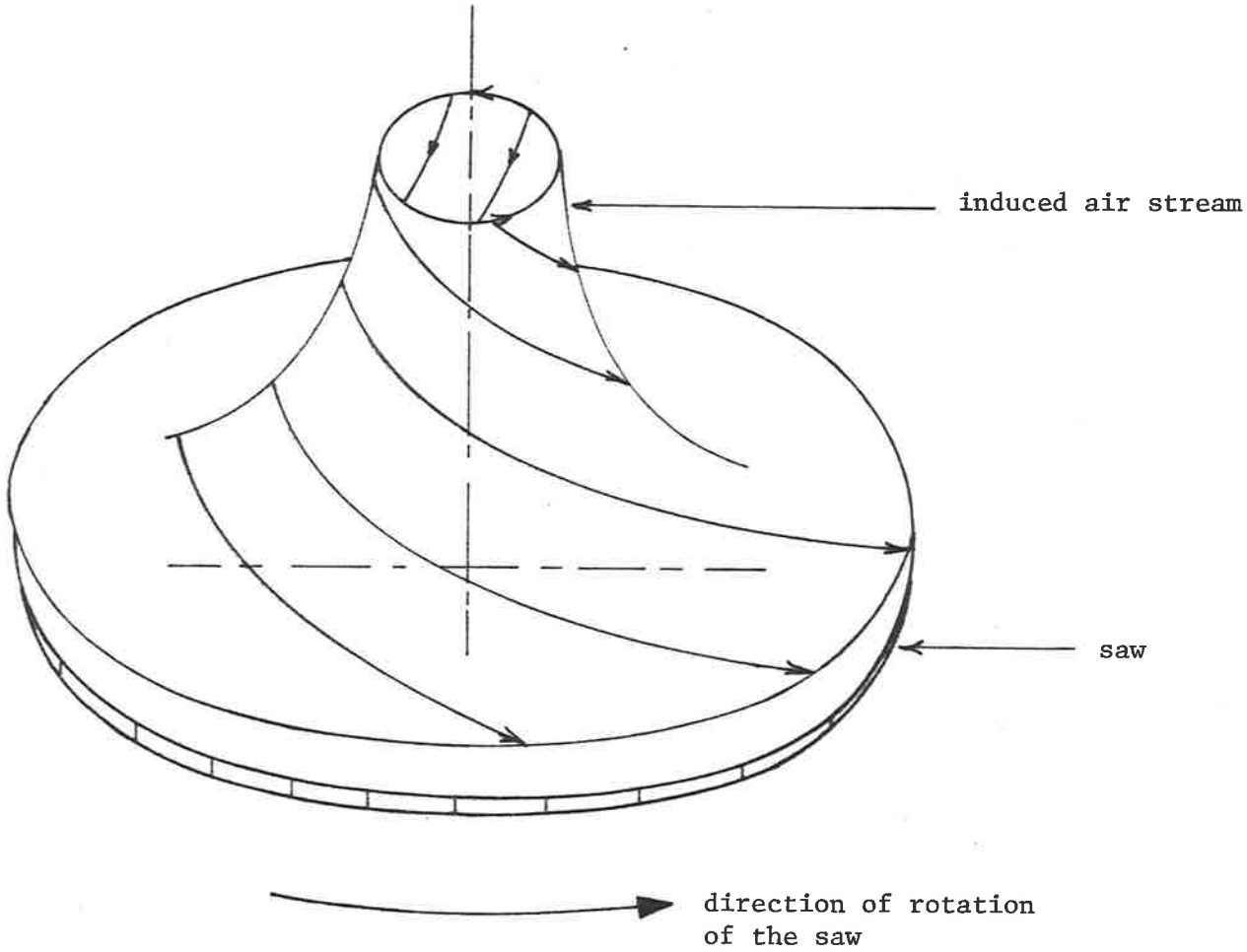


Figure 3.1 Rotating saw and the induced air stream

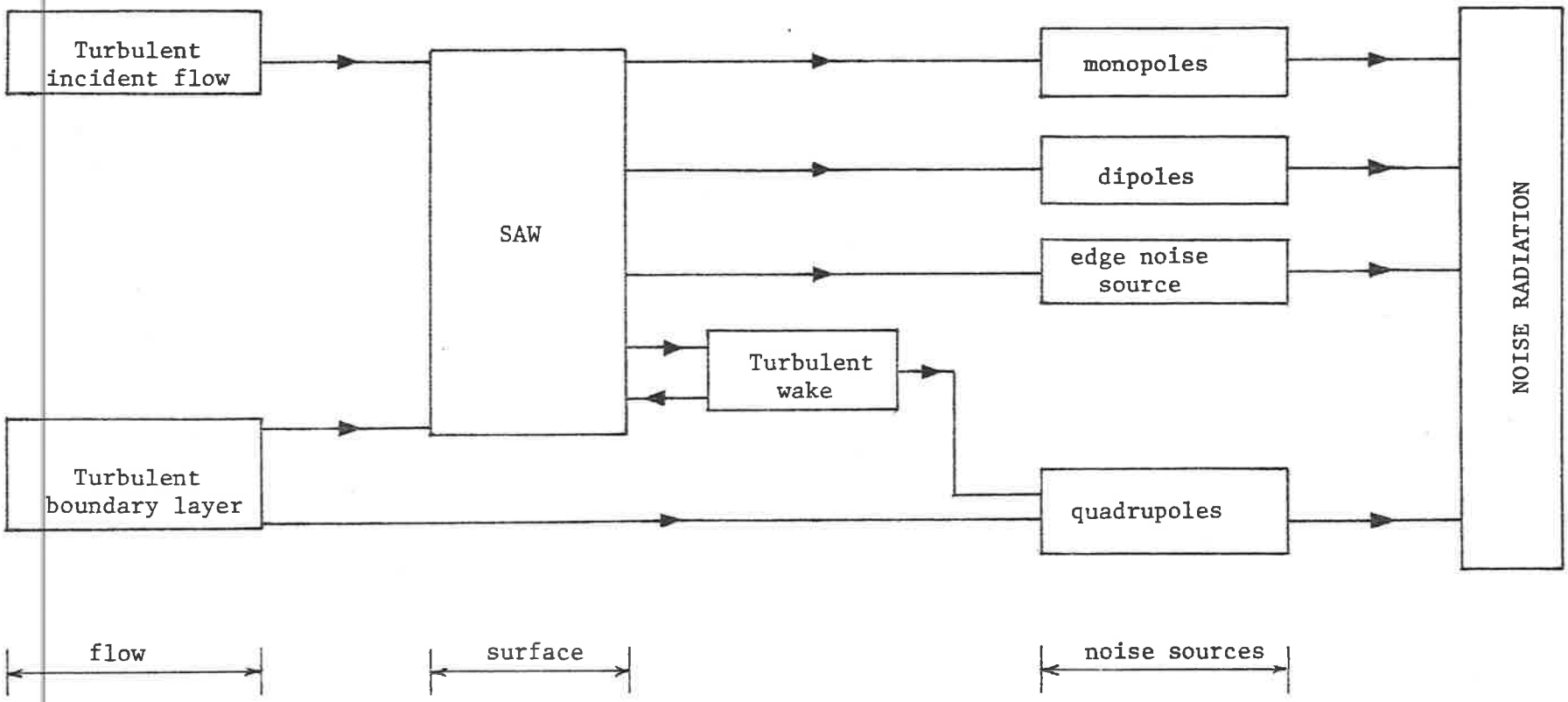


Figure 3.2 Noise sources in a rotating saw

displaces the surrounding air stream causing mass fluctuations in the air stream. Dipole sound sources are produced due to the force fluctuations on the saw teeth arising from the interaction of the turbulent boundary layer and the surface, and due to the interaction of teeth with the turbulent wakes from preceeding teeth. Quadrupole sound sources arise from density fluctuations in the turbulent boundary layer and the turbulent wake surrounding the periphery of the saw blade. The possibility for the existence of monopoles, dipoles and quadrupoles is described by Cho [14] in his investigation on saw noise.

Another possible type of noise source, called trailing edge noise, may contribute to the noise generated by circular saws. The trailing edge noise occurs when the characteristic wavelength of the radiated sound is smaller or comparable with the dimensions of the saw. A brief review on edge noise is given in Appendix II.

The significance of the trailing edge noise depends on the direction of the air flow at the periphery of the saw blade. Experimental results presented in Section 5.1 indicate that the relative air flow direction is approximately tangential. This means that the wake from the surface will be intercepted by the following teeth and thus results in dipole type acoustic source rather than a trailing edge noise source. Therefore the trailing edge noise contribution to saw noise can be neglected.

In order to investigate the saw noise characteristics, the point dipole models suggested by Cho and Mote [36] and Bies [1] are considered in the next Section.

3.1 Noise Source Model

Lighthill's theory [22] of aerodynamic noise generation which was directed towards fluid flows in the abscence of solid boundary surfaces was extended by Curle [21] to allow for the presence of such surfaces. Curle's result for the radiated acoustic pressure $p(\underline{x},t)$ at the observation point $\underline{x}$ at time t is given by,

$$p(\underline{x},t) = -\frac{1}{4\pi}\frac{\partial}{\partial x_i}\int_{S'} F'_i\,(\underline{y},t-\frac{r}{c})\,\frac{dS'(\underline{y})}{r}$$

$$+\frac{1}{4\pi}\frac{\partial^2}{\partial x_i\,\partial x_j}\int_{V} T_{ij}(\underline{y},t-\frac{r}{c})\frac{dv(\underline{y})}{r} \qquad (1)$$

The symbols in equation (1) are defined in the Section on nomenclature. For quick references the following symbols are re-defined here.

$T_{ij} = \rho V_i V_j + p_{ij} - c^2\delta_{ij}$ is the fluctuating fluid stress tensor per unit volume

$i = 1,2,3$

$j = 1,2,3$

$$\delta_{ij} = \begin{cases} 1 \text{ when } i = j \\ 0 \text{ when } i \neq j \end{cases}$$

F'_i is the force per unit area exerted on the fluid by the boundary surface S'.

p_{ij} is the compressive stress tensor

As mentioned in Section 3.0 there is a possibility of monopole sound sources as well as dipole and quadrupole sound sources which may contribute to the total saw noise. However, the previous observations indicate that the noise from monopole sources is not significant [20]. Consequently equation (1) seems applicable to circular saws.

The following assumptions are made for the far field analysis of the saw noise.

(1) Each saw tooth is identical in producing noise.

(2) Tooth motion does not effect the total noise level received at the observation point $\underline{x}$.

By considering a stationary frame of reference (X1,X2,X3) as shown in figure 3.3, equation (1) can be rewritten as,

$$p(\underline{x},t) = \frac{x_i}{4\pi c r^2} \int_{S'} \frac{\partial F_i'}{\partial t} (\underline{y}, t - \frac{r}{c})\, dS' + \frac{x_i x_j}{4\pi c^2 r^3} \int_{V} \frac{\partial^2 T_{ij}}{\partial t^2} (\underline{y}, t - \frac{r}{c})\, dv \qquad (2)$$

The first integral denotes the contribution to acoustic pressure (p) by the surface dipoles while the second integral represents quadrupole sound sources. On the basis of the experimental observations made by Chanaud [33] for rotating discs, the noise due to pressure fluctuations on the saw surface excluding the teeth area can be neglected. Based on the assumption that the dipoles acting on the saw teeth are uncorrelated [1,14], the time retardation term in equation (2) can also be neglected for far field analysis, hence the expression for p becomes,

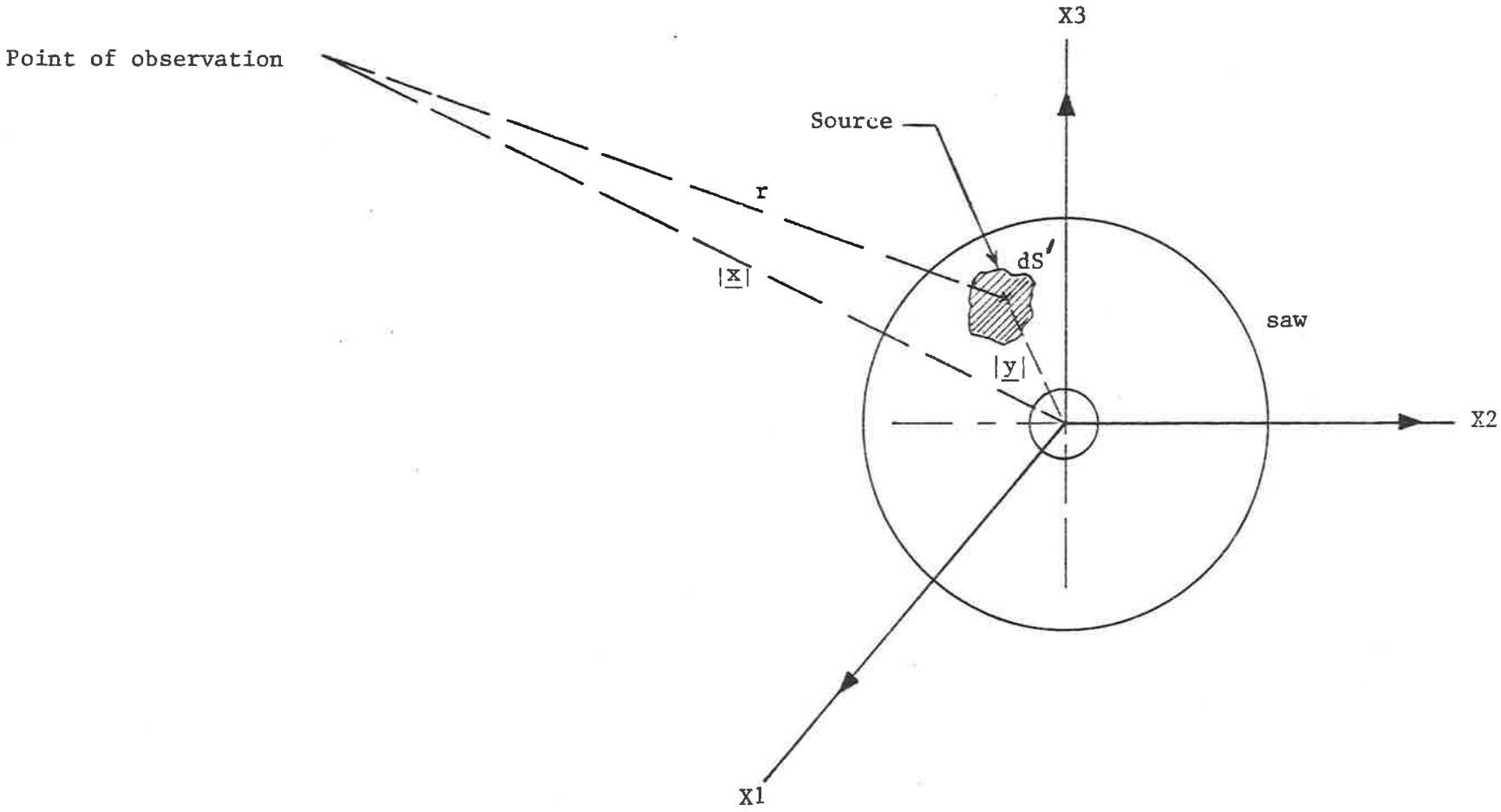


Figure 3.3 Saw noise radiation in free field.

$$p(\underline{x},t) \simeq \frac{x_i}{4\pi c r^2} \int_A \frac{\partial F_i'}{\partial t} \, dS' + \frac{x_i x_j}{4\pi c^2 r^3} \int_V \frac{\partial^2 T_{ij}}{\partial t^2} \, dv \qquad (3)$$

where A is the total surface area of the teeth.

For low Mach number flows, Curle [21] suggested that the dipole sound radiation should be greater than that of the quadrupole sound sources. This is the case in circular saws where the incident air flow has a Mach number of approximately 0.2. By neglecting the effect of quadrupole sound sources [14], equation (3) can be reduced further to

$$p(\underline{x},t) \simeq \frac{x_i}{4\pi c r^2} \int_A \frac{\partial F_i'}{\partial t} \, dS' \qquad (4)$$

As a saw tooth surface has been found to be acoustically compact [3], fluctuating forces on each tooth can be represented by a point dipole. It follows that the integral in equation (4) can be considered for each saw tooth individually as expressed by,

$$p(\underline{x},t) \simeq \frac{1}{4\pi c} \sum_{k=1}^{N} \left(\frac{x_i}{r^2}\right)_k \left(\frac{\partial F_i}{\partial t}\right)_k \qquad (5)$$

where N is the number of teeth on the saw.

F_i is the fluctuating force acting at each saw tooth in $\underline{i}$th direction

$(\ )_k$ represents the parameters with reference to $\underline{k}$th tooth.

A rotating saw tooth can be regarded as an airfoil passing through the surrounding air stream [1]. From the observations made

by Hersh and Meecham [25], the fluctuating drag forces are negligible for such an airfoil so that only the lift forces need to be considered and equation (5) leads to

$$p(\underline{x},t) \simeq \frac{1}{4\pi c} \sum_{k=1}^{N} \left(\frac{x_1}{r^2}\right)_k \left(\frac{\partial F_1}{\partial t}\right)_k \tag{6}$$

As fluctuating lift forces $(F_1)_k$ are considered to be of a random nature [1,14], the expression for the mean square sound pressure becomes,

$$\langle p^2 \rangle \simeq \frac{1}{16\pi^2 c^2} \sum_{k=1}^{N} \left(\frac{x_1}{r^2}\right)_k^2 \left\langle \left(\frac{\partial F_1}{\partial t}\right)_k^2 \right\rangle \tag{7}$$

where < > denotes the mean value.

Equation (7) is important because the mean square of the acoustic pressure can be determined by measuring the rate of change of the fluctuating forces acting at each saw tooth. However, for the present investigations, the following approach is adopted.

Experimental results presented in Section 5.4 indicate that the frequency response of the saw noise is of a broad band nature with a centre frequency (ω) determined by the peripheral speed of the saw (U) and the maximum width of the tooth (h) combined as in the Strouhal number [50].

$$\text{i.e.} \qquad \omega \propto U/h \tag{8}$$

The time rate of change of fluctuating lift force is given by [14],

$$< \left(\frac{\partial F_1}{\partial t}\right)^2 > \propto \omega^2 <F_1^2> \tag{9}$$

The fluctuating lift force F_1 can be expressed in the form [43],

$$F_1 \propto \frac{1}{2} \rho S U^2 \tag{10}$$

where ρ is the density of air (Kg/m^3)

S is the lateral surface area of the tooth (m^2)

From equation (7) and expressions (8), (9), and (10), equation (11) is obtained as follows,

$$<p^2> = \frac{\gamma^2 \rho^2 S^2 U^6}{64\pi^2 c^2 h^2} \sum_{k=1}^{N} \left(\frac{x_1}{r^2}\right)_k^2 \tag{11}$$

where γ is a function of the tooth geometry and the Reynolds number of the air flow over the teeth [1,14].

Based on the assumption made by Cho, γ is written as

$$\gamma = \alpha U^{\beta} \tag{11a}$$

where α is constant of the tooth geometry of the saw, and β is a function of the tooth speed.

Hence, equation (12) can be written as

$$<p^2> = \frac{\alpha^2 \rho^2 S^2 U^{6 + 2\beta}}{64\pi^2 c^2 h^2} \sum_{k=1}^{N} \left(\frac{x_1}{r^2}\right)_k^2 \tag{12}$$

Equation (12) may be expressed in terms of a sound pressure level in dB re 20×10^{-6} N/m^2, viz:

$$SPL = 10(6 + 2\beta) \log U + 20 \log S + C_1 \tag{13}$$

where $$C_1 = 10 \log \left[\frac{\alpha^2 \rho^2}{64\pi^2 c^2 h^2} \sum_{k=1}^{N} \left(\frac{x_1}{r^2} \right)_k^2 \right] + 94$$

Finally, the radiated sound power (W) for the circular saw becomes,

$$W = \frac{\alpha^2 N \rho S^2 U^{6 + 2\beta}}{32\pi c^3 h^2} \tag{14}$$

Or in terms of a sound power level (SPR) in dB re 10^{-12} Watts for the circular saw is expressed by

$$SPR = 10(6 + 2\beta) \log U + 20 \log S + C_2 \tag{15}$$

where $$C_2 = 10 \log \left[\frac{\alpha^2 N \rho}{32\pi c^3 h^2} \right] + 120$$

Note that C_1 and C_2 in equations (13) and (15) respectively are constants for a particular tooth geometry.

4. EXPERIMENTAL METHODS

4.1 Introduction

Industrial saws in general are manufactured with a variety of tooth designs to fulfil specific cutting requirements. However, previous research indicates that tooth geometry is not an important contributor to aerodynamic noise generated by circular saws [37]. Thus, in these experiments normal industrial saw blades were replaced by a conveniently reproducible model which was essentially a slotted disc as shown in figure 4.1(a). The gullet design was chosen to control parameters such as gullet width and gullet depth in a systematic manner and to allow careful observation of the effect of these characteristics on the radiated noise. Figure 4.1(b) shows a portion of a slotted disc in which mild steel tips were brazed on each tooth to simulate tooth set.

Experiments were conducted to investigate the effect of such saw blade parameters as gullet width, tooth area and tooth set on the radiated sound pressure levels, sound power levels, and directivity patterns. In addition to these investigations, the speed and direction of the incident air flow near the blade periphery was measured.

All experiments were performed in the Acoustic Laboratories of the Department of Mechanical Engineering of the University of Adelaide.

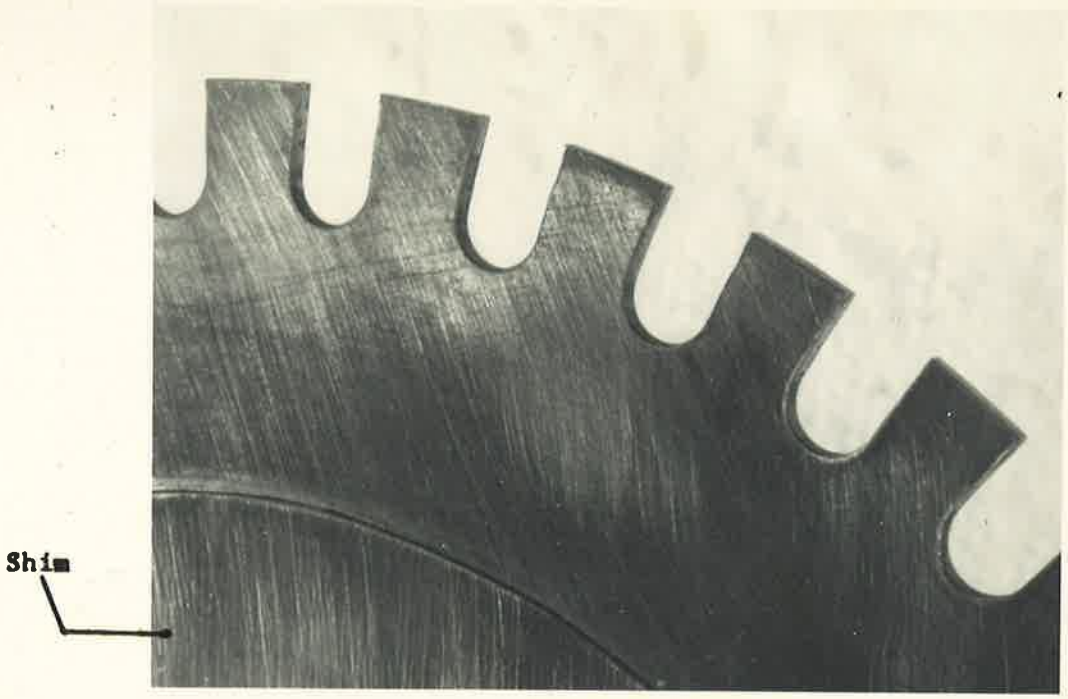


Figure 4.1 (a)

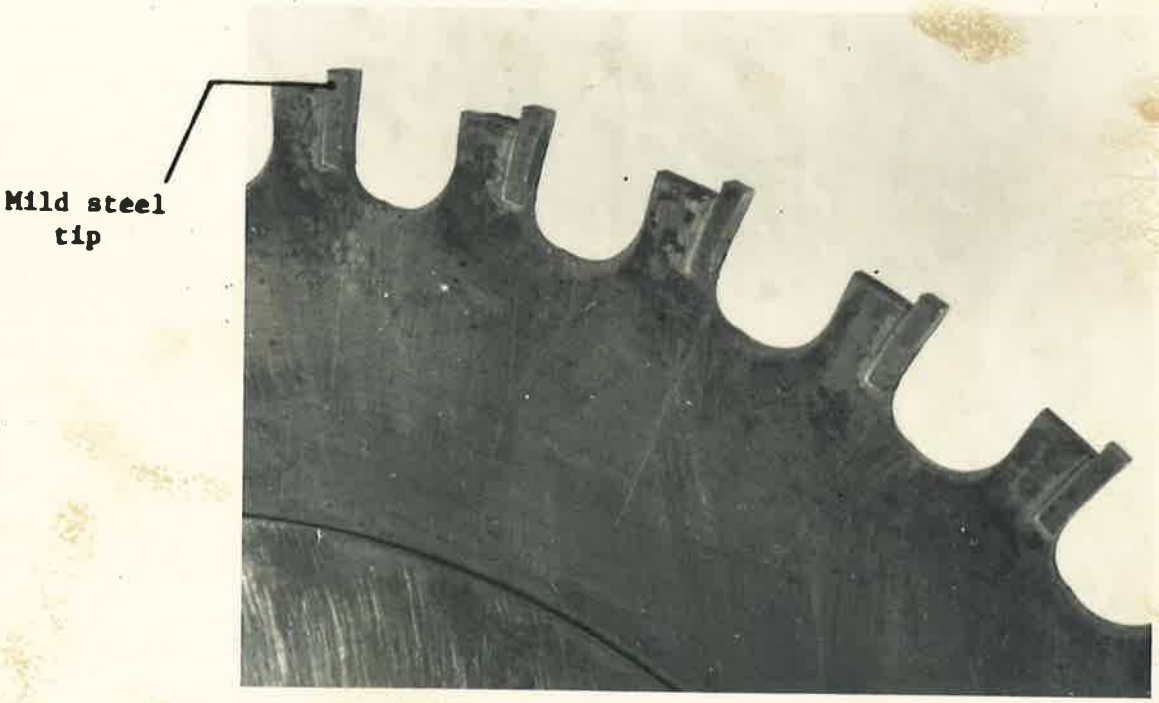


Figure 4.1 (b)

Figure 4.1 Model saw teeth.

4.2 Model Saw Blades

Thirteen experimental saw discs each 3 mm thick and 350 mm in diameter made out of standard saw steel were used in this experiment. Each disc tested had forty teeth and mild steel tips were brazed on four of the discs. A range of tooth set from 1.00 to 2.66 was used in the investigations. To eliminate the whistling noise created by resonant vibrations, all saw blades were damped using 0.25 mm thick, 235 mm diameter steel shims attached on both sides of the saw discs with double sided adhesive tape. The dimensions of the shims were chosen on the basis of suggestions made by Zockel et all [44].

4.3 Free Field Measurements

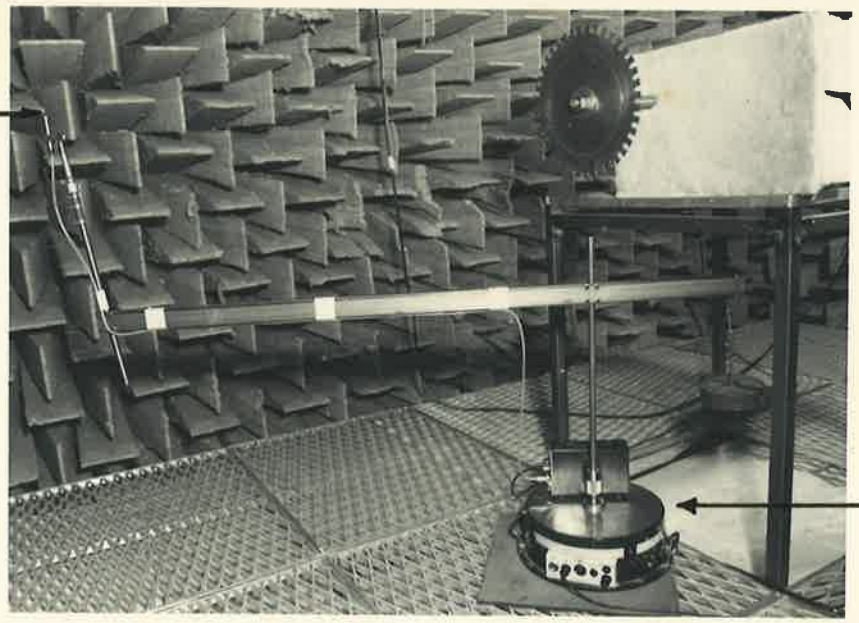
Free field measurements involved in the current study were made in an anechoic room of 249 cubic metres volume. This room consists an inner shell of steel reinforced concrete in which the interior walls are lined with fibreglass 1 metre long wedges to eliminate internal reflections. The room weighs approximately 200 tonnes and is mounted on a large number of coil springs. In addition, rubber pads are used to provide isolation from ground borne vibration. The floor of the room consists of removable sections of aluminium grating; the sections are approximately 1 metre square and are supported by vertical steel posts attached to the concrete floor. The frequency response of the room is limited to above 200 Hz. Thus in the experiments conducted in the current study, a 200 Hz high pass filter was used.

4.3.1 Directivity

Figure 4.2 shows the experimental saw apparatus positioned inside the anechoic room. The variable speed motor used to drive the saw blades was limited to a top speed of 3000 rpm by the vibration of the motor stand. The motor was enclosed in a chipboard box to minimise background noise and the motor shaft was extended 350 mm beyond the edge of the box to minimise sound reflections. The chipboard box was covered with fibre-glass lining to reduce edge and reflection effects on the incident sound.

The saw and motor assembly was mounted on a table which was made out of square section tubular steel and its top surface was a section of square aluminium grating from the floor of the anechoic room. Use of this table was intended to minimise the sound reflection effects from the floor.

A Bruel & Kjaer (B&K) half inch microphone (4133) was used to traverse a horizontal circular path by means of the rotating boom shown in figure 4.2. The radius of the circular path was 1.2 m with the centre at the point of intersection of the saw axis and the plane of the saw. The height of the microphone was kept the same as the saw shaft so that the microphone traverse was on the X1-X2 plane as in figure 3.3. The traverse was dirven by a B & K turn table (3921). A Bruel & Kjaer level recorder (2305) was used to produce polar plots of the measured sound pressure levels for analysis of directivity patterns.



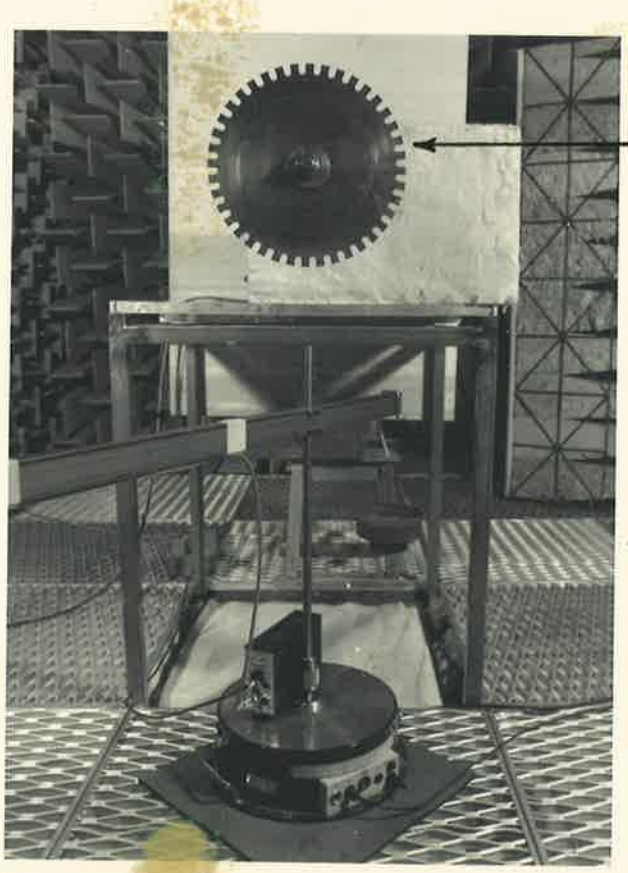
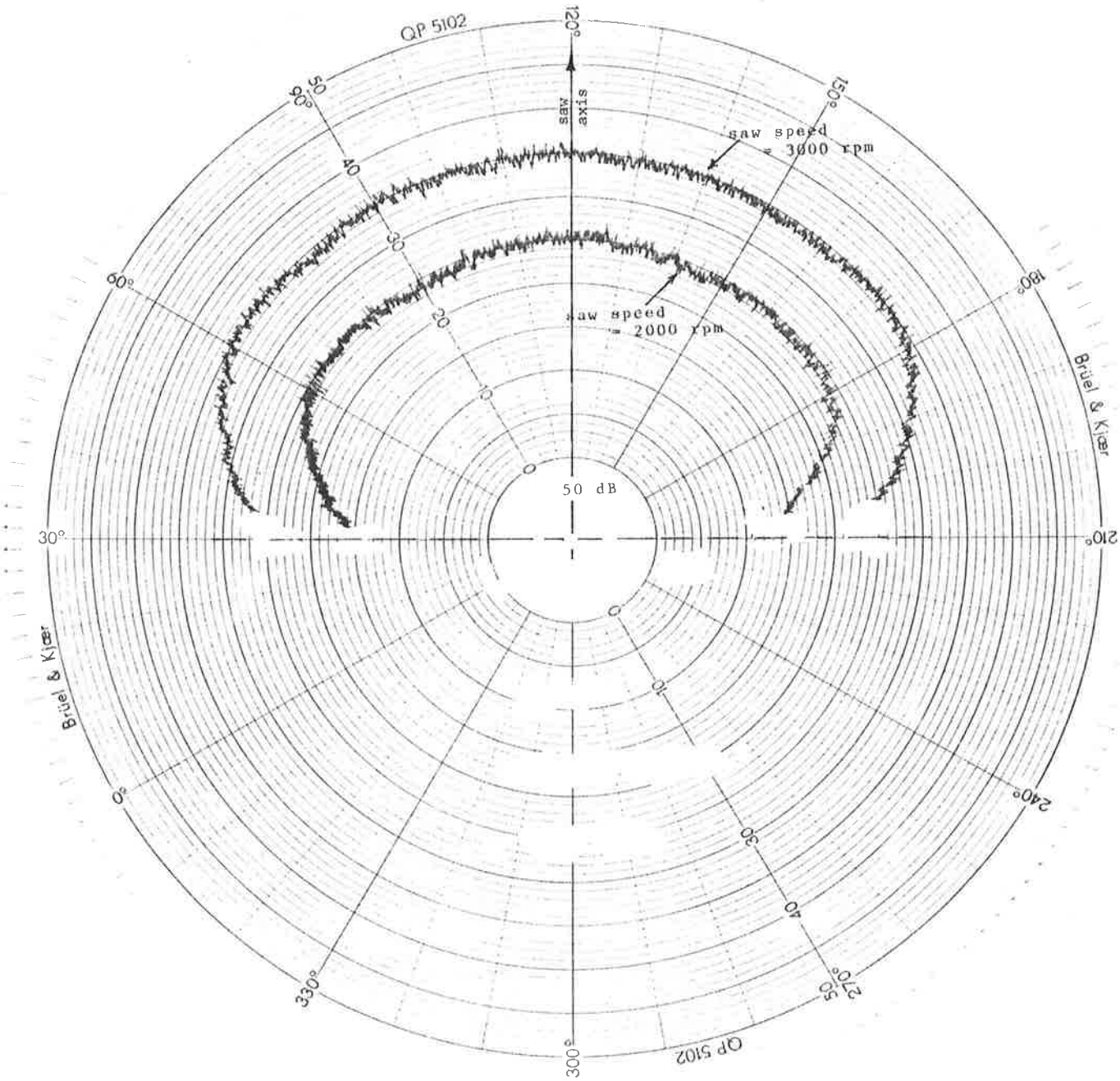


Figure 4.2 Apparatus for noise directivity measurements.

By comparing the directivity patterns of the motor alone and with a saw blade in place the effect of the motor noise was found to be negligible at saw speeds above 1200 rpm. Initial experiments with undamped saw blades showed noise directivity patterns which appeared to result from coherent noise sources. As such coherence was not expected [1,14], attention was given to damp the saw blades as stated in Section 4.2. Figure 4.3 shows a polar plot which illustrates the general characteristics of the radiation patterns of the damped saw blades which were tested at speeds of 2000 and 3000 rpm.

4.3.2 Sound Pressure Levels

Free field sound pressure level measurements were made to investigate the effect of tooth area and gullet width to saw noise generation. The operator's position was considered as the reference point for the sound pressure levels measurements. The saw blades, motor, and supporting apparatus were identical to that described in the previous Section. The turn table and the microphone traverse mechanism were taken out of the room and a B & K half inch microphone (4133) was used in conjunction with a B & K spectrometer (2112) to measure the sound pressure levels generated by the circular saw. The microphone height was kept at the same height as the saw shaft for sound pressure level measurements. The spectrum analysis of the saw noise was carried out with a Hewlett Packard narrow band spectrum analyser (3582A). The schematic diagram of figure 4.4 shows the arrangement used for data collection.



gullet width = 14 mm
gullet depth = 16 mm
tooth set = 1.00

Figure 4.3 Noise directivity pattern of a saw.

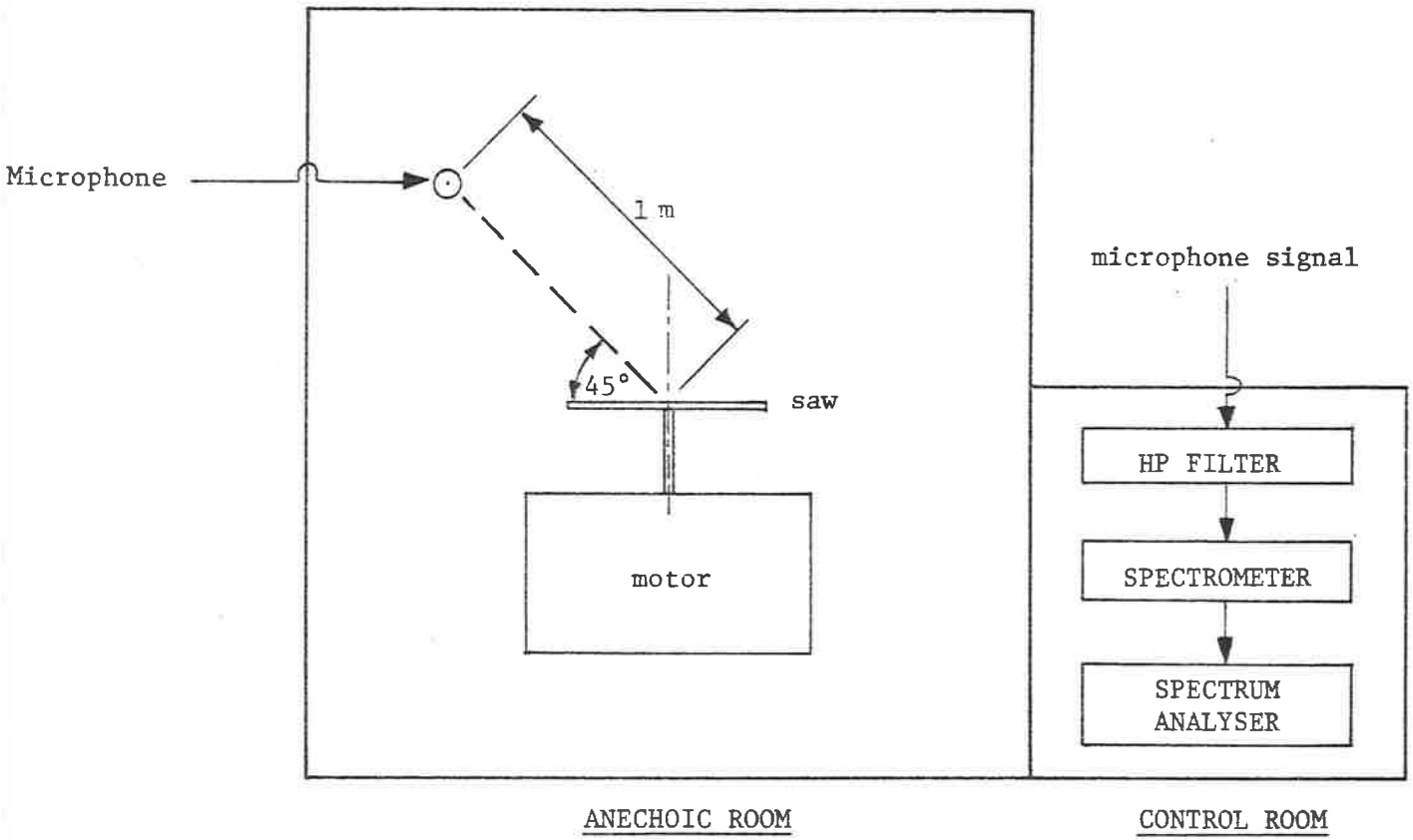


Figure 4.4 Layout for sound pressure level measurements at the operator's position in the anechoic room.

Measurements of the sound pressure levels produced by nine saw blades, each run at various speeds between 1500 rpm to 3000 rpm, were made to study the effect of gullet width and gullet depth on the generated noise levels. A range of gullet width from 3 mm to 16 mm was investigated by utilising various discs with gullets of equal depth. The series of experiments was repeated for a range of gullet depths from 8 mm to 16 mm in increments of 2 mm.

4.4 Diffuse Field Measurements

Experiments to determine the sound power radiated by the circular saw involved measurements in a reverberation room of 105.6 cubic metres volume. This room is constructed with reinforced concrete floor, walls and ceiling, and diffusion is enhanced by a rotating conical diffuser. Standard practices indicate that the room volume of 105.6 cubic metres defines the low frequency limit as 115 Hz [46].

Saw blades were mounted on a horizontal shaft as shown in figure 4.5 for sound pressure level measurements in the reverberation room. The saw was driven by a variable speed motor, enclosed in a chipboard box which was lined internally with fibre glass insulation. A linear traverse was used to perform spatial averaging of the sound pressure levels. A half inch B & K microphone (4133) was used to sense the diffuse sound field generated by the rotating saw blades. The traverse length was 3.75 m and the microphone was traversed at a speed of 50 mm/s. Figure 4.7 shows a typical arrangement of the experimental apparatus used for data collection. The rotating diffuser was run at the recommended speed of 30 rpm to enhance the diffusion of the reverberant sound field [45]. The location of the saw, the

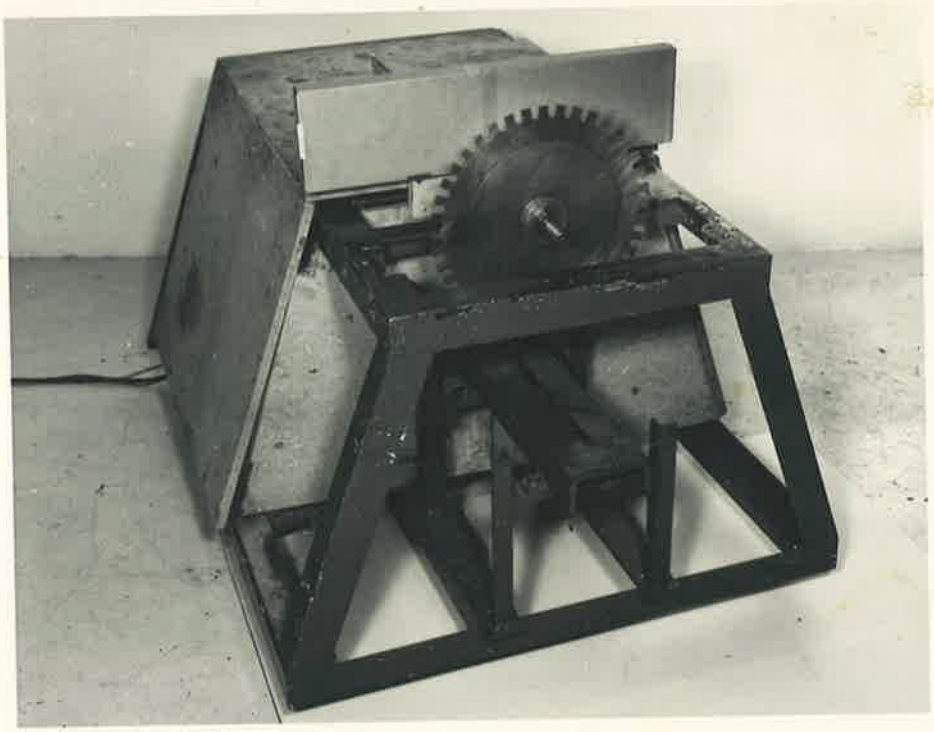


Figure 4.5 Saw rig for reverberation room measurements

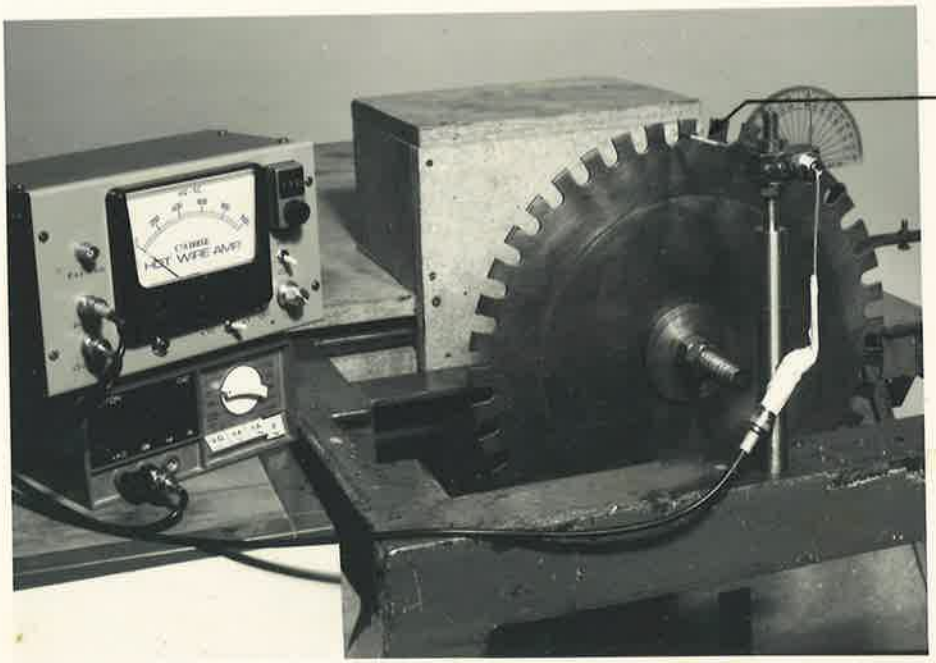
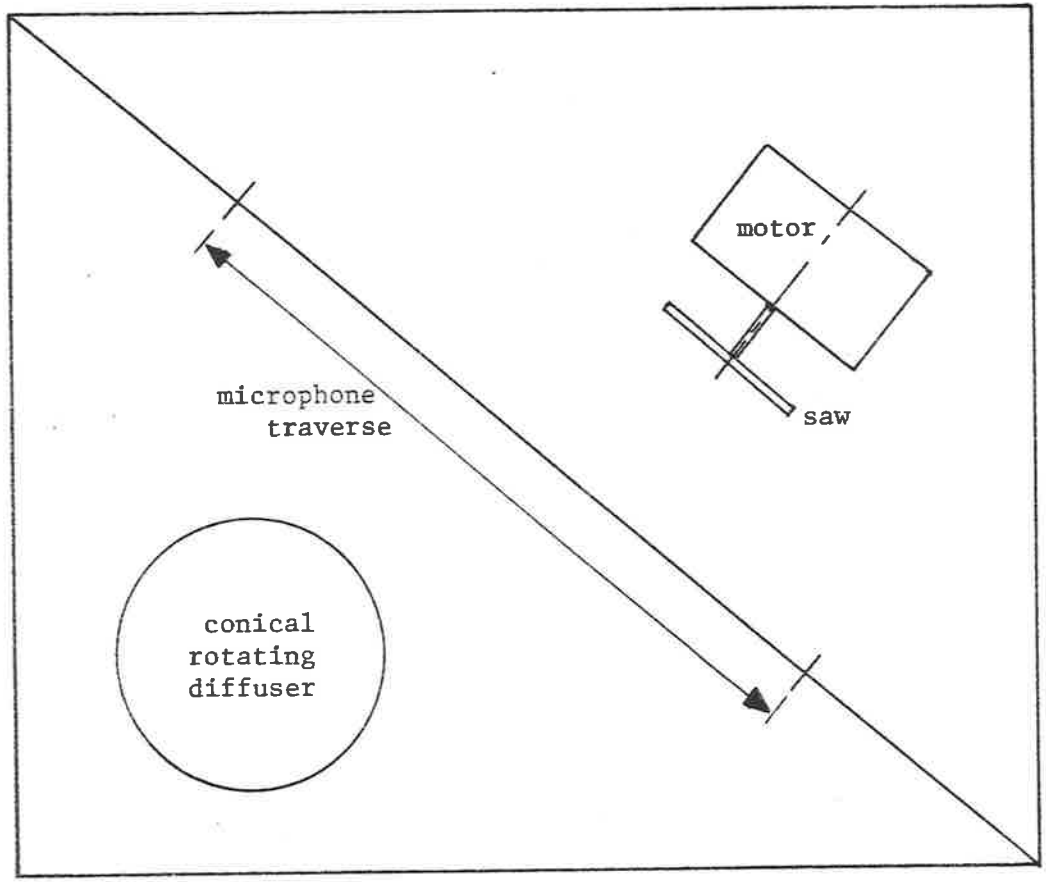


Figure 4.6 Arrangement to measure the incident air speed near the saw surface.



REVERBERATION ROOM

Figure 4.7 Layout for reverberation room measurements.

minimum distance between the microphone and the source, the minimum distance between the microphone and the nearest wall, and the length of the microphone traverse were selected to comply with Australian Standards [47].

Each saw blade was driven at four selected speeds; 1500, 2000, 2500, and 3000 rpm. To improve spatial averaging of the sound power levels, tests were conducted with the saw placed at three locations in the room. The spatially averaged sound pressure levels obtained using these methods displayed a standard deviation of $\pm$ 1 dB.

A one third octave B & K spectrometer (2114) was used to perform spatial averaging (rms) of the sound pressure levels sensed by the microphone over its traverse. The spatial averaging of the sound pressure levels was performed for the dominant one third octave bands. Averaged values of spectral data were transferred to a DEC LSI - 11 micro computer for processing and storage.

The reverberation room was calibrated with an ILG reference sound source and the one third octave band sound power levels were determined by comparison method [45]. The overall sound power radiated by the saw was computed by intergrating the one third octave band sound power levels.

4.5 Incident Air Flow Measurements

The speed of the incident air stream at the saw blade edge was measured with a constant temperature hot wire anemometer.

4.5.1 Description of the Hot Wire

A typical hot wire probe used for this experiment is shown in figure 4.8. The wire used was 5 μm diameter tungsten and was soft soldered to the needles. The ends of the wire was copper plated to facilitate soldering. The active portion of the wire was 1.00 mm long giving a length to diameter ratio of 200. The cold resistance of the wire was typically 6 ohms. Constant temperature operation was used at an overheat ratio of 1:1.5. Before the operation, the time response of the hot wire and the associated feed back loop was set using the square wave injection technique.

In figure 4.9 the circuit diagram for the hot wire operation is presented. The system is based on a high speed National Semi Conductor LM 318 operational amplifier. The current output of this unit is increased by the high performance 2N 2270 power transistor. The bridge ratio is 1:1 with 20 Ω resistors in the upper arms. A 200 KΩ potentiometer was connected to the internal offset control of the LM 318 to control the output offset. No trimming inductances were required because of the compact design and high band width of the LM 318. Frequency response of the hot wire was raised to 15 kHz. A square wave could be injected into one side of the bridge with protection provided by a 10 kΩ resistor and 1.0 μF capacitor.

4.5.2 Calibration of the Hot Wire

The hot wire was calibrated in an air jet by placing it within the core region [48]. The flow velocities were measured with a Pitot tube. To avoid flow disturbance it was necessary to place only one (either the hot wire or the Pitot tube) in

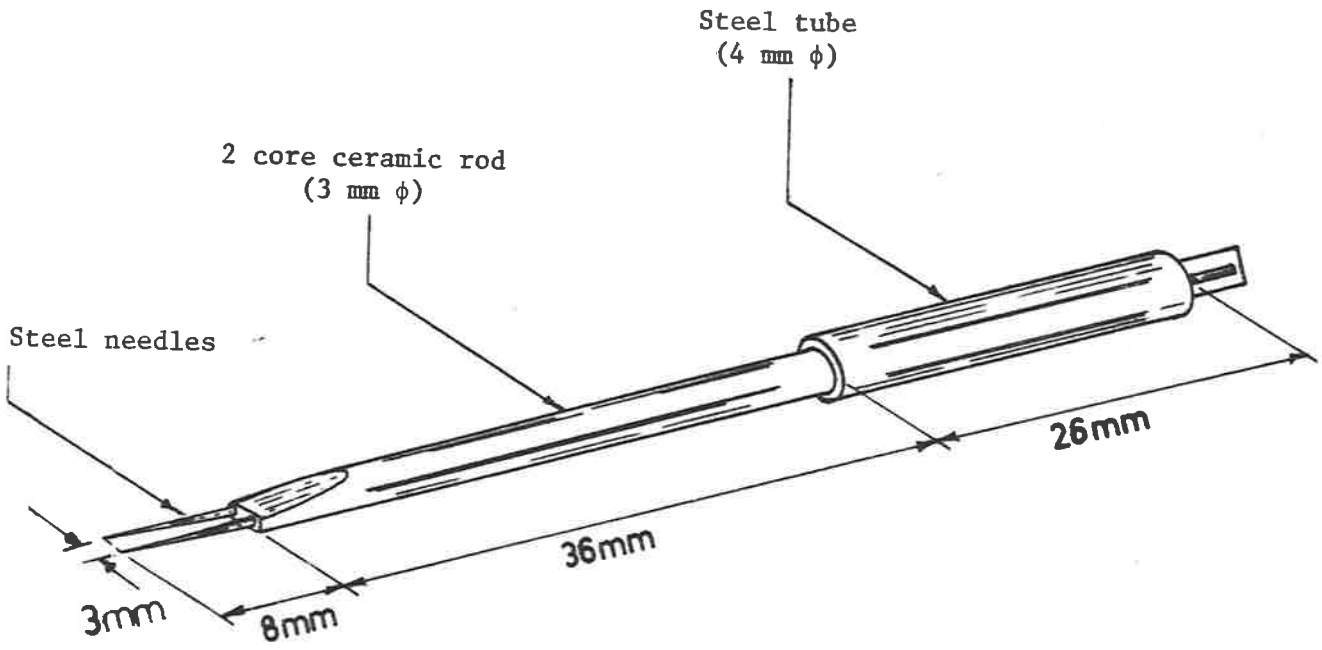


Fig. 4.8 Hot-Wire Probe

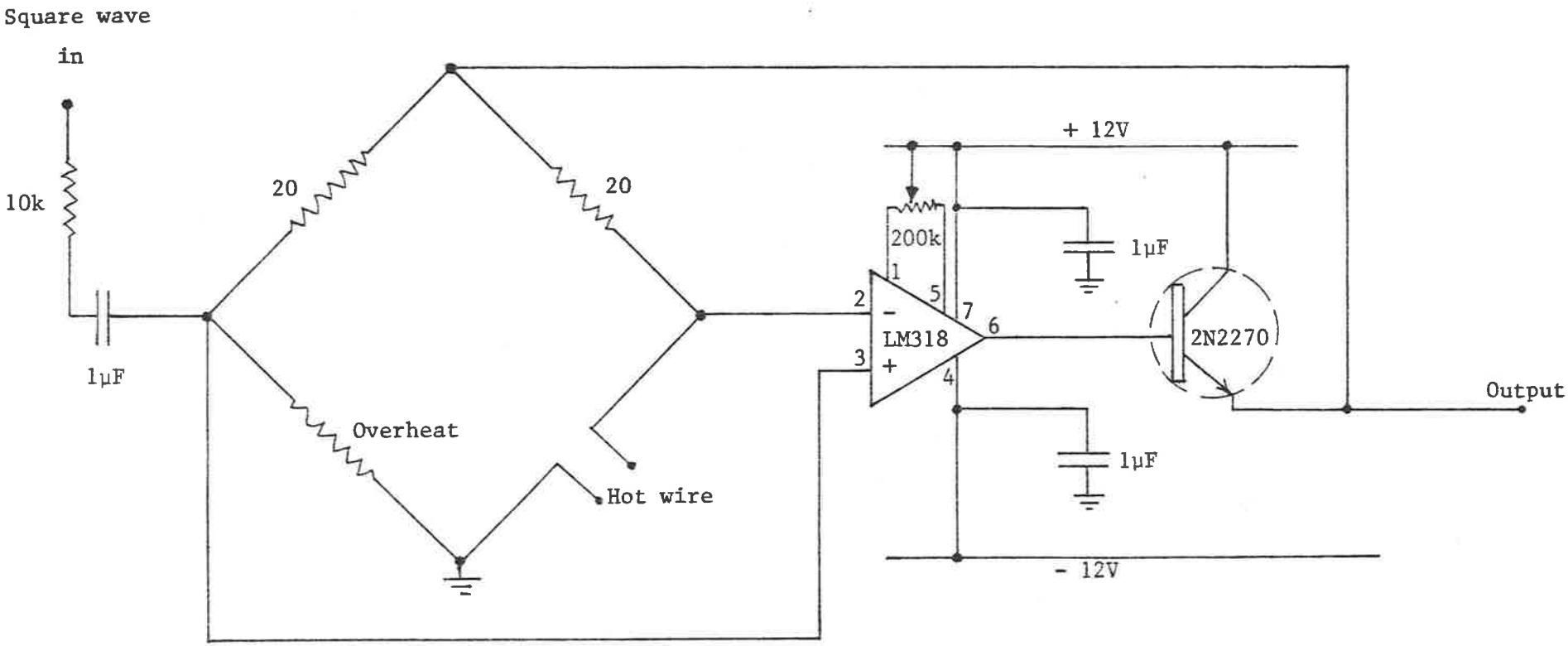


Figure 4.9 Circuit diagram of the constant temperature feedback bridge.

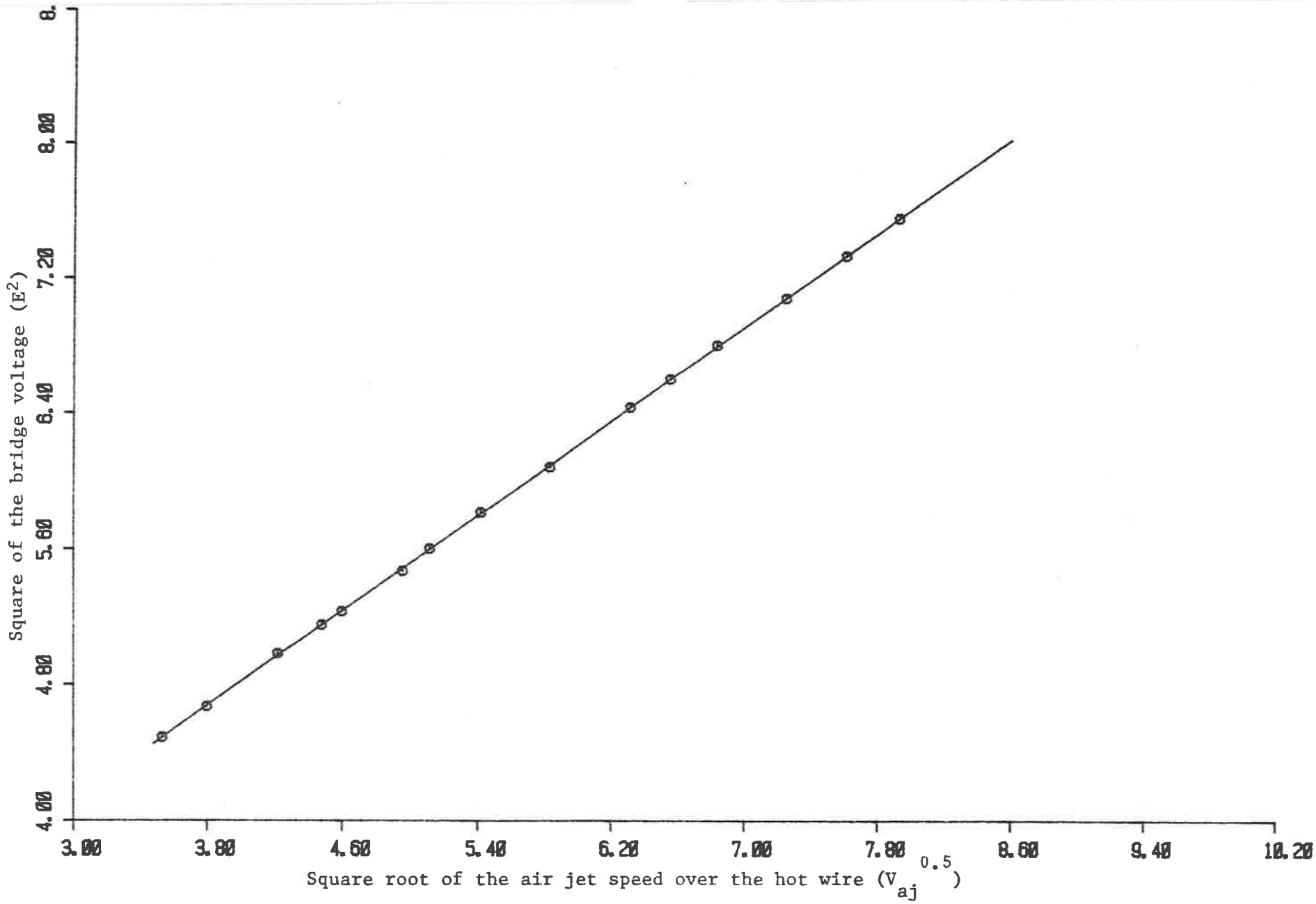


Figure 4.10 Calibration curve of the hot wire.

the air jet, and also to ensure that both were placed in the same location they were held in an especially designed mount using a micro-manipulator.

Calibration law for the hot wire is given by [49].

$$E = A + B\,V_{aj}^{n}$$

where E is the voltage across the hot wire

V_{aj} is the air jet speed and

A, B and n are constants.

Previous experimental results show that the value of n ranges between 0.45 and 0.5 [49]. For the present experiment the value of n equal to 0.5 was found to give the best least square fit line through the data points as shown in figure 4.10.

4.5.3 Measurement of Flow

The speed of the incident air stream was measured by placing the hot wire anemometer near the surface of the rotating saw as shown in figure 4.6. The hot wire probe stand was designed such that the movement of the hot wire probe could be adjusted both in the axial and radial directions of the saw. In addition, the hot wire probe was arranged to have a rotating motion without axial movement, so that the direction of the wire can be altered to measure flow speed at any angle in the plane of the saw blade. A protractor was fixed on the stand to determine the direction of the hot wire.

Measurements of the incident air flow were made at saw speeds of 1000, 1500, 2000, 2500 and 3000 rpm, with the hot wire located 1.5 mm from the face of the saw and at various radii between periphery and 10 mm below the bottom of the gullet.

At each observation point the hot wire was rotated until the flow speed was a maximum.

5. RESULTS AND DISCUSSION

Specific examples of experimental results which are representative of overall trends have been selected for inclusion in the body of the thesis.

5.1. Incident Air Flow

Measurements of the speed of the air flow entrained by the surface of the rotating saw indicated that the absolute air velocity is maximum at an angle of approximately 30 degrees to the tangent. The flow direction remained approximately the same for the saw speed range of 1000 to 3000 rpm. By neglecting the air speed relative to the saw in the axial direction [14], the velocity of the air stream relative to the saw can be determined from the velocity diagram shown in figure 5.1. From the plots in this figure it is evident that the relative air speed (V) varies linearly with the peripheral speed of the saw (U) irrespective of the distance from the saw axis. Hence the relative air stream speed is related to the peripheral speed of the saw by equation (5.1).

$$V = mU + n \quad \text{--------} \quad (5.1)$$

$U > 15\ m/s$

where V is relative air stream speed over the saw tooth (m/s)

U is the peripheral speed of the saw (m/s)

$m \simeq .7567$ and $n \simeq 1.425$

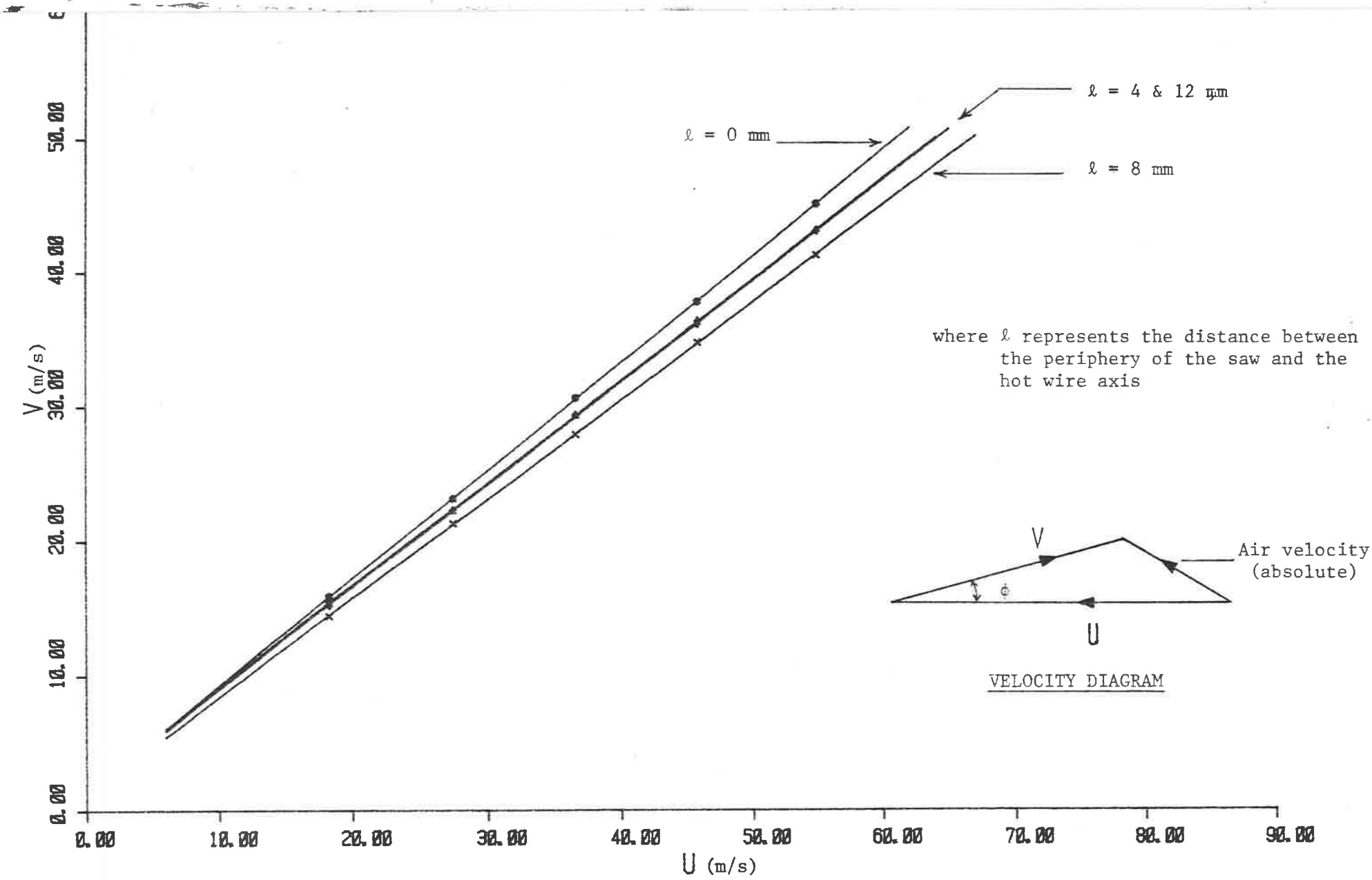


Figure 5.1 Relative air speed over the saw tooth (V) vs peripheral speed of the saw (U)

Plots shown in figure 5.2 indicate that the relative air speed varies non linearly over the tooth and reaches a minimum at a point approximately in the middle of the tooth height. However, from these plots it is evident that the relative speed of the air is within 81 percent of the peripheral speed of the saw. It can be seen from the figure that the relative air speed over the saw tooth decreases with increase in saw speed. However, the variation is of the order of one percent for a change in saw speed from 2000 to 3000 rpm.

In figure 5.3, the variation of the direction of the air stream relative to the saw blade is plotted against the radial position of the hot wire. It is evident that the relative air flow makes an angle of 2 to 9 degrees with the tangent and reaching maximum at a point where the minimum relative air speed was observed. From the figure it is also seen that the angle (ϕ) of the relative air flow increases with saw speed. The change in ϕ is only within a degree for the saw speed varying from 2000 to 3000 rpm. Thus the direction of relative air flow is in close agreement with Dugdale's [5] observation on the flow direction. Because these angles are small, negligible error is introduced by assuming that the air flow over the saw teeth is tangential to the blade.

5.2 Scaling Law

Sound power dependence on the peripheral speed of the saw was investigated for a series of gullet dimensions. Based on the dipole model of equation (15), the sound power values determined from the reverberation room measurements were normalised on tooth area (S) as shown in figure 5.4. The data shown in

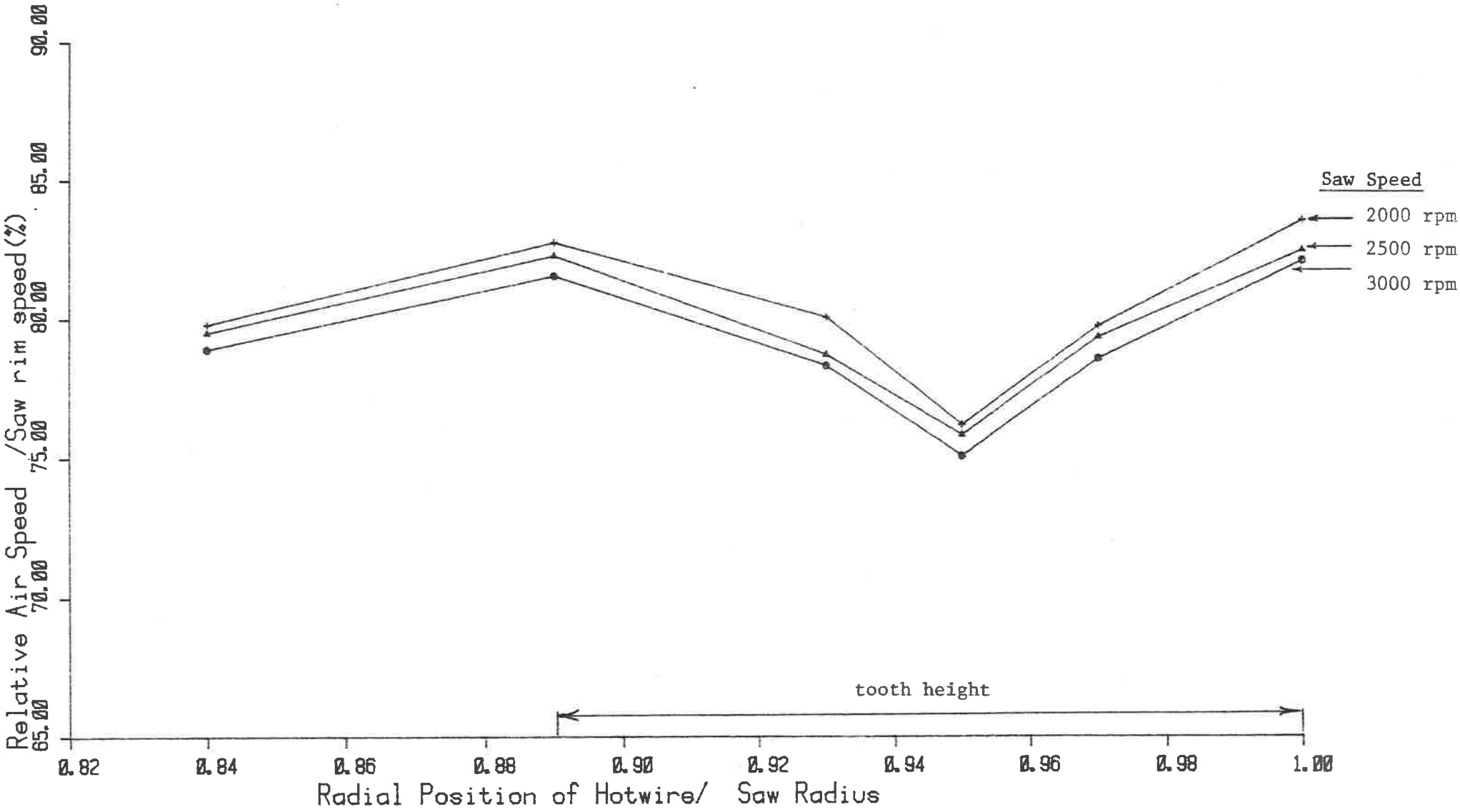


Figure 5.2 Relative air speed vs hot wire position

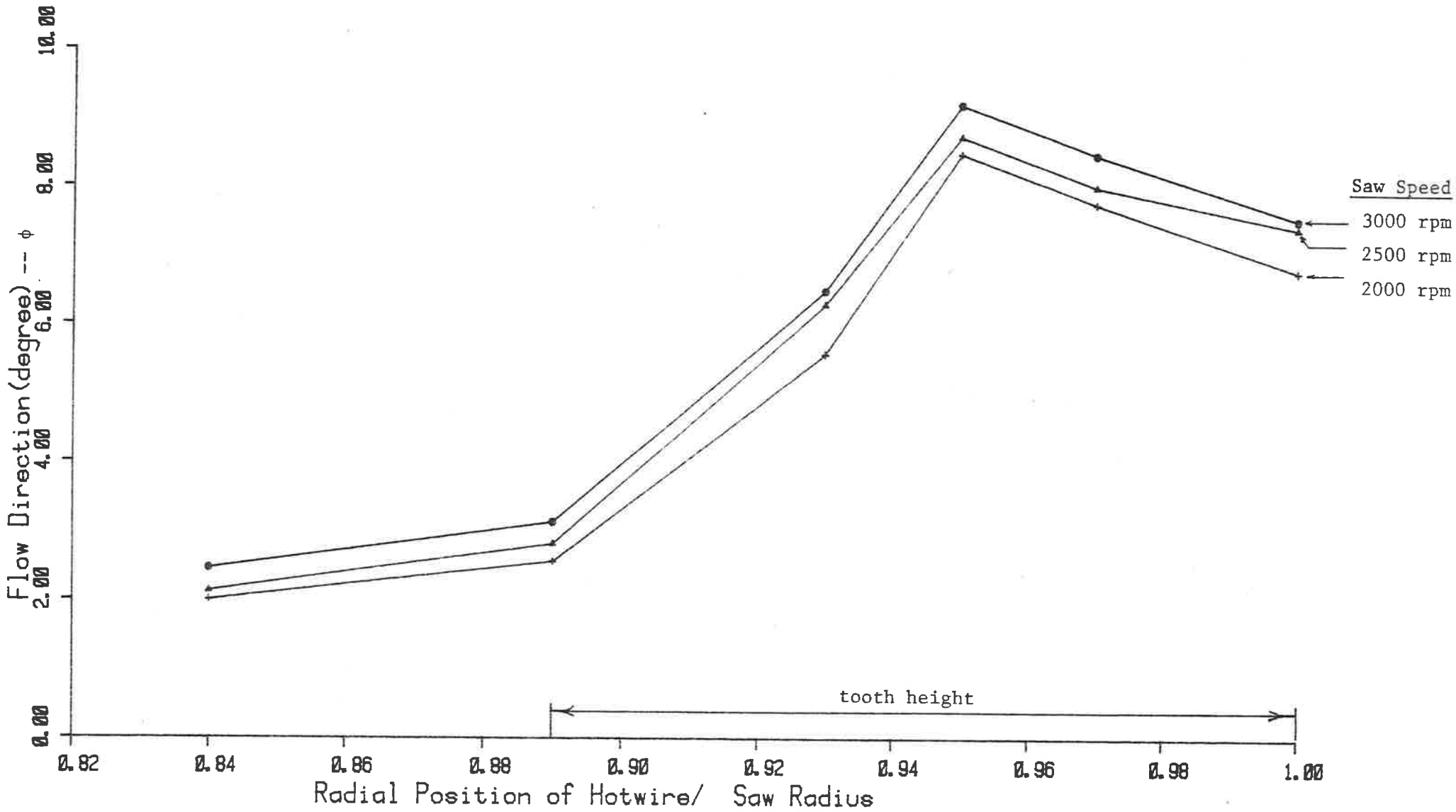


Figure 5.3 Direction of air flow relative to the saw vs radial position of the hot wire

this figure corresponds to a range of gullet width from 8 to 16 mm at a fixed gullet depth of 14 mm. By considering the same range of gullet width and a range of gullet depth of 8 to 16 mm, the radiated sound power has been found to be proportional to the tooth speed raised to the power of 5.25. This is less than the characteristic value of six for point dipole sound radiation. Previous investigators [1-3,18,20,34,35,37] have observed values within a range of 4.9 to 5.6.

5.3 Effect of Tooth Set to Noise Generation

Sound power levels determined from the reverberant room measurements were used to investigate the effect of tooth set. By normalising radiated sound power levels to tooth area (S) and tooth speed (U) as shown in figure 5.5, it can be seen that the effect of increasing the tooth set from 1.00 to 2.66 gives a maximum difference of approximately 3 dB in sound power levels. Unfortunately no trend could be found between the sound power and the tooth set. However, it is evident from this figure that the sound power level normalised on tooth area and tooth speed is approximately a constant and the effect of tooth set appears to be unimportant to the noise generation of an idling saw.

The observation that the tooth set is unimportant in the saw noise generation is not in agreement with the conclusion made by Zockel et all [17], who reported that doubling tooth set increases the sound pressure level by 15 dB(A) at the operator's position. This may be because the effect of tooth area and number of teeth was apparently neglected in their investigation. These parameters are contributing factors in the point dipole models [1,36] proposed for the aerodynamic noise radiated by the circular saws.

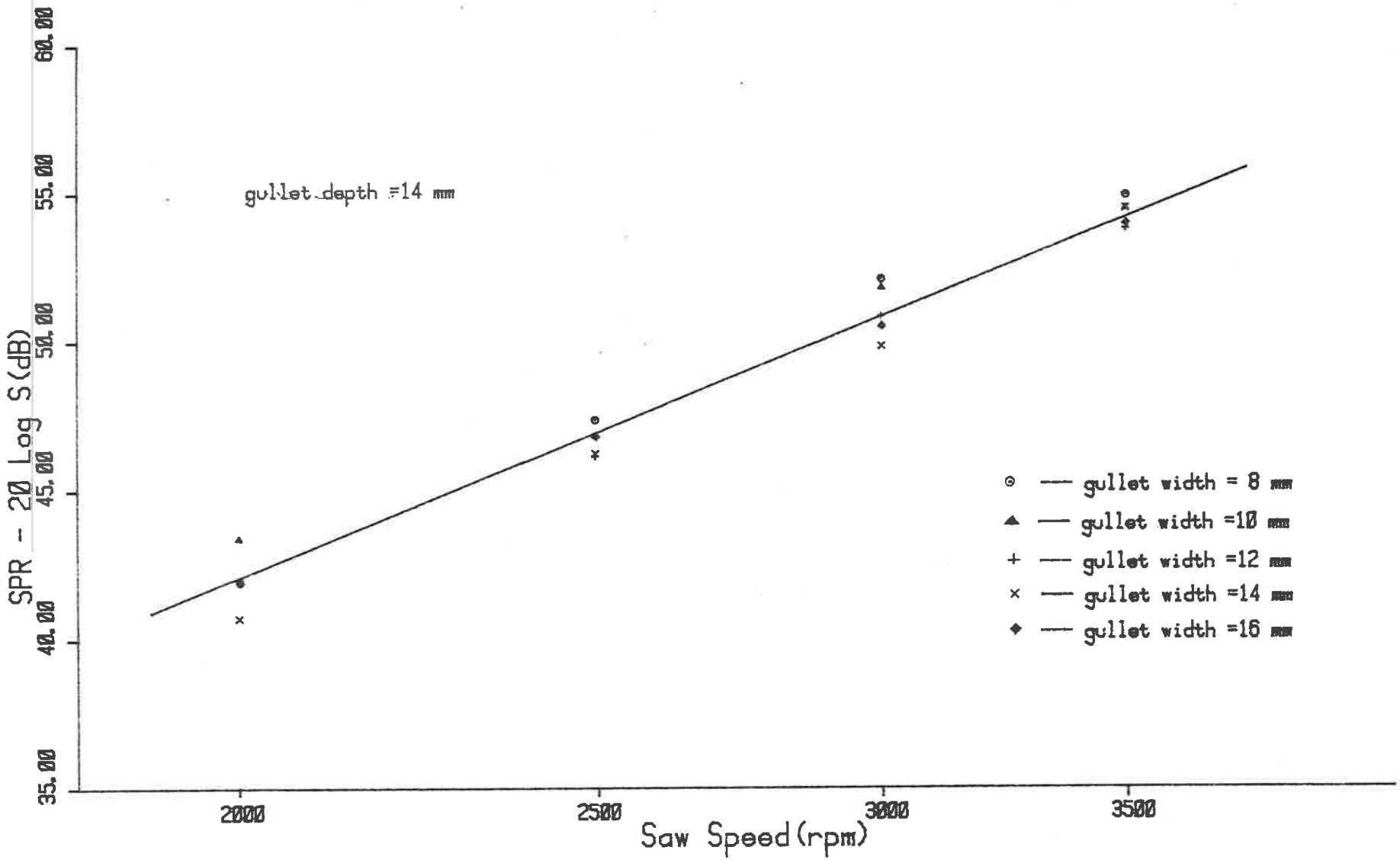


Figure 5.4 Overall sound power dependence on saw speed

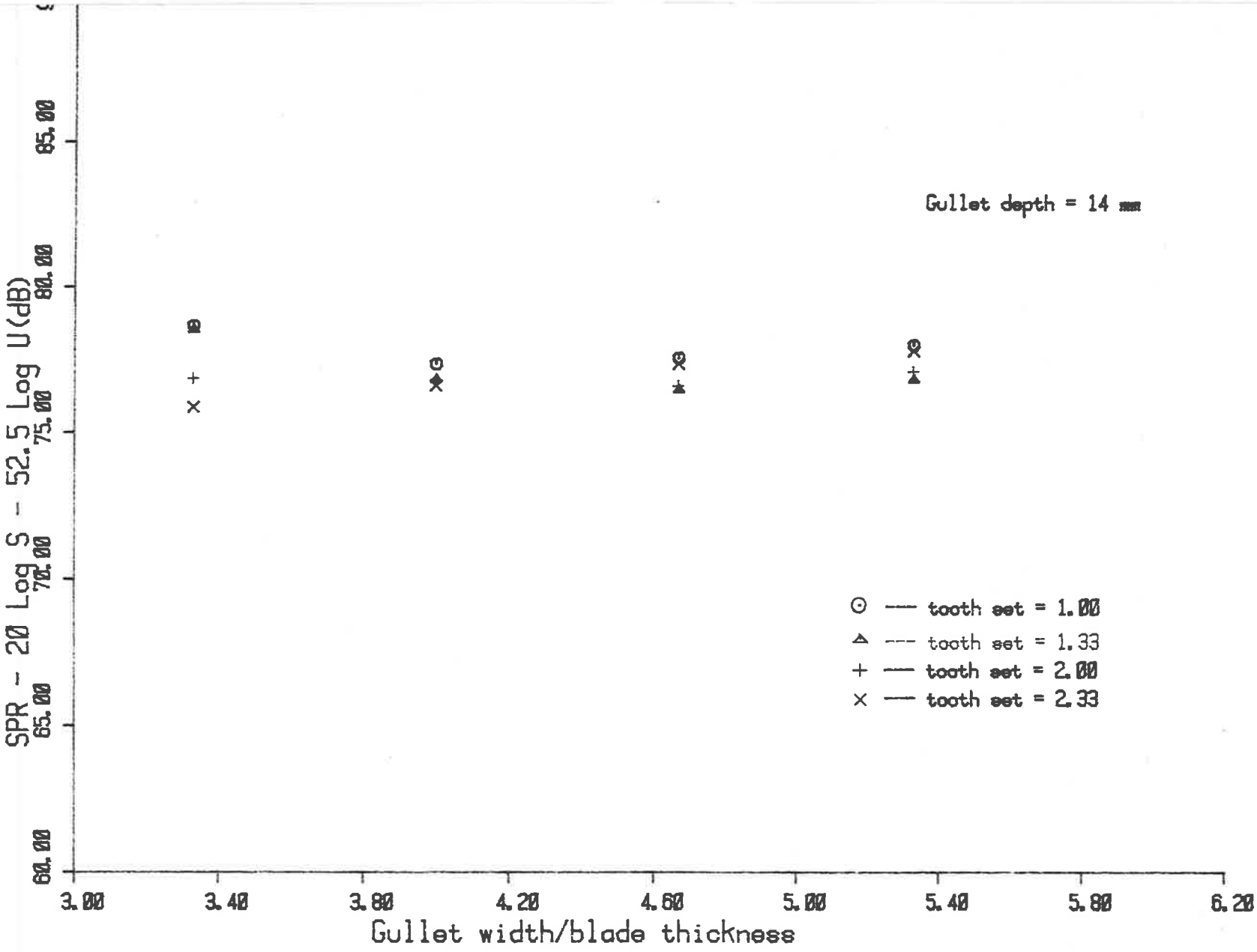


Figure 5.5 Effect of tooth set on noise radiation

Noise directivity patterns obtained for saw blades with increasing amounts of tooth set indicate an increase in the radiated noise levels in the region close to the plane of the disc (see figures A3.1 and A3.2 in the Appendix). It appears that an increase in tooth set strengthens the monopole sound sources, as the volume of the displaced air stream is related to the tooth volume [14]. However, for the range of tooth set from 1.00 to 2.66, maximum increase in sound pressure level was less than 3 dB.

5.4 Sound Spectra

The broad band noise which is characteristic of the saw noise [1-3,14,20,36] is evident from the sound spectra shown in figure 5.6. It is seen from this figure that the centre frequency of the dominant one third octave band increases with saw speed, and it can be shown that this frequency is approximately proportional to tooth passage frequency.

In figure 5.7 sound power levels normalised on tooth area and tooth speed are presented in terms of the non dimensional number (fh/U); where f is the third octave band centre frequency of the radiated noise from the rotating circular saw, h is the maximum tooth thickness across the saw blade and U is the saw tooth speed. This non dimensional number is referred to as the Strouhal number [50], when f represents the representative frequency of a broad band noise.

All of the superimposed sound spectra obtained at various saw speeds show a linearly rising sound power up to a value of 0.14 for the non-dimensional number (fh/U); beyond (fh/U) is equal to 0.18, the sound power is approximately a constant.

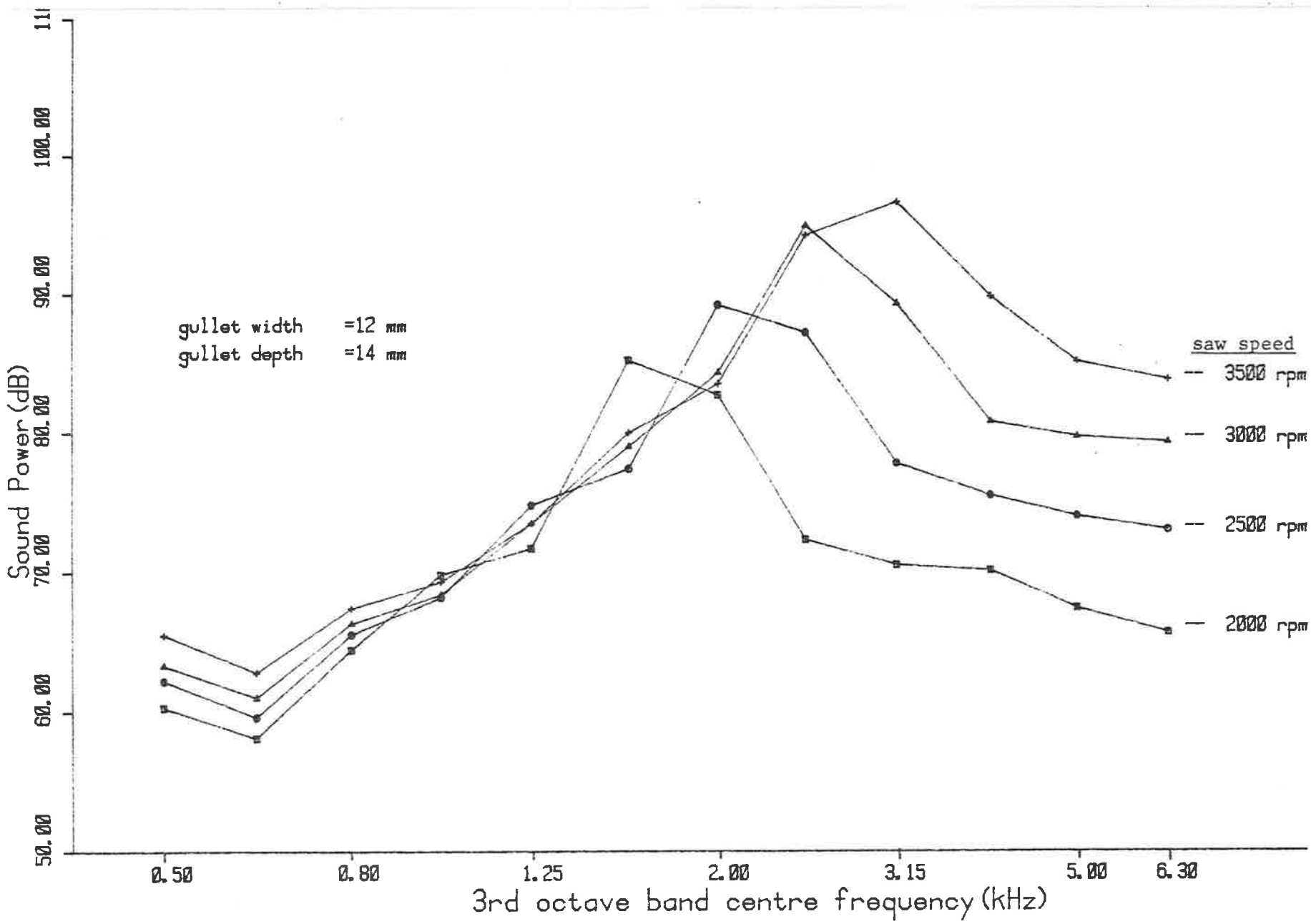


Figure 5.6 Sound spectra; data obtained from reverberation room measurements.

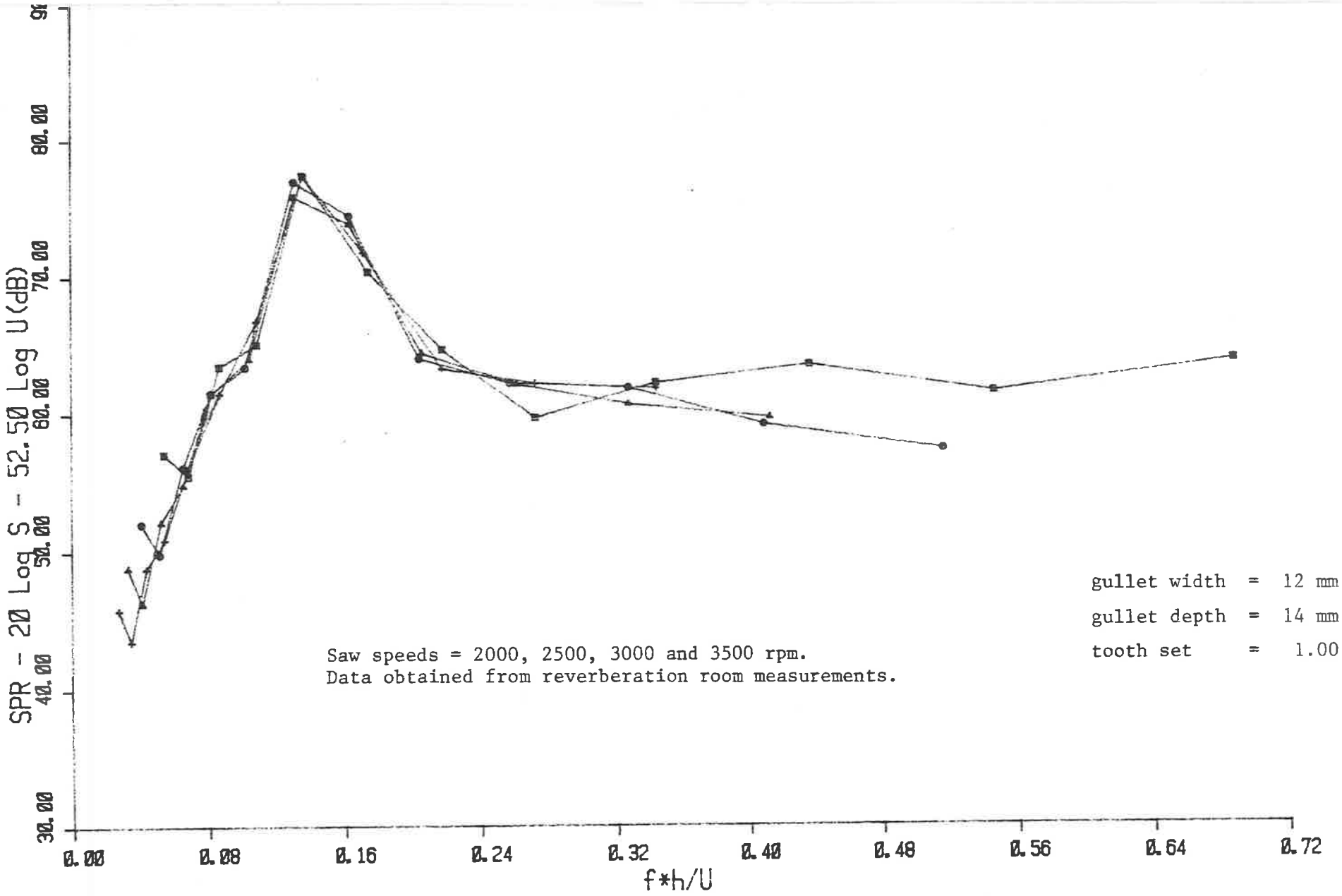


Figure 5.7 Sound power (one third octave bands) as a function of the non dimensional number (fh/U), for a saw running at various speeds.

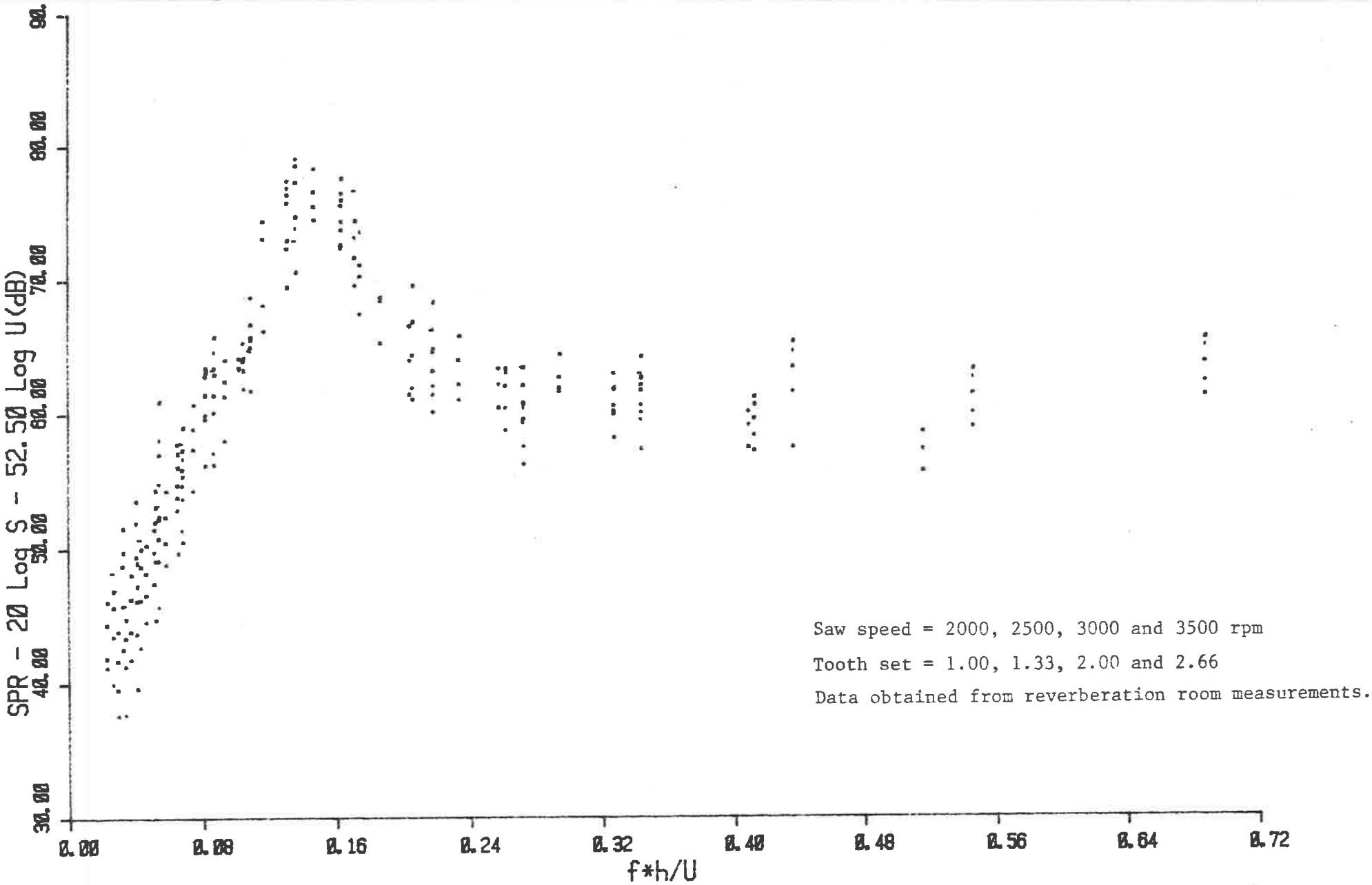


Figure 5.8 Sound power (one third octave bands) as a function of (fh/U); saw speed, tooth area and tooth set are variables.

The peak in the sound spectra corresponds to the Strohual number on the order of 0.14.

The data in figure 5.7 corresponds to the saw running at different speeds, with unity tooth set. Figure 5.8 shows collection of data points for saws with different tooth areas and tooth sets run at various speeds. Examination of figures 5.7 and 5.8 reveals that they represent nearly the same information irrespective of tooth speed, tooth area and tooth set.

Although the use of the Strouhal number is not new in saw noise research, the observation that the Strouhal number must be based on the tooth set to normalise the data for comparison is important. It is evident that the sound power normalised on tooth speed and tooth area is approximately a constant, which indicates that the saw noise is primarily controlled by tooth speed and tooth area.

It is to be noted that the results discussed in this Section correspond to saws which have a gullet width greater than twice the blade thickness.

5.5 Sound Pressure Level at the Operator's Position in Free Field.

In figure 5.9, the normalised sound pressure level is plotted against the gullet width to blade thickness ratio at two saw speeds (2000 and 3000 rpm). The data shown in this figure is obtained from the plots presented in figures A3.3 and A3.4 in the Appendix. It can be seen that the data can be approximated by two straight lines, one of which increases linearly up to gullet width (w) to blade thickness (b) ratio less than 2.0 and a second which is approximately a constant for ratios greater than 2.0. The latter result is as predicted by equation (13)

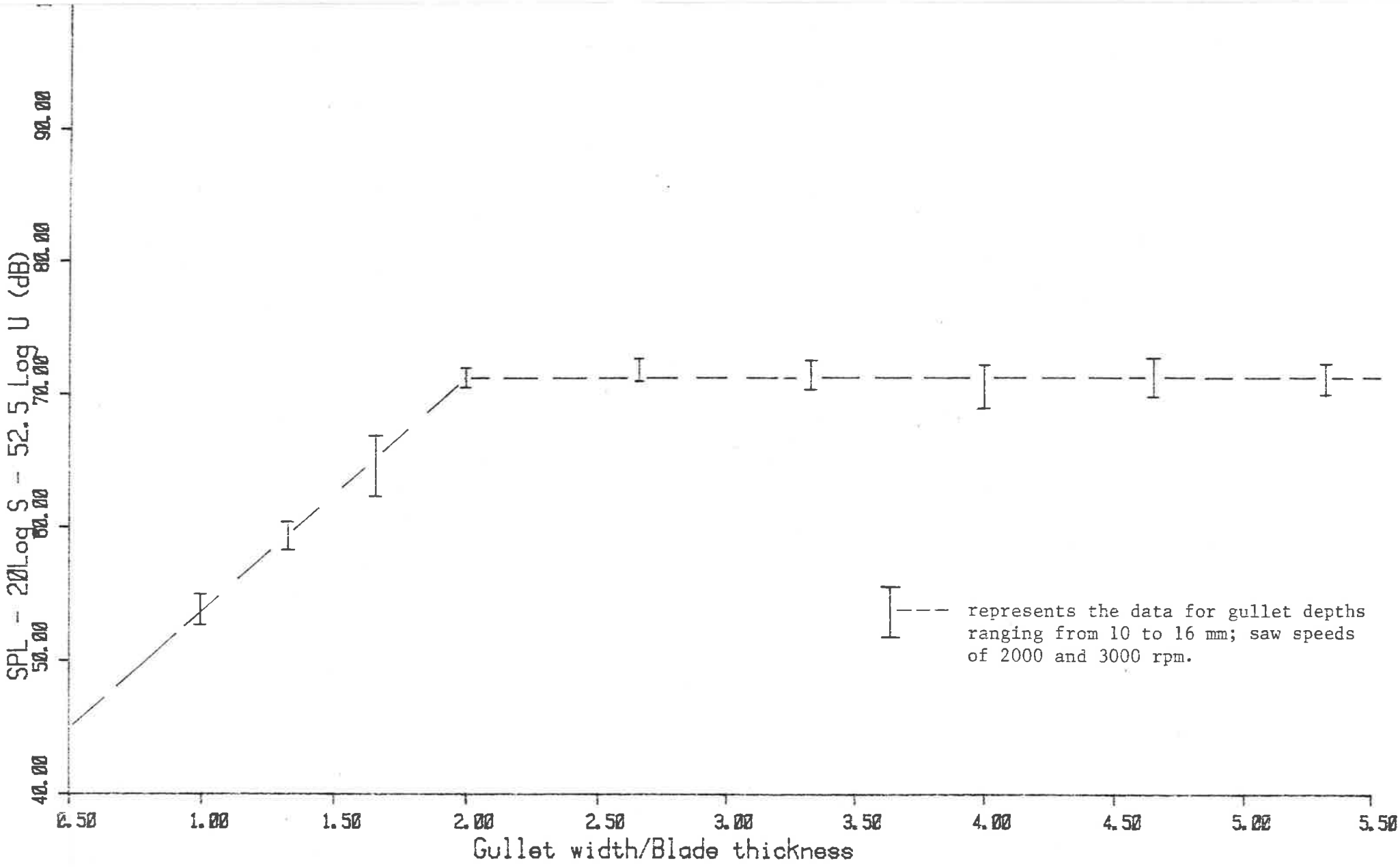


Figure 5.9 Variation of the normalised Sound Pressure Levels with gullet width to blade thickness ratio; the SPL values measured at the operator's position in the anechoic room

and implies that the air flow is fully developed and each tooth produces point dipole sound source as suggested by Curle's [21] theory for turbulent flow on a compact surface. The behaviour of the sound pressure level when (w/b) less than 2.0 is not fully understood. One possible explanation is that the flow is not fully developed when $w < 2b$.

From the results presented in figure 5.9, the following empirical equations are suggested to predict the sound pressure levels at the operator's position in free field.

when $w/b \geqq 2.0$,

$$SPL = 52.5 \log U + 20 \log S + 71.0 \quad \text{------} \quad (5.2)$$

when $w/b \leqq 2.0$,

$$SPL = 52.5 \log U + 20 \log S + 17.5(w/b) + 36.0 \quad \text{---} (5.3)$$

Although these values are more appropriate for the model saw blades used in the current experiments, it is expected that these results will serve as guide lines for the noise level produced by industrial saws, since tooth geometry is not a significant factor for saw noise radiation.

Figure 5.10 demonstrates the effect of gullet width on saw noise. The data in this figure indicates that doubling the gullet width results in an increase in sound pressure level of approximately 15 dB when gullet width is less than twice the saw blade thickness.

5.6 Noise Source

The noise directivity information obtained for various saw blades was generally similar irrespective of the gullet dimensions as long as the tooth set is held constant. Figure

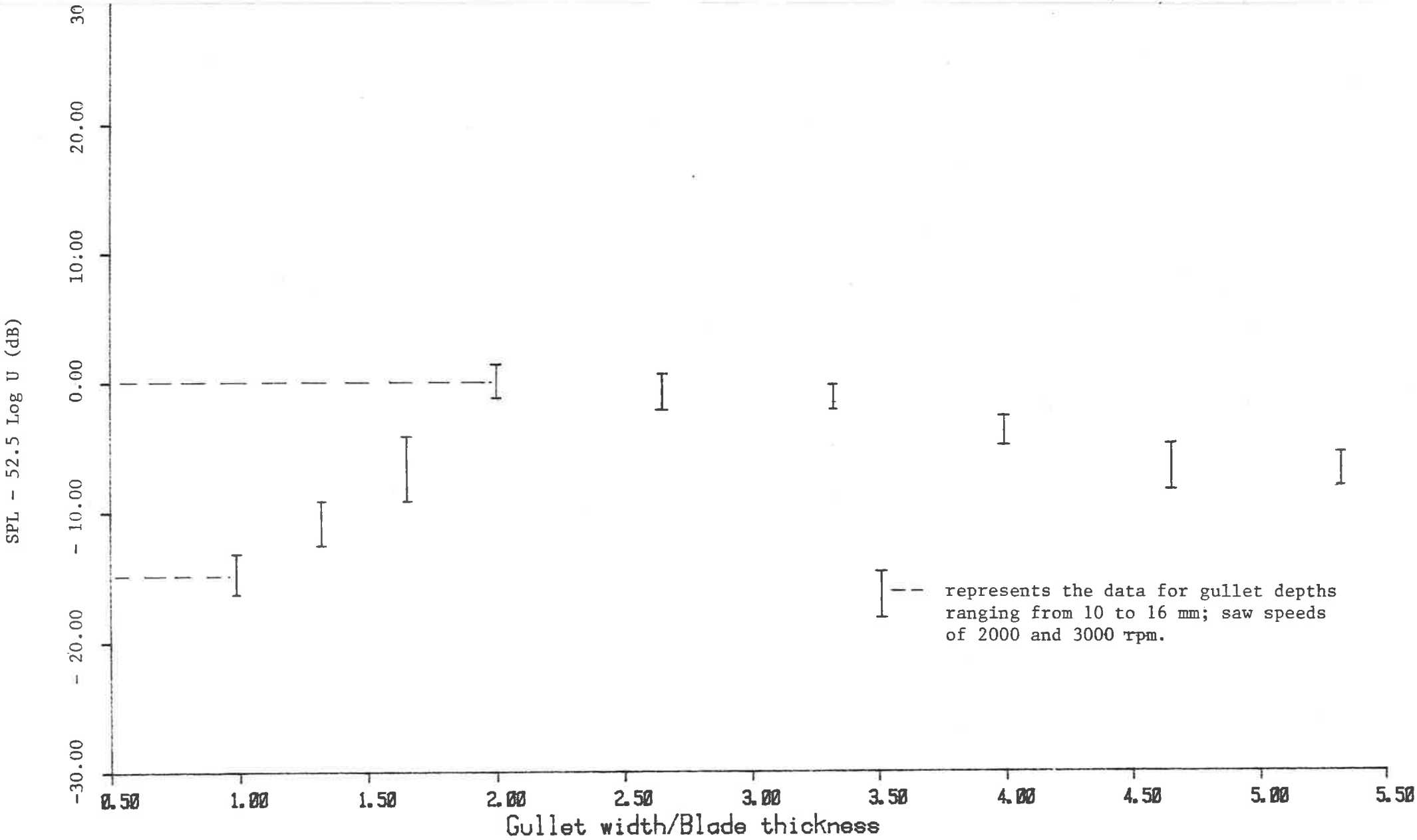


Figure 5.10 Effect of gullet width to noise generation; SPL values measured at the operator's position in the anechoic room

5.11 shows the noise directivity data produced by a saw with no tips running at speeds of 2000 and 3000 rpm together with the directivity values (SPL) computed from equation (13). In this figure, normalised sound pressure levels are plotted against angles (theta) as measured from the plane of the saw blade. It is to be noted that both experimentally observed and predicted sound pressure levels occured at points located at a radius of 1.2 m from the centre of the saw, and also on the X1 - X2 plane (see figure 3.3)

To provide a comparison of both experimentally observed and predicted sound pressure levels, a value of α in equation (13) was selected such that the data points (both predicted and experimentally observed) approximately coincide on the saw axis as shown in figure 5.11. For the data in the figure, α was found to be equal to 0.045 irrespective of saw speed.

It is evident from this figure that there is a considerable deviation between experimental and predicted sound pressure levels, which implies that the point dipole model does not adequately describe the directivity pattern of the saw noise. Nevertheless, the contribution of tooth area suggests that the source of the aerodynamic noise generated by circular saws is basically of a dipole nature as proposed by most investigators [1-3,14,18,36].

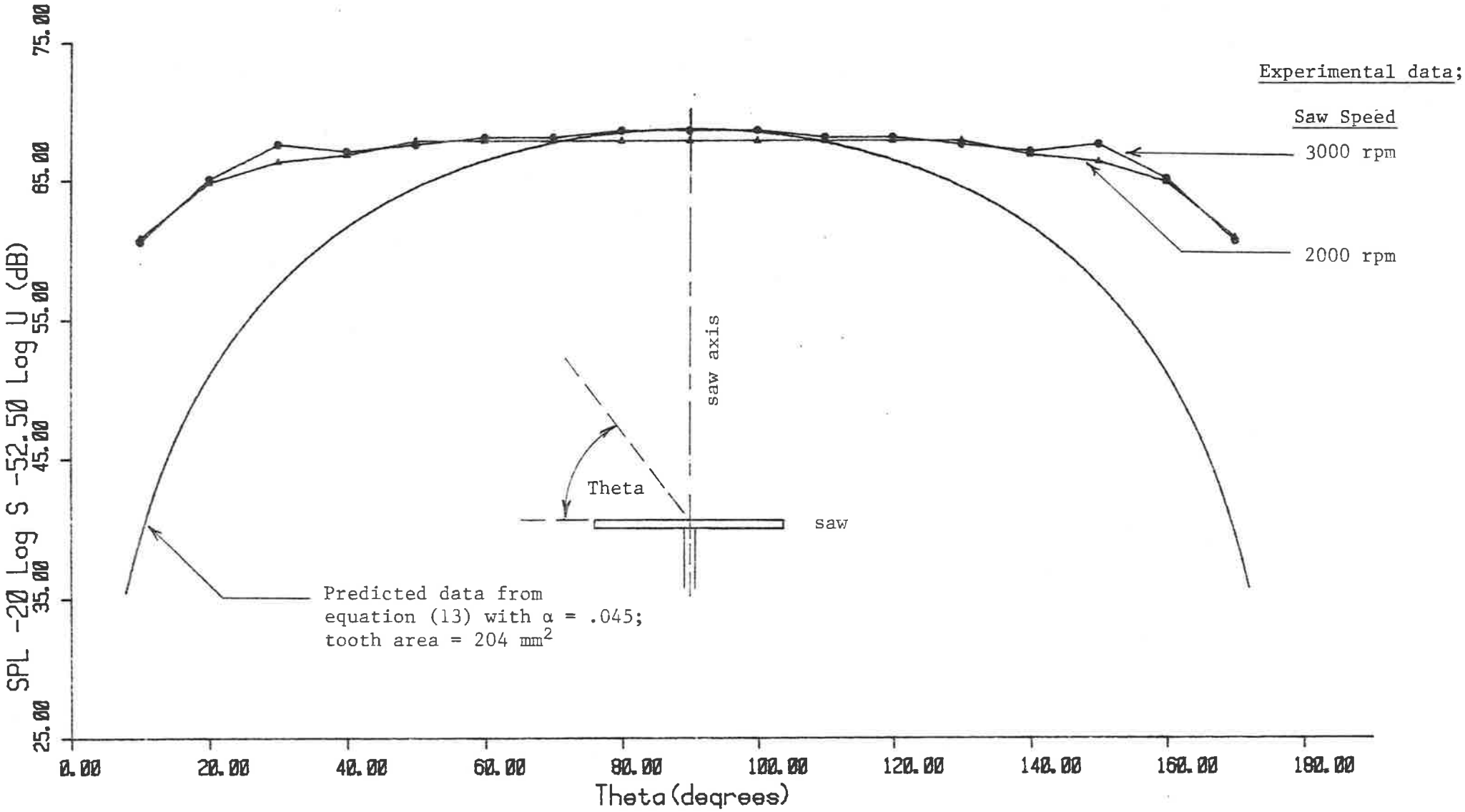


Figure 5.11 Noise directivity pattern

6. CONCLUSIONS AND RECOMMENDATIONS

The results obtained indicate that the radiated sound power is dependent on the peripheral speed of the saw raised to the power of 5.25. This is consistant with the generally accepted theory that the peripheral speed of the saw is the prime contributor to the aerodynamic noise generation.

Tooth area is a parameter which, while less important than the peripheral speed of the saw blade, can contribute significantly to saw noise generation. Results of the empirical investigation indicate that the radiated sound power is proportional to the square of the tooth area when gullet width is greater than twice the saw blade thickness. When gullet width is less than twice the saw blade thickness, gullet width controls the saw noise generation; doubling gullet width results in an increase in noise levels of approximately 15 dB. Thus the saw noise can be reduced substantially by keeping either the tooth area or the gullet width to a practical minimum, depending on the specific working requirements of gullet size.

The effect of carbide tips (the tooth set) has not been found to be significant to saw noise generation. Within a range of tooth set from 1.00 to 2.66, the variation in radiated sound power is less than 3 dB.

The aerodynamic noise generated by the circular saw is characterized by a Strouhal number of approximately 0.14, based on the maximum tooth thickness across the saw blade.

The relative speed of the incident air stream over the saw tooth is found to be proportional to the tooth speed and the relative air speed is approximately 81 percent of the tooth speed at an angle of 2 to 9 degrees to the tangent.

The point dipole theory which applies to the aerodynamic noise generated by the circular saw suggests that the radiated sound power is proportional to the square of the tooth area. Thus the experimental observation on the tooth area contribution to saw noise is supportive of the point dipole model when the gullet width is greater than twice the blade thickness. However, the point dipole theory fails to satisfactorily explain the observed noise directivity pattern. This leads to the conclusion that the source of the saw noise is only partially identified.

It is recommended that the point dipole model for the saw noise be further investigated by direct measurement of the rate of change of the fluctuating lift force acting on each saw tooth. If the results are not in agreement with the point dipole model then the degree to which dipoles acting on individual saw teeth are correlated should be investigated.

APPENDIX I

TERMINOLOGY

Operator's Position:

The operator's position is defined as the point (P) which is one meter away from the centre of the circular saw and at an angle of 45 degrees to the axis of the saw as shown in figure A1(a); the point of observation (P) and the axis of the saw are at the same height.

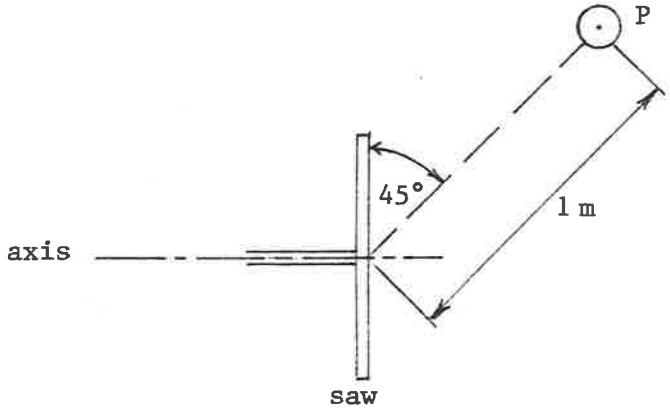
Gullet and Tooth:

The gap space between two adjacent saw teeth is called gullet. Parameters such as gullet width (w) and gullet depth (d) are shown in figure A1(b). Gullet area is shown in the figure by cross - hatching in the gullet space; tooth area is represented by S. Note that tooth height is equal to gullet depth.

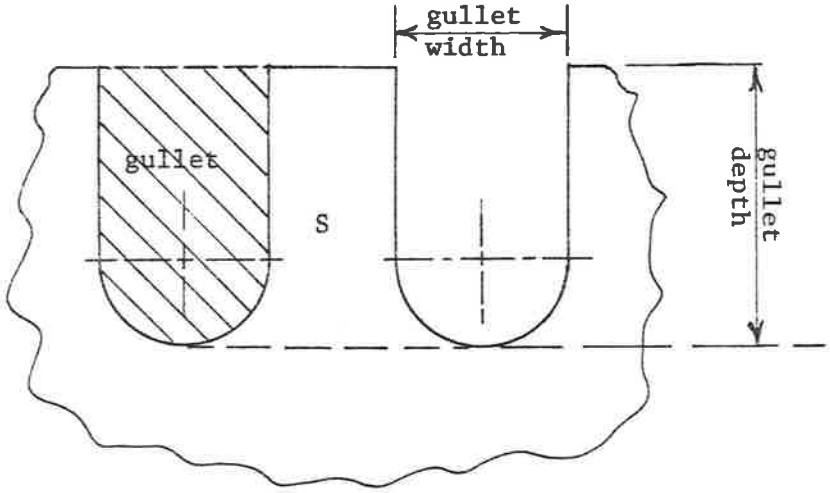
Tooth Set:

Tooth set is defined as the ratio of maximum tooth thickness (h) across the saw blade to the saw blade thickness (b). In the case of (mild steel) tipped saw tooth shown in figure A1(c),

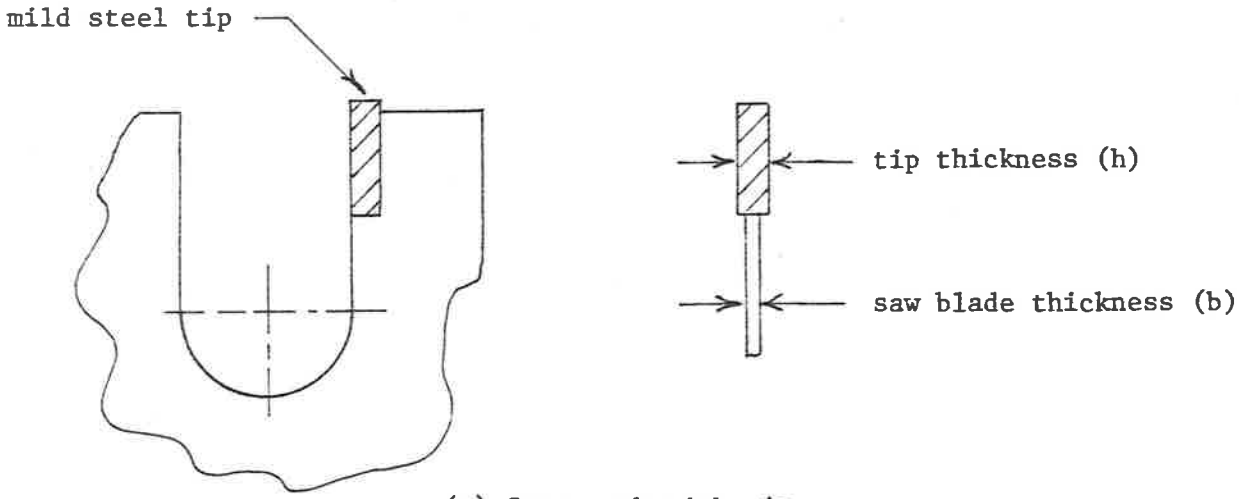
$$\text{tooth set} = (h/b)$$



(a) Operator's position (P)



(b) Saw tooth and gullet



(c) Saw tooth with tip

Figure A1 Diagrams for terminology

APPENDIX II

A BRIEF REVIEW ON EDGE NOISE

Powell [40] examined the turbulent flow over a flat plate and predicted that the resulting sound power is proportional to the characteristic flow speed raised to a power between 4 and 5. He postulated that the source of the noise is a distribution of dipoles located near the edge of the plate and that the noise from elsewhere on the surface is negligible. This was based on his contention that the pressure dipoles on the surface have the cancellation effect. Powell's treatment for the prediction of the noise from the edge, called edge noise, was based on dimensional and similarity concepts.

Ffowcs Williams and Hall [30] analysed the effect of a sharp edged half plane exposed to a turbulent flow, and suggested that the edge behaves like a scatterer and the resulting sound power is proportional to the characteristic flow velocity raised to the power of 5. They concluded that the noise is produced by acoustically compact turbulent eddies located at a distance much less than the representative wavelength from the edge of the half plane.

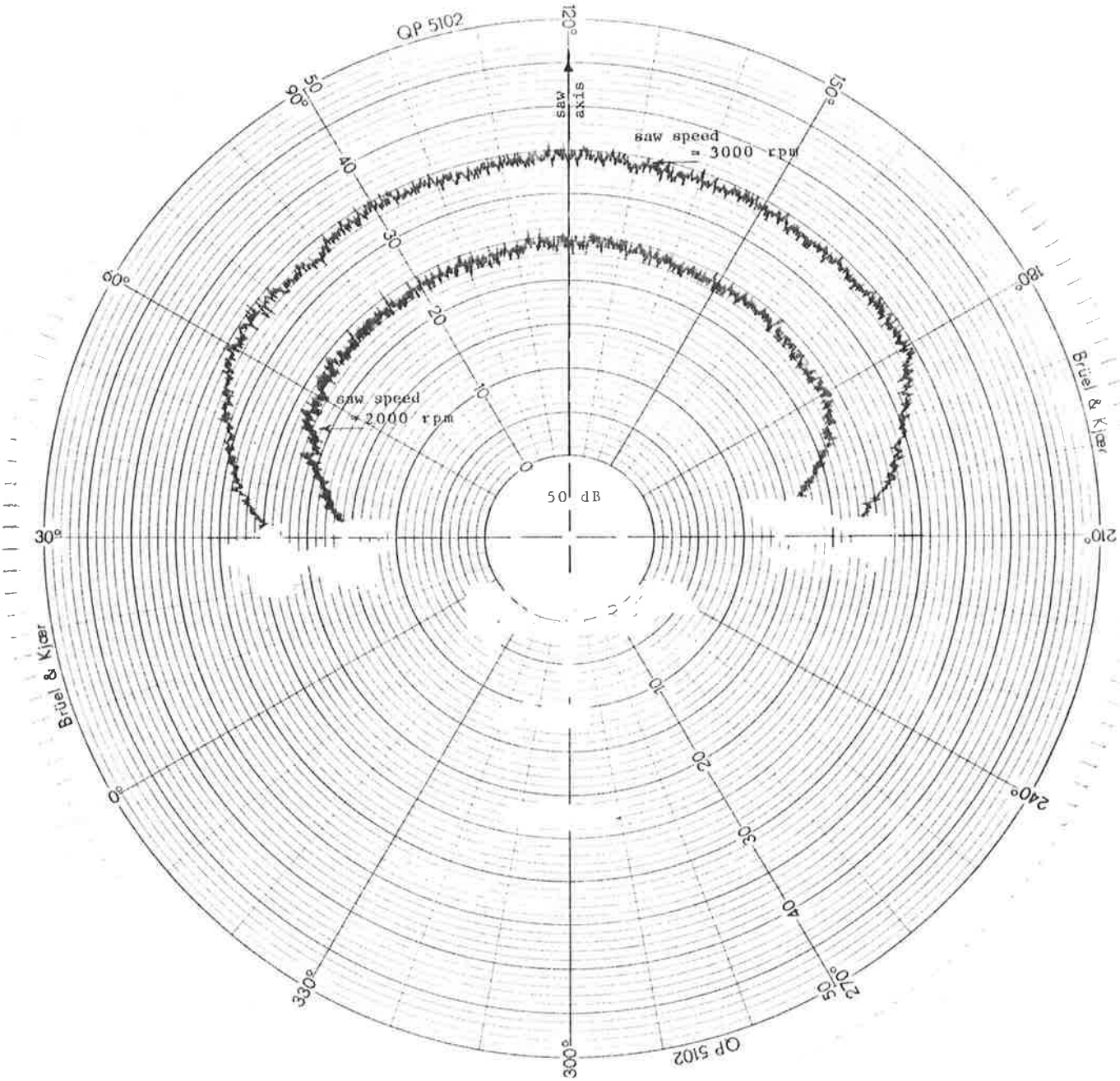
Based on Powell's edge noise concept, Hayden, Fox and Chanaud [42] proposed a model for the turbulent flow over finite surfaces, which suggests that the noise produced at the trailing edge of an acoustically non-compact surface is due to the dipole distribution at the edge or downstream of the edge at a very small distance compared to the characteristic wavelength.

Hayden's [41] previous prediction indicated that the acoustic power radiated from a trailing edge to be proportional to the characteristic flow speed raised to the power of six for flows over rigid surfaces. Later, Hayden et all [42] found that the scaling law for the acoustic power dependence on the characteristic flow speed was equal to 5 for semi-infinite surfaces as a low Mach number limit, and the scaling value was observed to decrease with increase in Mach number. For finite surfaces, the scaling law was expressed as equal to 6 as a low Mach number limit and the scaling value was found to decrease with increase in Mach number. Howe [31] reviewed the work done by Hayden et all [42] and showed that the acoustic power is dependent on the characteristic flow speed to the fifth power. Hayden's work [41] showed that the dipole concentrated near the edge of the surface, which was called an edge dipole, produces a noise directivity pattern in the form of a cardioid normal to the surface and the trailing edge.

In summary, a number of different approaches have been adopted to analyse the edge noise problem. Despite the different approaches however, as shown by Howe [31], they all arrive at the same acoustic power dependence on flow speed raised to the fifth power and the same noise directivity pattern as observed by Hayden [41].

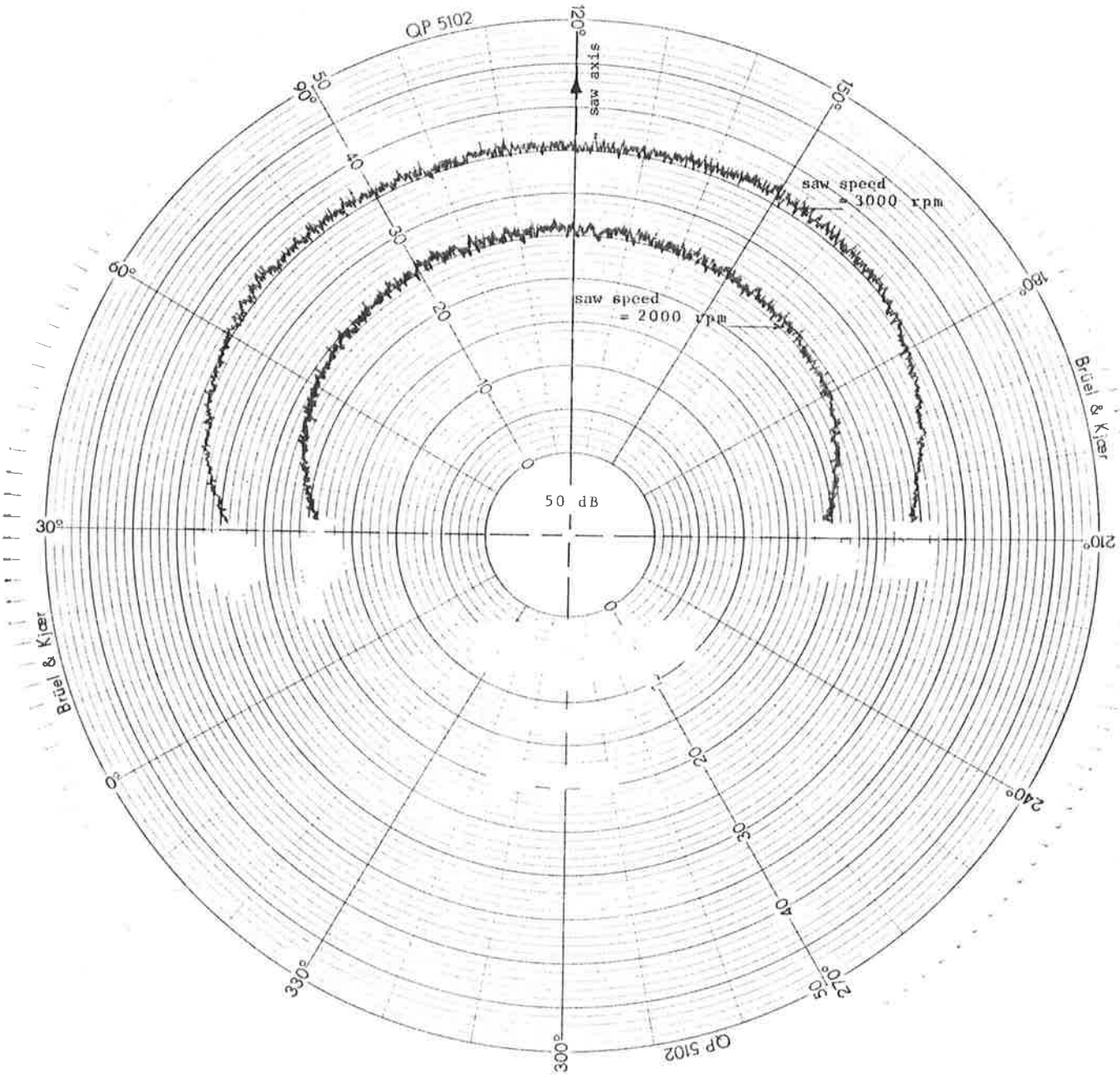
In most cases, models are limited to flows over semi-infinite rigid flat plates. None of those models could lead to accurate predictions in relation to acoustic power dependence on flow speed or noise directivity pattern for finite surfaces such as the surface of the circular saw where the representative acoustic wavelength is comparable to the dimensions of the body of the blade. For example, in the current study the wavelength

(corresponding to the centre frequency of the dominant one third octave band) was found to be approximately 135 mm at a saw speed of 3000 rpm; which is comparable to the radius of the saw (175 mm). However, the deviation in the scaling value for the acoustic power dependence on tooth speed (from the expected value of six for a point dipole sound source) may be attributed to the trailing edge noise.



gullet width = 16 mm
gullet depth = 14 mm
tooth set = 1.33

Figure A3.1 Noise directivity pattern of a saw with tooth set = 1.33.



gullet width = 16 mm
gullet depth = 14 mm
tooth set = 2.66

Figure A3.2 Noise directivity pattern of a saw with tooth set = 2.66.

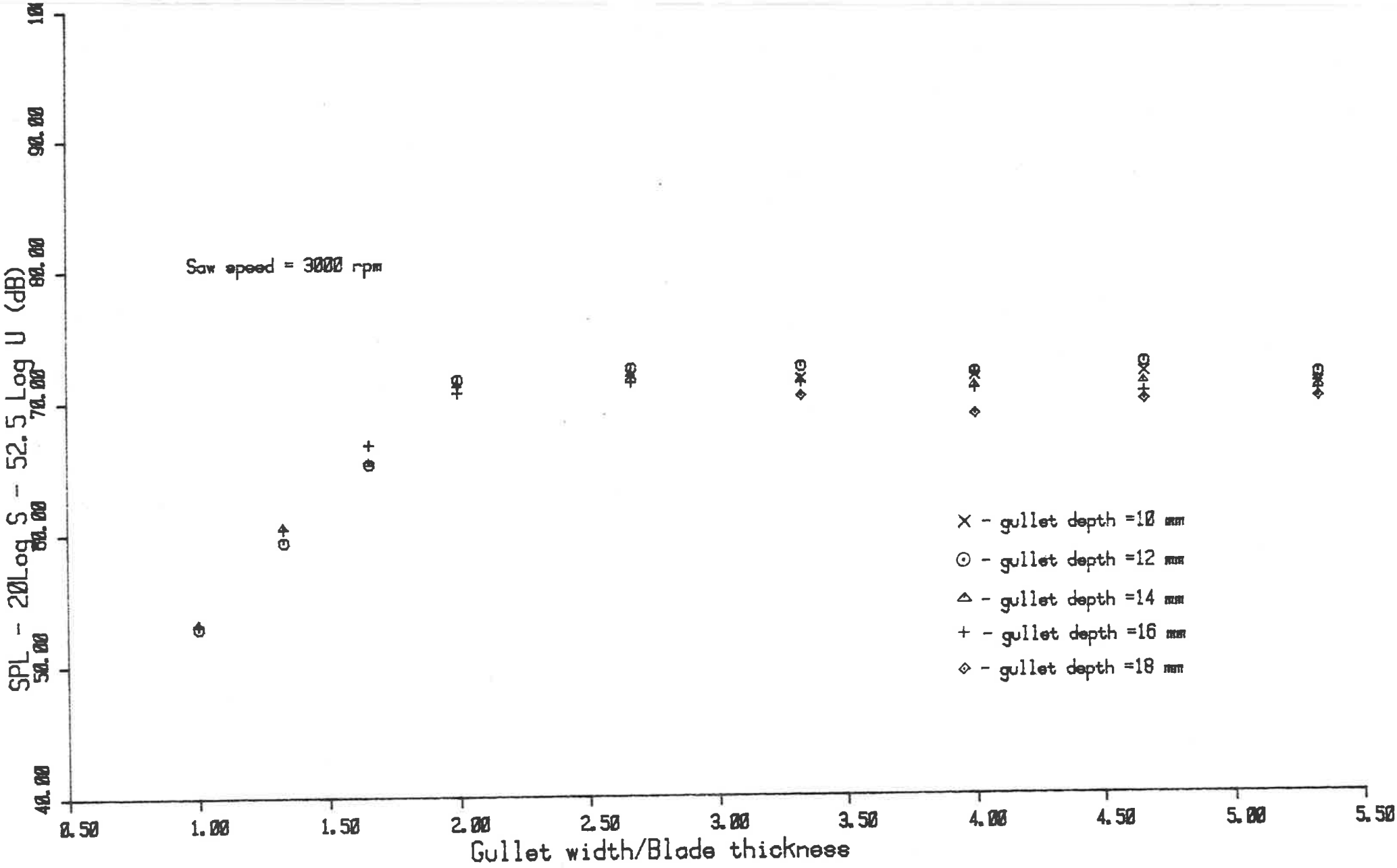


Figure A3.3 Sound pressure levels measured at the operator's position in the anechoic room

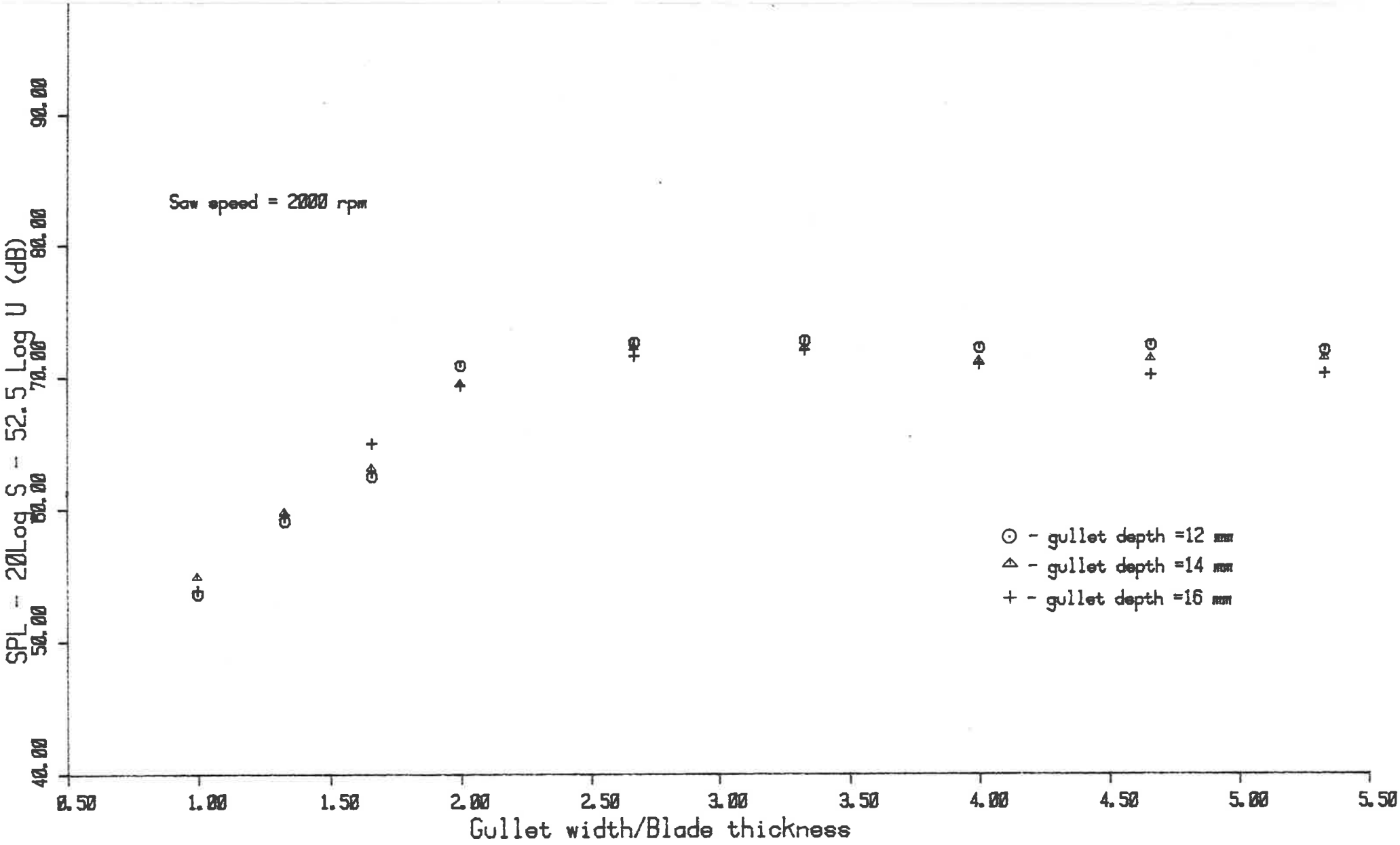


Figure A3.4 Sound pressure levels measured at the operator's position in the anechoic room

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