THE UNIVERSITY OF ASTON IN BIRMINGHAM

REDUCTION OF NOISE LEVELS IN VACUUM CLEANERS

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Thesis Summary

The work described in this thesis is directed towards the reduction of noise levels in the Hoover Turbopower upright vacuum cleaner. The experimental work embodies a study of such factors as the application of noise source identification techniques, investigation of the noise generating principles for each major source and evaluation of the noise reducing treatments. It was found that the design of the vacuum cleaner had not been optimised from the standpoint of noise emission. Important factors such as noise "windows", isolation of vibration at the source, panel rattle, resonances and critical speeds had not been considered. Therefore, a number of experimentally validated treatments are proposed. Their noise reduction benefit together with material and tooling costs are presented. The solutions to the noise problems were evaluated on a standard Turbopower and the sound power level of the cleaner was reduced from 87.5 dB(A) to 80.4 dB(A) at a cost of 93.6 pence per cleaner.

The designers' lack of experience in noise reduction was identified as one of the factors for the low priority given to noise during design of the cleaner. Consequently, the fundamentals of acoustics, principles of noise prediction and absorption and guidelines for good acoustical design were collated into a Handbook and circulated at Hoover plc.

Mechanical variations during production of the motor and the cleaner were found to be important. These caused a vast spread in the noise levels of the cleaners. Subsequently, the manufacturing processes were briefly studied to identify their source and recommendations for improvement are made.

Noise of a product is quality related and a high level of noise is considered to be a bad feature. This project suggested that the noise level be used constructively both as a test on the production line to identify cleaners above a certain noise level and also to promote the product by "designing" the characteristics of the sound so that the appliance is pleasant to the user. This project showed that good noise control principles should be implemented early in the design stage.

As yet there are no mandatory noise limits or noise-labelling requirements for household appliances. However, the literature suggests that noise-labelling is likely in the near future and the requirement will be to display the A-weighted sound power level. However, the "noys" scale of perceived noisiness was found more appropriate to the rating of appliance noise both as it is linear and therefore, a sound level that seems twice as loud is twice the value in noys and also takes into consideration the presence of pure tones, which even in the absence of a high noise level can lead to annoyance.

Key Words

NOISE CONTROL - NOISE REDUCTION - VIBRATION ISOLATION - VACUUM CLEANER NOISE-LABELLING

To my family

Acknowledgements

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CHAPTER ONE: PROJECT PROPOSAL AND THE RESEARCH ENVIRONMENT

Contents

1.1 Introduction

This opening chapter provides an introduction to the aims, objectives and the nature of the research project. The ethos of the Interdisciplinary Higher Degrees Scheme and its approach to research is described. Reference is also made to the role of the supervisory team and the real world environment in which this research was conducted.

The sponsoring organisation for this project was Hoover plc based at Perivale, Middlesex. A brief history of Hoover plc and its parent company in the USA is given. The research project concerns reduction in the noise emitted by a particular range of domestic vacuum cleaners: the Hoover Turbopower. The objective and subjective aims of this project are outlined together with its role within the sponsoring organisation. Finally an outline of the thesis structure is presented.

1.2 The Research Objectives

Hoover plc's requisites of this project can be judged from the brief project proposal which was submitted to IHD dated 6th July 1983;

"Object:

> To examine and identify noise centres of upright and cylinder vacuum cleaners with the object of reducing the noise levels. This noise reduction should be achieved without reducing the cleaning effectiveness of the cleaner. As such the interaction between noise and performance needs to be studied and understood. It is also an objective of this project to reduce energy requirements, particularly since excess noise represents excess energy.
>
> In order to achieve the objective, an understanding of noise measurement standards and legislation will be needed, as well as a commercial awareness of the likely cost premiums (or

not?) to be borne by noise/energy efficient products and how these relate to the market place."

Based on the above brief, this project embraces all the features of an Interdisciplinary Higher Degree, which will be discussed later. The main objective from the company's standpoint was reduction in the noise levels of the selected vacuum cleaners without loss of performance. In addition and probably of more significance was the understanding of noise generating phenomena and their influence upon cleaner performance.

The circumstances of this project were such that the goals were well defined at an early stage. However, two major changes in direction were experienced. Hoover plc suggested that the emphasis be shifted from "perceived noise" to reduction of the overall cleaner noise, and guidance on designing for lower noise. This change took account of the prospective "noise labelling" requirement. An EEC Directive which concerns noise labelling on household appliances has been discussed for several years and may be introduced in the near future. Subsequently, the market research survey to determine consumer opinion on aspects of noise, its relation to performance and efficiency, and customers' willingness to pay extra for a quiet cleaner was dropped. In exchange, the supervisory team agreed that a handbook of noise control guidelines should be produced as this was more beneficial. It was hoped this handbook would become a designers guide.

The research objectives can be summarised as follows:

1. To identify the major noise sources in a vacuum cleaner
2. To understand and explain noise generation principles
3. To hypothesize noise reduction methods,
4. To test hypotheses and to obtain noise reduction without loss in performance
5. To produce a handbook of noise control guidelines.

1.3 The Collaborating Organisation

The name "Hoover" is synonymous with vacuum cleaners. The fact that this name is sometimes used wrongly to imply cleaning the floor covering is of serious concern at Hoover plc. The company wishes to maintain its identity as a manufacturer and not be known as a "product". Hence the company protects its trademark by challenging guilty parties for misuse. For instance, a Penguin dictionary implied that Hoover was an appliance for cleaning carpets (or floor covering). Subsequent to a challenge from Hoover plc that entry was corrected. The entry in the Shorter Oxford English dictionary (1959) is "A vacuum cleaner manufactured by Hoover Ltd. Hence Hoover (verb transitive) to clean with a hoover (or, by extension, any vacuum cleaner)".

1.3.1 A brief history of the Hoover Company

The story of Hoover the family company, began in 1908 in North Canton (formerly Berlin), Ohio, USA. The company was a monument to the foresight of Mr W H "Boss" Hoover and his son. They were successful manufacturers of harness and leather goods but Hoover foresaw the decline of horse and carriage age and the dawn of the motor car era. Subsequently, Hoover was looking at other enterprises in which to diversify. In 1907 a local inventor, who was a relative of the Hoovers, brought them a model of his improved "electric suction sweeper". Electric suction sweepers had been known for a decade at that time, and this improved version was a crude machine made of tin and wood complete with a broom handle and fitted with revolving brushes. After further improving this electric sweeper Hoover decided to manufacture it. Afterwards, Hoover travelled the American mid-West selling the idea of cleaning by a machine.

With the growing popularity of this Hoover sweeper, the entire business was turned over to its manufacture. Hoover expanded by opening a factory in Canada in 1918 and the first Hoover cleaners which were sold in

Britain by Selfridges were shipped from the Canadian factory. In 1919 Hoover Ltd was registered in London. The continuing success of the electric sweeper led to the establishment of the Perivale factory in Middlesex in 1932. The "art deco" design of the facade of the Perivale factory is one of the best known examples of the style of that period and is now a listed building which attracts tourists and students of architecture alike. The growth of Hoover plc continued with factories at Cambuslang, Lanarkshire (in 1946), where small fractional horse power motors are produced for Hoover products and supplied to industry and Merthyr Tydfil Glamorgan (in 1948), where 'white goods' are produced. The Hoover company has diversified into the manufacture and marketing of a range of domestic appliances: portable vacuum cleaners, floor polishers and shampooers, washing machines, tumble and spin dryers, dishwashers, fridges, irons, and recently home security systems such as burglar alarms, smoke alarms and locks.

Hoover is now a worldwide organisation with its headquarters and a factory in North Canton, Ohio. A description of Hoover, its products, its marketing and growth as a multinational company is given by Morgan (1974). As recently as November 1983 the Hoover Company Inc. took full ownership of all subsidiary companies. In January 1986 the Hoover Company Inc. was itself taken over by the Chicago Pacific Corporation (CPAC) of the United States.

1.3.2 Hoover plc in Britain

Hoover plc is the largest of the Hoover companies worldwide, even larger than its 'parent' company in North Canton, Ohio. During the last economic downturn of 1980-82 Hoover plc suffered badly and undertook drastic action. As a result, manufacturing ceased at Perivale in October 1981 and subsequently, all vacuum cleaners together with a.c. motors are made in Cambuslang, near Glasgow, where some 1700 people are employed. A product-led recovery plan spearheaded by the newly designed Turbopower upright vacuum cleaner was instigated at the time of rationalisation.

Despite the transfer of manufacturing, Perivale has remained the company's administrative headquarters and centre of engineering research, design and development in the UK. Further cost-cutting exercises initially closed two nearby sites and brought all of the departments to the main Perivale site. Subsequently, having sold the Perivale site, Hoover plc has relocated to Merthyr Tydfil from 1 July 1987.

1.3.3 Role of this project within Hoover plc

Mr Levy (1985), the Marketing Manager, noted that,

> "For our part we have to accept that the success of the Turbopower stands or falls on our ability to achieve noise reduction on the hard-bag models"

The approach to noise control at Hoover plc began approximately 17 years ago when it purchased a mini single-channel frequency analyser and constructed an acoustic room for better sound measurements. This step towards new technology was the trend among the domestic appliance manufacturers of the day. However, Hoover plc did not pursue their noise investigation further than simple measurement of the sound power level of appliances, because;

> "Hoover plc's basic philosophy is that it will not lead but will spend money to remain competitive".
>
> (notes from meeting held 6 Dec 1979)

An example of Hoover plc's stance on noise control is the Turbopower upright vacuum cleaner. This cleaner was completely new: not an adaption of an existing model. During its design, priority was assigned to cleaning efficiency, performance and reliability; these being the hallmarks of Hoover cleaners. The attainment of low noise level was given a low priority and consequently the Turbopower is one of the noisiest cleaners on the market today: a matter of some concern at Hoover. Despite the high noise level the Turbopower range of cleaners was launched in 1983 and has been successful.

To appreciate what the collaborator wanted from this project, it is necessary to see how the project was instigated and the circumstances at that time. Origins of this noise control project were due to two factors:

1. An increase in the general awareness of noise among the public, consumer organisations and the environmental-protection pressure groups,
2. Developments in legislation to reduce noise levels and in particular to enforce noise-labelling on domestic appliances.

Although the developments in legislation began in the early 1980's, the emphasis changed from mandatory noise limits to noise-labelling. But, even so, the pressures upon the industry remain. Competition among appliance manufacturers for lower noise levels is rife.

Despite Hoover plc's change of emphasis from "perceived noise" to a handbook of noise control guidelines, the objectives of this project were unaltered. In the short term, the collaborator expected a definite reduction in the noise level of the Turbopower cleaner, and in the long term, an appreciation of the sources which generate noise and principles of noise control to be collated into a handbook.

The company never claimed to know precisely what level of noise reduction it wanted. Likewise, the supervisory team was uncertain as to the potential of noise reduction possible. It was made clear that it would be up to the researcher to identify the target. Thus a target of 6 dB(A) was set after a review of the subjective rating of noise levels (discussed in Chapter 2).

1.4 The Interdisciplinary Higher Degree Scheme

1.4.1 Origins and aims

The Interdisciplinary Higher Degree Scheme at Aston University was launched in 1968 after the Swann Committee (1968) called for greater co-operation between the industry and the universities, and "bold new initiatives with the PhD".

From an academic standpoint the PhD is;

> "not only a training in methods of research but also a means of preserving the continuity of university research - the aim is to generate new knowledge" Anon, Nature(1974)

Much support for the IHD approach comes from industry where,

> "some engineers and employers claimed that a number of PhD topics had been too academic and irrelevant to the practical world of engineering" Haydon (1986)

The IHD Scheme can be considered to nurture an extension of the sandwich principle from undergraduate to postgraduate degree. The main aim was (and is) to attract talented graduates into central areas of industry and to train them for interdisciplinary careers. A high academic standard is maintained with added the benefit of searching for a solution to a real world problem with real-world constraints. Further details can be found in Cochran (1981).

1.4.2 IHD Research Projects

It is a feature of this scheme that all its research projects are nominated by and conducted in collaboration with an external organisation which has a real 'live' problem. This ensures that all projects have a practical value. The collaborator benefits from the process by way of new data and a structured methodology whilst the student is able to study a problem that is of immediate and genuine importance to someone outside the university environment. The research environment for an IHD project is therefore necessarily more complex than that of the conventional PhD.

The supervisory team for this project is shown below.

Main academic supervisor - Dr John Penny, Head of Mech and Prod Engineering Division, Aston

Associate academic supervisor - Prof Richard Booth, Health and Safety Unit, Aston

Collaborator - Mr A G Breakwell, Hoover plc, Perivale

IHD Tutor - Dr David van Rest, Aston.

The main academic supervisor is responsible for academic standards and overall project management; the IHD tutor ensures the co-ordination and balance between the academic and collaborative interests; the role of the collaborator is to check that the project remains of value to the organisation; and the associate supervisor provides perspectives from other disciplines. The student plays a major managerial and administrative role in the project.

1.4.3 Research Environment and Project Methodology

This noise control project is an example of the transfer of technology. The test methods employed to identify the noise sources of the cleaner are not new. These techniques have been used in high technology areas such as aerospace and automobile industries. In the writer's opinion however, some of these techniques were applied to an upright vacuum cleaner for the first time. Most of the practical work was conducted within the Floorcare Laboratory, at Hoover plc, Perivale. In research of any kind, the historical context of current research must be appreciated. In this respect, there was no shortage or any barrier to the information, formal and informal, which was deemed necessary.

The methodology generally adopted in IHD projects is "action research" using an interdisciplinary approach. This is almost inevitable due to the multiple facets of the problem and the fact that, at the outset, these problems are not fully understood and rarely formulated into research questions.

IHD projects generally fit the "action research" category as depicted by Cherns (1976), see Figure 1.1. The nature of the problem is negotiable, the type of solution is open and the method discretionary. This project can additionally be considered as consultative because at the outset, the nature of the problem had been determined; that is noise reduction.

It was stated earlier that the research environment of an IHD project is more complex that a conventional PhD. Firstly, there are conflicting demands made of the researcher from the various parties to the Scheme. The academic demands of the university are not necessarily synonymous with the pragmatic needs of the collaborator.

Secondly, the researcher is both an apprentice and a consultant, sometimes simultaneously, during the project. The researcher is learning a new discipline in a new organisation yet is expected to provide guidance on and solutions to the collaborator's problems. As the problem becomes more clearly defined, the researcher is seen increasingly in the role of a consultant.

Thirdly, the conventional approach of the research process (problem definition, selection of method, data collection, analysis, conclusions) is seen by Clark (1972) as inappropriate to "action research" which is a cyclical process in which the elements, such as problem definition, investigation, analysis and feedback, may occur at approximately the same time.

Finally, a successful "action researcher" must be in harmony with the collaborating company if he is to obtain the necessary co-operation to produce more valid information and to produce plans that are of value and are likely to be accepted. So, as aptly stated by Pickard (1986);

Figure 1.1 Classification of the Researcher-Client relationship. (Adapted from Cherns (1976))

"IHD by concept, ethos and organisational practice allows the student to undertake research at the interface of different academic disciplines and at that of industry and academia."

1.5 Thesis Structure.

The material described in the remaining chapters of this thesis is presented in a logical order and not necessarily in chronological order. This is for reasons of clarity and because of the cyclical nature of an "action research" project.

Chapter 2 reviews the fundamentals of acoustics, measurement of sound, and the background to the project in relation to the pressures on the domestic appliance industry arising from legislation, standards and consumer opinion. It also summarises the previous noise control experience at Hoover plc.

Chapter 3 reviews the literature on the current techniques for noise measurement, source identification and methods for noise reduction.

The experimental approach, detailed research investigations and the final results of this project are presented in Chapters 4, 5 and 6. Chapter 4 discusses the instrumentation and methods employed during this research project both at Hoover plc, Perivale, and in the Advanced Dynamics Laboratory, Aston University. The chapter finally outlines the experimental test programme which was formulated around the facilities that were available.

Calibration of the measuring equipment and validation of the methods and facilities are presented in Chapter 5 together with all of the experimental results of sound source identification, source ranking and the general results of noise reduction methodology.

Appreciation of the successful noise reducing techniques (as found to apply to the Hoover Turbopower) is presented through the concept of a "Quiet Cleaner" in Chapter 6. The analysis of results is in terms of the noise benefit gained through various modifications and an evaluation of the "Quiet Cleaner" in the context of recognised noise and vibration criteria. The most promising solutions are compared by considering the tooling costs and the cost of extra materials.

Chapter 7 discusses the long term aspects of integrating noise reduction features in designing for quietness. The current design strategy at Hoover is briefly outlined and details of the proposed design policy, geared towards noise reduction, is presented. There were found to be significant gaps in the knowledge base of Hoover's design team in relation to the fundamentals of acoustics and principles of noise control. In helping to bridge the gap to the new strategy, a handbook of design guidelines has been produced. A copy of this handbook is attached in Appendix 4.

Major findings and the conclusions of this noise control study are presented in Chapter 8. Recommendations for further work are summarised in this Chapter 8.

1.5.1 Plan of this Research Project

Figure 1.2 illustrates the main events of this project and their interaction.

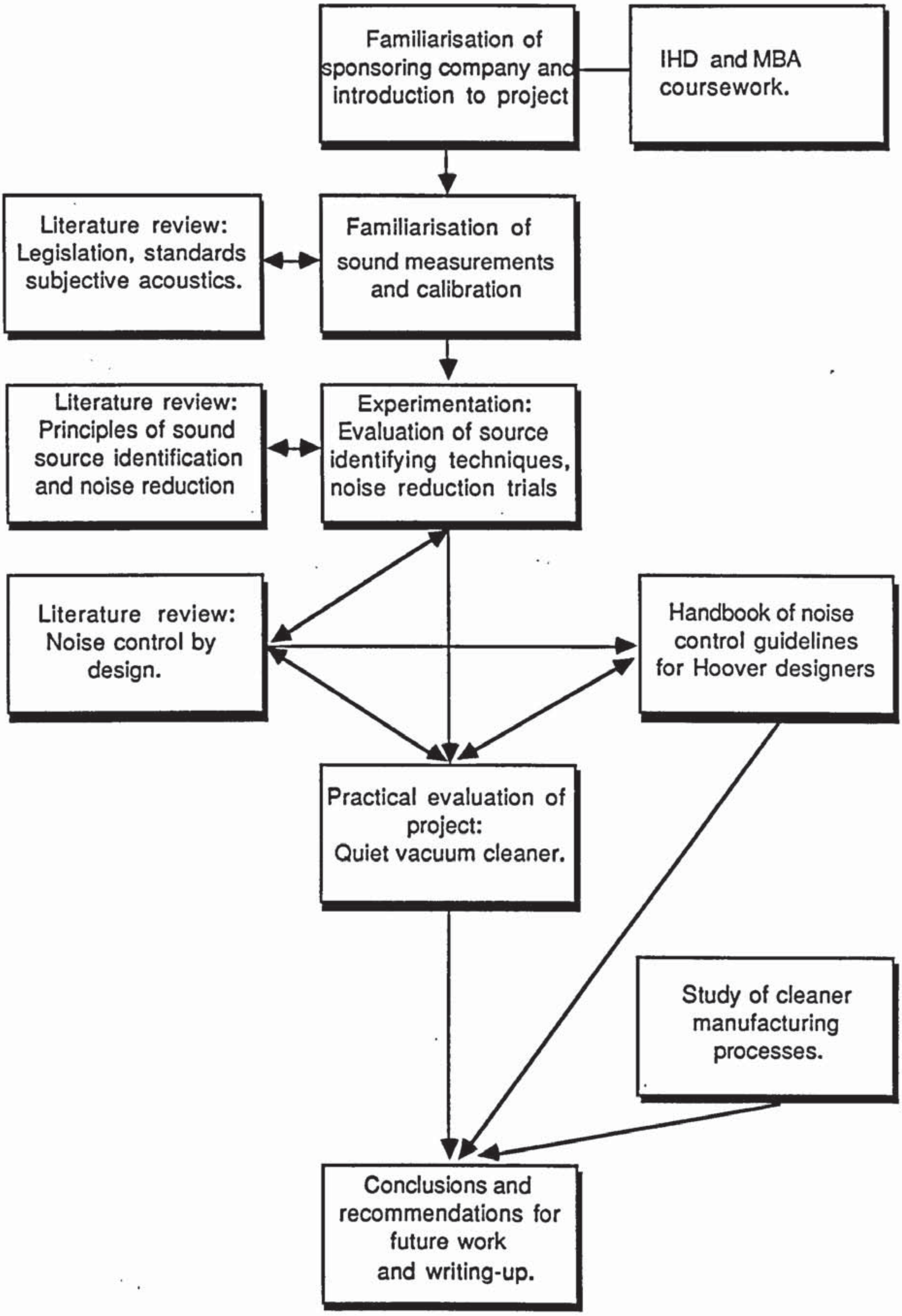

Figure 1.2 Major phases of this project.

CHAPTER TWO: PROJECT BACKGROUND

Contents

2.1 Introduction

Reduction of noise emission of household appliances has become of considerable importance in recent years. This is partly due to a rise in the public's awareness of environmental noise pollution and partly due to the fact that more appliances are now in use in the modern home. As a consequence of increased consumer awareness, greater pressure has been exerted upon the domestic appliance industry to reduce the noise levels of their products. This pressure took the form of a threat of statutory noise limits or noise labelling on appliances. Over the term of this project there has been changes in trends in consumer noise awareness and these are brought out by the results of recent market research studies.

To achieve noise reduction, an understanding is required of the basic definitions of noise parameters, the measurement quantities and procedures. The first section of this chapter therefore, provides an introduction to the subject of acoustics by presenting definitions of the quantitative terms such as units and sound fields and the qualitative terms such as annoyance and speech interference. The physiology of the ear and its hearing characteristics are then briefly described - which introduces the subjective effects of noise. This chapter then reviews the background to this research project and describes the issues which led to an increased awareness of noise. Finally, brief details of Hoover plc's past experience in noise expertise is outlined.

2.2 Fundamentals of Acoustics

The general public are, quite naturally, not familiar with the quantitative aspects of sound. According to Taylor (1970), the word "acoustics" symbolises mysterious phenomena and skills. Although every fifth form physics class is taught the theory of sound; tuning forks, wavelength and frequency, it seldom goes further than that. Consequently, there is great latitude in the meaning of acoustical terms. Therefore an understanding of quantitative and qualitative aspects are important in the study and subsequent interpretation of results of this project.

The production, propagation and detection of sound waves is generally related to the setting up of oscillations. Hassel and Zaveri (1979) defined sound as the sensation perceived by the sense of hearing, or alternatively, as the mechanical radiant energy that is transmitted by pressure waves in the air and is directly the cause of hearing. Noise was defined in Lord et al (1980) as unwanted sound resulting from the negative effects the sound may have on the observer. Thus, noise is a sound that annoys, disturbs, irritates, distracts or causes harm.

The subject of sound is not new. Lord Rayleigh published his account in "The Theory of Sound" in 1894, and since then considerable time and effort has been expended by researchers in producing a comprehensive and solid foundation. Many textbooks such as Morse and Ingard (1968) and Pierce (1981) are available for a detailed background to the theory of acoustics.

2.2.1 Terminology.

a) Sound pressure level, SPL, is the magnitude of the pressure wave fluctuations from the mean atmospheric pressure as a result of a disturbance. Its value depends upon such external factors as temperature and density of air, distance and direction from the source. Sound pressure level is readily and accurately measured by a precision sound level meter.

b) Sound power level, SWL, is the total sound energy radiated by the noise source per unit time. Its value is unique and a fundamental property of a source, all other conditions being equal.

c) Sound intensity level, IL, is the acoustic energy flowing through a unit area (perpendicular to the direction of propagation of the sound wave) in unit time. Sound intensity was not used in this project, its definition is included here for distinction from sound power and sound pressure.

d) Frequency spectrum. The frequency components of a sound signal are determined and displayed over the frequency range. The frequency axis can be linear or logarithmic.

e) Broad-band noise is a signal comprising components of a wide range of frequencies.

f) Discrete tone is a signal comprising a single frequency component.

g) Free-field. A sound field in which there are no reflections of the sound waves from objects or boundary walls.

h) Reverberant sound field is a sound field in which there are total reflections from boundary walls.

Subjective terminology.

i) Loudness, is defined as the strength of a sound signal. It is a subjective impression and is different in concept from noisiness and annoyance. The numerical designation of loudness is provided by "sones" and "phons".

j) Annoyance was defined by Borsky (1972) as being the feeling of discomfort or displeasure associated with sound. Annoyance considers not only the physical characteristics of noise but also the psycho-social circumstances of the observer.

k) The concept of perceived noisiness was proposed by Kryter and Pearson (1963). They defined perceived noisiness as the subjective impression of the unwantedness of sound and because of its clear distinction from loudness, developed into a rating scale of units such as "noys" and Perceived Noise Level.

Sound pressure waves fluctuate about the mean atmospheric pressure. The overall sound level is generally averaged over a length of time. The description of random processes is not relevant to this study but can be found in Bendat and Piersol (1971) and Clarkson (1982). The application of statistics to sound has helped to improve its measurement and understanding. For example;

Leq - equivalent continuous sound level,

L_{90}- continuous level achieved 90% of the time,

Ldn - day-night averaged sound level. This is the Leq value with a 10 dB penalty as a result of increased annoyance for noise during night-time hours of 23:30 - 07:00.

Other special terms have been created for particular industries, for example;

TNI - traffic noise index,

NNI - noise and number index for the aircraft industry,

Lnp - noise pollution level for the aircraft and traffic noise.

The use of the above terms to domestic appliances is not justified as the vacuum cleaner is only operated for short periods. As yet, however, there is no suitable index applicable to domestic appliances.

2.2.2 Units of sound

The hearing mechanism of the human ear is a non-linear system. The quietest sound that can be heard by the average person, known as the threshold of hearing, has a sound pressure of about 20 micro-Pascals at a frequency of 1000 Hz. At the other end of the hearing scale is the pain threshold at a pressure of about 100 Pascals. This is a ratio of more than a million to one, and of course there are higher and lower noise levels than the ones stated here. A logarithmic scale was therefore devised to reduce or compress this 7-figure range to a more manageable size.

Decibel. By definition, a decibel is 10 times the logarithm to base 10 of a ratio of two powers:

$$\text{decibel, (dB)} = 10 \log \left[\frac{Wo}{Wi} \right]$$

where, Wo - output power

Wi - input power.

Decibel levels represent only ratios. No information is given concerning actual magnitude, unless a reference level is stated.

Sound power level SWL. In acoustics the reference level has been set at a pico-Watt, 10^{-12} Watt.

$$\text{Sound power level SWL} = 10 \log \left[\frac{W}{Wref} \right] \text{ dB}$$

where, W - measured sound power,

Wref - reference power level.

Sound pressure level SPL. Some textbooks, for example Diehl (1973) refer to the electrical analogy that for constant impedance, the power ratio is equal to the square of the voltage ratio. That is,

$$\left[\frac{Wo}{Wi} \right] = \left[\frac{Vo}{Vi} \right]^2$$

Vo and Vi are the output and input voltage, respectively. Voltage is the electrical "pressure" which corresponds to sound pressure in acoustics. So, by substitution the sound pressure level is given by,

$$SPL = 10 \log \left[\frac{Po}{Pi} \right]^2 = 20 \log \left[\frac{Po}{Pi} \right]$$

In the same way, a reference value has to be chosen to obtain the actual magnitude. This equation then becomes:

$$SPL = 20 \log \left[\frac{P}{Pref} \right]$$

where, SPL - sound pressure level in dB,

P - measured pressure level, N/m

Pref - reference pressure level 2.10^{-5} N/m

Addition of decibels. As stated above, the decibel is a logarithmic unit and therefore this has to be borne in mind for mathematical operations. In this project, however, a simpler method using a chart shown in Figure 2.1 was used. (There is one limitation to this technique. It cannot be applied when combining sounds of the same frequency. In this case, the phase angle between the signals must be considered. This limitation however, was not important in relation to this project because the frequency spectrum was broad-band.)

Subtraction of decibels. Similar mathematical procedures to those above need to be applied to determine the overall effect due to subtraction of one noise source. The curve for subtracting sounds is shown in Figure 2.2.

2.2.3 Subjective impressions of noise level

Subjectively, a noise reduction of 3 dB is barely noticeable although, 50% of the original sound power has been removed. A 10 dB reduction in noise level represents a reduction of 90% in sound power and yet subjectively it gives the impression of halving of the noise level. The table below, adapted from Hassel and Zaveri (1979 p33), summarises the change in the subjective impression in relation to the actual energy level. It follows that the ear perceives noise in a way which is approximately logarithmic.

Decrease in noise level	The actual effect	Subjective effect
3 dB	half the energy	barely perceptible
5 dB	third of "	clearly perceptible
10 dB	one tenth "	half as loud

Loudness units. A better way to assess noise reduction is in terms

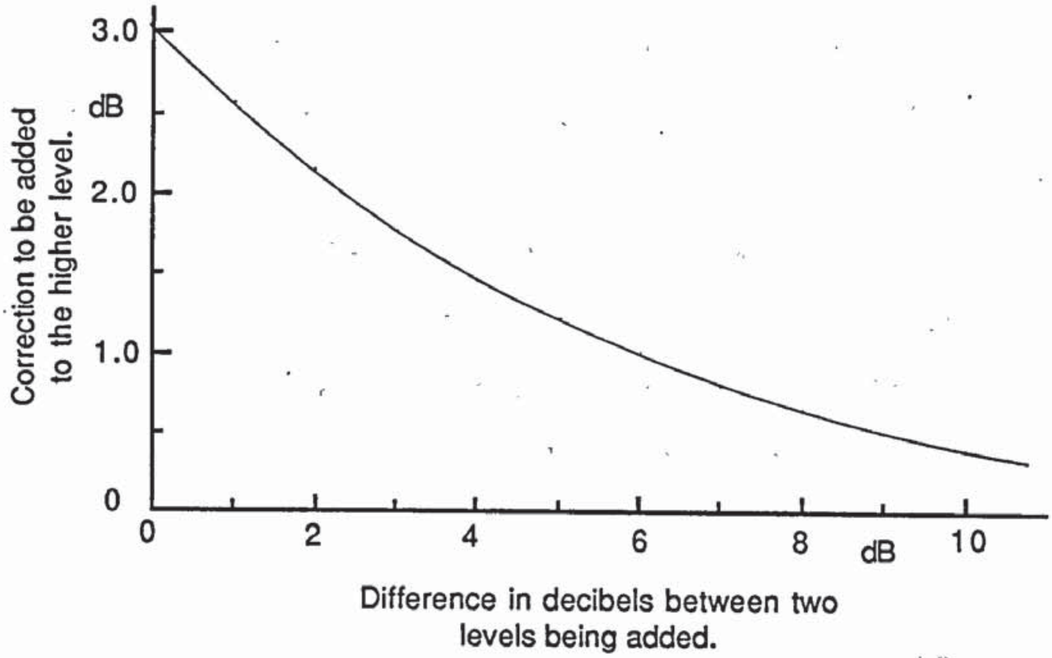

Figure 2.1 Addition of 2 sound levels.

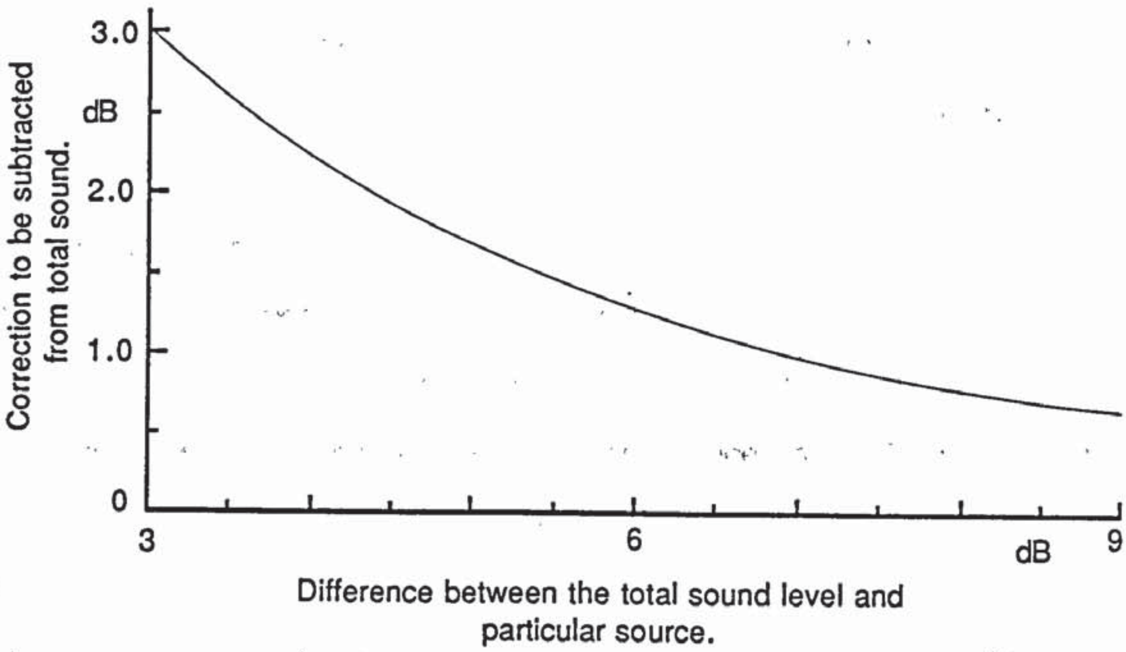

Figure 2.2 Subtraction of one source from the total sound level.

of loudness because this is based on the opinions of a group of listeners.

a) Phons. This represents the loudness, in phons, of a 1000 Hz tone that sounds as loud as the noise in question. The phon scale is logarithmic and consequently it does not express the "subjectiveness" of a sound signal effectively: eg. a sound that is twice as loud as another does not correspond to twice as many phons.

b) Sones. By definition, a tone of 1000 Hz, 40 dB above the hearing threshold produces a loudness of 1 sone. This scale is linear. Hence, a sound that is twice as loud corresponds to twice as many sones.

An early aim of this project was to reduce the "perceived" noise of a vacuum cleaner. This would have considered the reduction in noise in terms of sones, phons and the perceived noise level. Although the project emphasis moved away from perceived value to a reduction in the overall noise level (because of the impending noise labelling requirements), for completeness, the noise reduction achieved by this project will be illustrated by a "Quiet" vacuum cleaner using the loudness parameters.

2.3 Characteristics of hearing

2.3.1 The ear and the mechanism of hearing

The hearing sensation is the result of a complex process which converts pressure fluctuations of varying amplitude and frequency into neurological impulses which the brain interprets as sound. There are still many parts of the process which are not fully understood, particularly the central functions taking place within the brain. A diagramatic structure of the ear is illustrated in Figure 2.3. The transmission of sound through the ear and details of the hearing mechanism can be found in Engstrom et al (1970) and Davis and Silverman (1970).

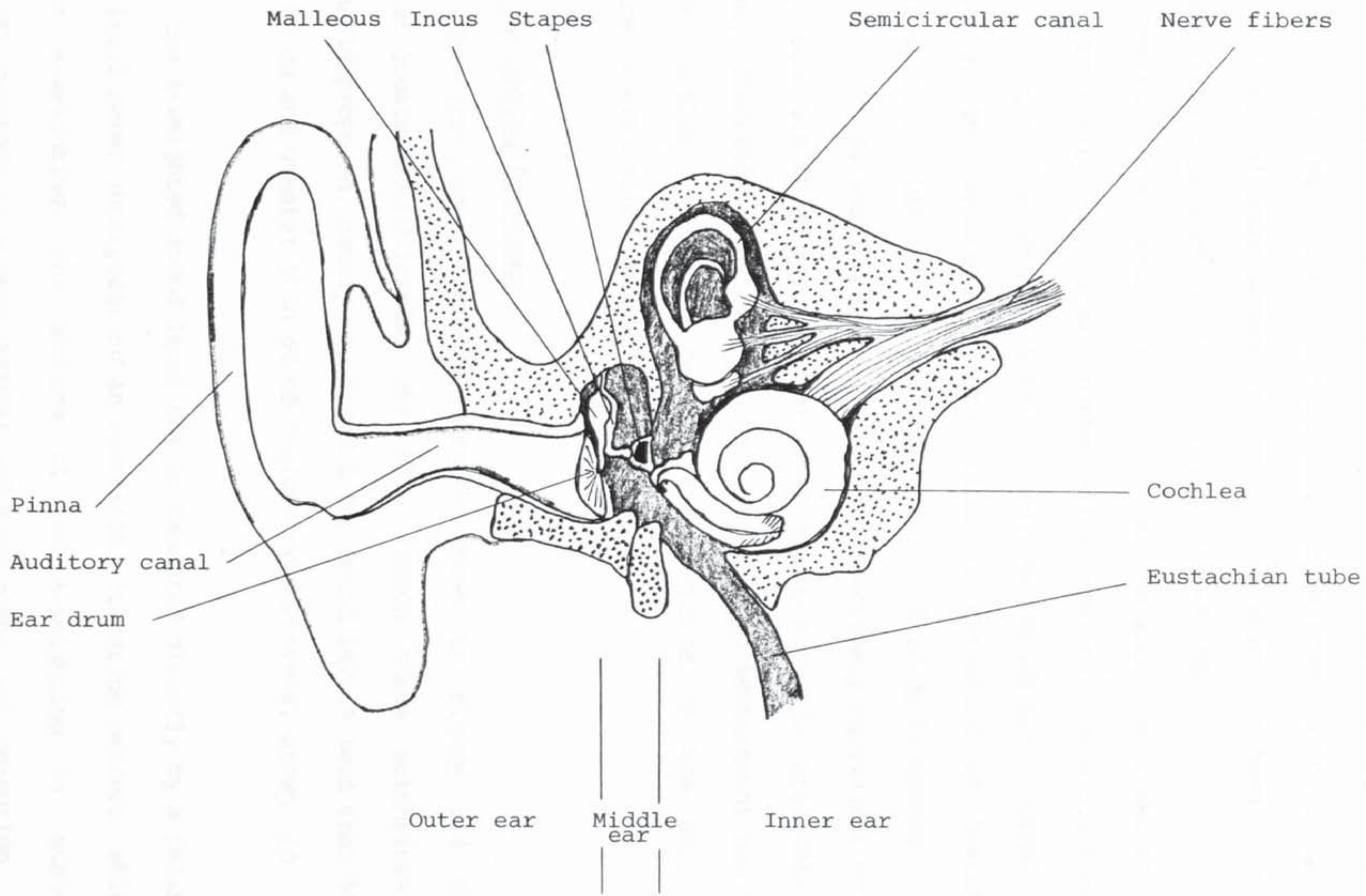

Figure 2.3 The main parts of the ear.

2.3.2 Frequency characteristics

The ear of a normal young and healthy adult is generally regarded as being sensitive to sounds in the frequency range 20 to 20000 Hz. The dynamic range of such an ear is considered to be from -15 to 150 dB sound pressure level, although damage tc the cochlea, which has been identified as a sensitive organ, starts to occur around 90-95 dB.

The ear is not equally sensitive over the above mentioned frequency range. Sensitivity is greatest in the 1-4 kHz region. Figure 2.4 shows the free-field equal loudness contours of pure tone sounds over normal range of hearing. The lowest curve is the binaural threshold of hearing at different frequencies. It follows that a sound of 20 Hz frequency having a strength of 70 dB sound pressure level is subjectively equivalent to sound of magnitude 5.0 dB at frequency of 1 kHz and both sounds are only just audible. Therefore, the "raw" or the initial sound measurement has to be adjusted so that it better matches the response of the ear. These adjustments are called weighting functions.

2.3.3 Weighting functions

The normal equal loudness contours shown in Figure 2.4 changes shape at greater sound levels. For this reason three weightings were originally proposed, namely, A, B and C for sound levels less than 55 dB, 55 to 80 dB and greater than 80 dB regions respectively, shown in Figure 2.5.

The A-weighted sound level may be measured directly by a calibrated sound level meter incorporating an electrical filtering network which is set for A-weighting. The success of the A-weighting in subjective acoustics has made it a very popular and easy means of measuring noise. Hence, because it is widely used, only the A-weighting was used for noise measurements through-out this project.

Figure 2.4 Normal equal loudness contours for pure tones.
(from Hassel and Zaveri (1979))

Figure 2.5 Standardised weighting curves.
(from Hassel and Zaveri (1979))

2.3.4 The effects of noise on people

The auditory impression of sound may be categorised in many ways. Considerable research and effort has been spent in trying to define and quantify human response to the physical characteristics of the sound exposure. High noise levels have been known for a long time to cause loss in hearing. An example is the occupational hearing loss; "boilermaker's deafness" was known in the nineteenth century. According to Smith (1984), excessive noise can also cause undue stress, thicken the blood and has recently been suggested by Connell (1984) to trigger heart attacks. Noise does not have to be excessive to cause adverse effect on people. The intensity, spectral and durational aspects of noise play differing roles depending upon the type of interference. These effects may be categorised as follows:

- noise induced hearing loss,
- annoyance,
- speech interference,
- masking of other signals.

a) Effect of excessive noise on hearing

There are two major types of hearing loss: conductive and sensori-neural. The former is associated with the outer or middle ear and is usually caused by perforation or an infection of the middle ear bones. This effectively blocks transmission of sound to the cochlea or the inner ear. A sensori-neural loss is irreversible and results from damage to the cochlea or neural structure of the inner ear. This type can be caused by birth defects, noise, fever or trauma and is associated with ageing (presbycusis). Conductive loss is medically correctable but sensori-neural is not.

Criteria for Hearing damage risk

The Code of Practice (1972) for reducing the exposure of persons to noise is based on the study of steady-state noise by Burns and Robinson (1970). This code suggests that the maximum benefit in terms of protection of hearing will be gained by limiting noise exposure to 90 dB(A) over an 8 hour day. Nevertheless, this code states that there is still a risk, albeit less, at levels below 90 dB(A). This level prevents loss of hearing in 80% of those exposed. Based on this noise exposure the safe values of sound level and duration can be extrapolated as shown in Table 2.1. In view of these findings, a domestic vacuum cleaner which has a sound pressure level of 80.0 dB(A) in a normally furnished room will not pose any threat to hearing loss.

b) Annoyance

The extent to which noise irritates people is the most intangible aspect of the subject of noise. The physiological and psychological

Sound level dB(A)	Permissible duration of exposure per day.
75	safe level
78	"
81	"
84	"
87	16 hrs
90	8 "
93	4 "
96	2 "
99	1 hr
102	30 min
105	15 "
108	7.5 "

Table 2.1 Permissible equivalent continuous sound level, Leq.

factors are complex and vary greatly between individuals. Therefore, the aim of much recent research was to identify the physical characteristics of noise and combine these to form a scale to predict subjective response consistently and accurately. A single scale for rating annoyance is not available. The A-weighted sound level is, however, a good proposition. Some of the widely used rating scales defined in section 2.2.1 are based on the A-weighted sound level but combine "corrections" of a social context.

The overall effect of annoyance is that it interferes with a broad range of activities, such as:

- speech communication, face-to-face and telephone
- listening to, radio, television, music
- mental concentration
- relaxation
- sleep.

The annoyance quotient of noise depends upon;

- background noise level,
- frequency components. Noise of extremely low or high frequency is more annoying than sound of middle frequency,
- magnitude of noise. Loud noise is judged to be more annoying,
- duration of exposure. That is intermittent, irregular or rhythmic,
- past experience.

Plunkett (1958) found that annoyance aspects were complicated by bias or previous experience to noise. Results of an analysis of community complaints to noise by Guz et al (1983), suggested that the instrusive noise is integrated, unconsciously, into perception of general noise environment and then, consciously, into memory of past arousals, before initiating a complaint.

Fields and Walker (1982) conducted a survey to investigate the relationship between the noise level and annoyance for different sources, namely railway, aircraft and road traffic. They found that railway noise was less annoying than road noise, which in turn was less annoying than aircraft noise. It can be argued that individuals have a positive attitude to things they admire, for example railways, because of romance and excitement (the Orient Express) and negative attitude to others, such as aircraft, for example the fear of the craft crashing.

c. Speech interference

Ambient noise can make speech completely inaudible or reduce its intelligibility by masking only some frequencies. This effect on face-to-face speech is usually assessed in terms of the maximum distance between the speaker and listener for a given background noise level. This relationship is given in Figure 2.6 for typical voice levels. Interference with speech reduces the opportunity for a relaxed and a reliable communication. In a noisy environment this has important implications. Crucial unexpected messages may not be comprehended. This could be a hazard to safety, cause inefficient working, lead to the adoption of non-communicating life style with reliance upon gestures or facial expression, as explained by Kryter (1970).

Early work by French and Steinberg (1947) and Sumby and Pollack (1954) led to the formation of the Articulation Index (AI) as a measure of speech interference. The AI was expressed as a value between 0 and 1, indicating 0% and 100% intelligibility, and was the average of A-weighted sound levels in three octave bands (1kHz, 2kHz and 4kHz). Recent studies found the octave bands at 500 Hz, 1 kHz and 2 kHz to be the most important for speech, stated in ISO Recommendation R 3352 (1974). Correspondingly, the average is called the Preferred Speech Interference Level (PSIL). The validity of the speech interference level as a predictor for face-to-face communication in high ambient noise, was questioned by Waltzman and Levitt

Figure 2.6 Rating chart for determining speech communication capability from speech interference levels. (from Lord, Gatley and Evensen (1980))

(1978). In their opinion, the data on which the PSIL was based was obtained in an idealised, free-field environment and did not consider visual cues. Waltzman and Levitt conducted experiments considering these factors and concluded that in practice the values of PSIL refer to a worst case.

d. Masking of other signals

In addition to face-to-face speech there are other forms of communication which are designed for delibrate arousal, such as warning, safety and indicative signals. Most of these signals are of high intensity and comprise frequencies which are most intrusive. Hence they are so unusual that they cannot easily be masked by noise of a vacuum cleaner. Usual household activities, such as television viewing, phone conversation and casual conversation were among the items discovered by the writer to be masked by the noise of a vacuum cleaner.

2.3.5 Effect of vibration on people

Like sound, vibration has been familiar for a very long time. Most vibration has a mechanical origin. Although, structural vibration may have its origin in vortex shedding or unsteady fluid-flow. The physical effects of vibration on the human body can generally be classified as whole-body, such as motion-sickness and sea-sickness, or hand-arm, such as dead-hand and vibration white-finger.

The objectives of most of the early studies was to determine the response to low frequency whole-body motion when standing or sitting. These tests were carried out on vehicle drivers and aircraft pilots, whose ability to perform complex tasks under adverse conditions, including vibration, is particularly important. Most of the available data was brought together in ISO 2631 (1978), which outlined the criteria for vertical and lateral vibration in the frequency range 1 to 80 Hz. The data on which this standard is based, and hence the recommendations, were queried by Osborne (1986). Osborne suggested that the lateral response curves are based on just two experiments, and hence unacceptable because according to him there were as many responses as subjects under test. Therefore, some vibration standards should be treated with caution.

Hand-arm vibration levels of many commonly used power tools are sufficiently high to cause damage if the exposure time is prolonged. Unlike whole-body discomfort, vibration applied to the hand or the arm may produce local physical damage. At low vibration levels the exposure may only lead to discomfort, but high levels may lead to diseases affecting the joints, blood vessels and circulation. In severe conditions, this leads to permanent damage, known as "dead-hand" and vibration-induced white-finger. Standards for hand-arm vibration are at an early stage. National and international bodies recently produced a Discussion and Draft Proposal which suggests guideline levels for exposure to hand vibration as shown in Figure 2.7.

2.4 Pressure on the Appliance Industry to reduce noise levels

In recent decades, society has placed increasing value on a safe place of work and a pleasant environment. The general public is now more aware of environmental factors. It would be naive to imagine that this trend will not continue and we must, therefore, look at the probable ways in which this awareness will cause pressure to be exerted on the appliance industry to reduce noise levels. This pressure may come from Parliament, through legislation and Select Committees, or the public themselves through consumer organisations. Pressure from the latter will take the form of bad publicity or non-recommendation of products. Pressure from government will be in the form of renewed standards, regulations and/or noise limits or noise labelling.

Figure 2.7 Exposure guidelines for vibration transmitted to the hand.
(from Brock (1980))

2.4.1 Standards

Standards generally imply a documented procedure agreed by a collective body of experts in order to give a common base to help in solving problems. By using such a method, noise levels can be determined to a known degree of confidence. The first noise standards were introduced in the early 1960s as consequence of much research in the previous decade. Bruel and Kjaer (1981) collated a list of the National and International standards. At present, the International Organization for Standardisation is the accepted body for establishing standards on an international basis. These standards are subsequently adopted, albeit with small changes, by various national bodies.

As mandatory regulations presuppose standardisation, so the latter presupposes state-of-the-art technology. In the event, according to the Noise Council (1986), "the UK is one of the few Western Countries still using a policeman's ear as the measuring tool".

Standards for household appliances.

The need for standards relating to sound emitted by household appliances and its measurement was first identified by Baade (1971) and Roewer (1973), who suggested a unified approach to sound ratings and standards to enable noise control to progress. Noise control depends on knowledge of the three major aspects involved in acoustics, namely emission, transmission and reception. Therefore, precise acoustical standards are required for each item. Moreover, it would help if these standards established engineering criteria in a manner useful for evaluating similar noise sources among the product range of a company that cannot afford proper acoustical facilities.

The test code for determining the sound power level of household appliances, IEC 704.1, is broadly based on ISO 3743 (1976). ISO 3743 outlines a practical engineering method based on the precise method of ISO 3745 (1977). Mansbach (1978) compared the values of the sound levels determined by precision measurement and engineering methods and found that the results were in good agreement. The difference was of the order 1.0-1.5 dB(A). Mansbach's results justify the engineering method, which was devised to take noise measurement out of the laboratory and into the factory.

The selection of facilities and especially the environment for noise testing is very important. Jackson and Leventhall (1973) proposed using an "acoustically average" domestic environment for a representative measurement of the noise level of domestic appliances. ISO 3745 requires either an anechoic room which is very costly, or a flat outdoor area away from obstructions. The latter can be extremely inconvenient. Kristensen (1976) and Rolando (1985) describe the design, structure and correlation of a cost-effective special reverberant test room.

2.4.2 Legislation of noise levels

The Wilson Committee (1963) recommended statutory noise limits on powered construction plant and on motor lawn-mowers. Initially, such a limit was not feasible probably because the methods for quantitative measurement of the noise level were not reasonably practical. It is only 13 years since the government set a mandatory limit. This was for noise exposure at work: as yet there are no noise limits or noise labelling requirements for household appliances. In order to identify the possible models for noise legislation on household appliances, we need to review the developments in other major areas, such as road traffic and occupational noise.

Automotive industry

In 1970, the EEC introduced their first Directive on motor vehicle noise, 70/157/EEC. This laid a noise limit of 82 dB(A) for private cars. Seven years later, a subsequent Directive 77/212/PEEC lowered this limit to 80 dB(A). This limit was made mandatory in the UK for cars manufactured on or after 1 April 1983; a delay of about 6 years. Meanwhile, in June 1983 a new lower limit of 77 dB(A) was proposed by the EEC in the document COM(83) 392, to apply with effect from 1 October 1986. In order to put extra pressure on vehicle manufacturers, Members of the EEC are free to exercise greater stringency (for example Switzerland has a 75 dB(A) level, refer to Verdan (1985)). Alternatively, with effect from 1 October 1988, EEC countries can prohibit the entry of all new vehicles which fail to meet these noise requirements. Some countries, however, choose to allow vehicles to be imported albeit with a substantial monetary penalty (a fine was levied on Jaguar Cars in 1985-86 for not meeting fuel consumption requirements of the USA).

Enforcement of such regulations is feasible because all mass produced cars have to meet vehicle type-approval laid down by the country to which the vehicles are exported. Type-approvals exist for other important areas, such as safety, crash-worthiness, lighting and exhaust emissions. There does not seem to be any real competition among vehicle manufacturers to produce cars of still lower noise levels. This may be because the vehicle is marketed as a "package" which differs for cars in a particular market. In general, there is little incentive to reduce noise or emissions below the statutory requirements once this is met. Attention is drawn away from noise to arbitrary added-value features.

Noise exposure at a place of work

The Department of Employment produced a Code of Practice (1972) which recommended a maximum noise level of 90 dB(A) (Leq) over an 8-hour working day. The employer has the duty to provide safe working conditions whilst the employee was required to take reasonable care for his own protection, including the use of protective equipment where provided. In 1978, the Health and Safety Executive produced a discussion document on audiometry in industry. It recommended mandatory audiometry to everybody exposed to a noise level of 105 dB(A) Leq.

In July 1984, the European Commission approved an amended proposal for a Directive on the protection of workers from noise. The original proposal was for a noise limit of 85 dB(A) Leq, but this level was considered by the European Economic and Social Committee to be

> "too low for the immediate future because of the technical and financial burden this may cause, and that it was based on criteria which are not fully scientific". NVB (1984a)

The amended proposal raised the limit to 90 dB(A) Leq. Nevertheless, the group agreed that 85 dB(A) must be the ultimate goal if the aim is genuinely to protect workers. With statutory noise limits at hand, it is up to the Factory Inspectorate to enforce the law. Rather than adopting a hard line and direct confrontation in the courts, the Inspectorate has in the past preferred gentle persuasion.

There is a stark contrast in these two examples. According to NVB (1984b) the automotive industry has kept abreast of the noise limits whereas noise levels at the work place have fallen behind expectations. As Pearce (1985) suggests, this is primarily because the new machinery in the working environment has become noisier.

2.4.3 Legislation for household appliance noise

In the early 1980's, there had been strong pressure from consumer groups for legislation of noise levels in one form or another. These consumer groups were too small in number to have any significant effect. Nevertheless, with the "Green Party" having won 37 seats in the German Parliament and the then likelihood of a greater number of SDP/Liberal seats in the 1983 UK elections, the demands for some type of legislation was envisaged in the UK. This led the European Association of Appliance

Manufacturers to review a legislative proposal which had been issued by the EEC Commission. The Council Directive OJ No: C181/2 was discussed by manufacturers, consumer groups and the environmental parties, and was a major reason in the allocation of a high priority at Hoover plc for noise reduction of upright vacuum cleaners. Since then, the emphasis has moved from mandatory noise limits to product noise labelling which was initially discussed in 1979.

2.4.3.1 Noise limits or noise labelling?

Since the early 1980s when the subject of noise limits or noise labelling was first addressed, Tatschner (1984) found the stage was dominated by conflicting views, that is;

"- a strong consumer lobby which aimed for a noise limit of 65 dB(A), and

- the appliance industry which was united in resisting this because in their view:-

  - no absolute noise limits should be set,
  - legislation should be for noise labelling,
  - noise labelling should be manufacturers' responsibility on the basis of self-certification."

Most of the large companies took the view that if absolute noise limits could not be avoided then 70 dB(A) could be tolerated as most of the new model suction cleaners were already around 68 dB(A). The author notes that these noise levels must be the sound pressure levels which are apparent from Table 2.2. This table shows noise levels for suction type cleaners.

2.4.3.2 Implementation strategy

The appliance industry was quick to respond to an increase in noise awareness and this led to improvements in noise emission during the initial period without any form of legislation. For this reason, the

	Rated power (watts)	Sealed suction mm H_2O	Open airflow lt/sec	Peak airflow efficiency %	Sound power level dB(A)	Sound pressure level * dB(A)
Hoover sensotronic	1000	2100	38.0	22.5	78.0	66.5
AEG 6004/8	1000	1850	32.5	18.6	84.0	72.5
Electrolux 350E	1000	2070	35.0	20.3	80.0	68.5
Goblin Electronic	1000	1875	31.5	12.2	91.0	79.5
Hitachi CV 3300	1000	1550	27.0	11.9	79.5	68.0
Hotpoint 8380	800	1610	34.5	21.0	78.5	68.0
Miele S240i	1000	2100	37.3	21.5	78.5	67.0
Philips P70/74	1000	1795	30.0	16.3	75.0	63.5
Kenwood C1000	1000	1860	31.5	15.6	79.0	67.5
Krups ST1300	1200	1350	27.5	8.6	84.0	72.5

* Sound pressure level in free-field environment at a distance of 1.5m

Table 2.2 Performance and sound levels for cylinder vacuum cleaners.

industry emphasised that free competitive market forces were the best way of ensuring continued progress. It argued that legislation may burden the industry with unreasonable standards which would be difficult to maintain and in addition, which would add to manufacturing costs for which the consumer would ultimately have to pay.

Household appliance noise is different from both the traffic noise and the noise exposure at place of work. The major difference lies in the duration of the noise level. Whereas traffic noise and industrial noise lasts for longer periods, vacuum cleaners are used for a short time each day: perhaps half an hour. Thus from an exposure viewpoint, there is no risk to hearing. There is, however, one element in common between vacuum cleaners and automobiles: both are in consumer markets. For this reason, noise levels of these products will be subjected to frequent checks and customer feedback. Consequently, the self-certification strategy that has been proposed by appliance manufacturers, may prove to be successful and self- sustaining. A test similar to a vehicle type-approval is considered as the alternative.

When a Directive is issued by the EEC Council it becomes mandatory within the EEC. Nevertheless, the Directive requires a formal adoption by each member. A Directive which concerns noise labelling of household appliances, reference no: 8425/86, has been the subject of discussion and revision for several years and its progress has been extremely slow. The problems are identified as follows:

- it is to be introduced after energy-labelling for which there has been a long wait because energy-labelling for vacuum cleaners is to follow that for washing machines which has not yet been resolved,
- indecision whether to quote the mean, or the range and a maximum, or minimum-maximum figures on the noise label,
- indecision whether to quote tolerance,
- measurement standards for a range of appliances have not yet been issued. In fact only those for convection heaters and vacuum cleaners have been published.

It appears then, that legislation for noise labelling of vacuum cleaners will not be introduced for a few more years: a far cry from the "panic" situation in the early 1980s. Subsequent relaxation of priority was vindicated during the term of this project. No action is anticipated for noise limits within Europe in the foreseeable future.

2.4.3.3 Future developments

We have seen how traffic noise was progressively reduced by lowering targets over a number of years. Similarly, the recommended noise level in industry will be reduced to 85 dB(A) Leq (proposal for a Council Directive laid out in the EEC Document No 10322/82) and possibly 80 dB(A)

Leq at a later stage.

A manufacturer-led noise reduction plan, as a result of self-certification and free competition, will only reduce the noise level to a certain extent. This is because appliances possess other value-added attributes. Consumer demand for a product is dependent upon its major features. The author believes that in a "free" market, the consumer may not value low noise levels as highly as other features and further noise reduction will be achieved by legislative incentives.

In Germany, noise emitted by lawn-mowers was progressively reduced by incentives. Kurer and Irmer (1979) discuss the virtues of an ordinance which formed a model for noise reduction by incentives rather than statute. In 1976, the ordinance arranged for the introduction in 1980 of noise levels that were 6-7 dB(A) lower than values at that time, with possibility of even lower target and noise labelling at a pre-determined later date. In addition to these noise targets, the use of noisier lawn-mowers was curtailed to specific times of a working day and banned on Sundays and Holidays. These actions induced manufacturers to lower the noise of lawn-mowers and advertise the appropriate machines as "Sunday mowers". This is the manufacturers' direct response to consumer noise awareness. In the future, this awareness will not be limited to electrical appliances. According to NVB (1984c), consumer awareness may cover other items such as telephones, especially cordless and door bells.

2.4.4 Consumer awareness

Awareness of noise pollution has been noted by various bodies, including commissions of the Department of the Environment. The problem of noise was reported by the Wilson Committee in 1963. More recently, the Royal Commission on Environmental Pollution (1984) identified noise as a matter of widespread public concern. Noise was the most frequent cause of complaint to local environmental departments as illustrated in Table 2.3.

Table 2.3 Environmental issues raised more than once to the Commission in 1982. (from Royal Comm on Env Pollution, Cmnd 9149)

Awareness of vacuum cleaner noise

Consumer reaction to noise of a vacuum cleaner shows an enormous spread. Hawkins (1979) realised that published work on subjective response to appliance noise lacked a systematic approach. His work to assess respondents' feelings towards the noise of washing machines and vacuum cleaners in particular, discovered mixed reactions: a majority expressed a neutral feeling and only slightly more people disliked the noise than those who "liked" it. The older generation expressed a positive reaction to vacuum cleaner noise as shown in Figure 2.8.

Appliance noise seems to be increasingly mentioned in the press and the literature of consumer associations. These publications have a wide circulation among the general public. By nature, most comments on noise reflect the subjective feelings or ratings, for example a report in Which? (1986) and Electrical Retail Trading, ERT (1986), yet their comments and respective recommendations can be very influential because these are literally taken at face value. With this background, it is the writer's view that even noise labelling will have restricted value in portraying a truer picture. The A-weighting scale on which such data would be based is not the best scale for subjective rating because it fails to emphasise discrete tones. An evaluation of the main rating scales will be presented in Chapter 6.

The results of market research

In March 1980, Hoover plc (1980) conducted a market study of upright cleaners in the UK, France and Germany. The aim was to investigate the broad views of current owners of Hoover and Electrolux upright cleaners with regard to their best "liked" and most "disliked" features. The overall conclusion was that noise reduction was ranked by owners of Hoover cleaners in all three countries as the most wanted improvement. Criticism of the noise level, particularly of Hoover cleaners, was strongest on the Continent, principally in Germany where 66% of the owners mentioned noise specifically. This was the main reason inhibiting sales growth of upright cleaners in the German market. Another reason why German owners criticised upright cleaners in particular is that the German market is predominantly of suction cleaners which are inherently quieter so that the upright variety were subjectively rated against them.

In March 1985, another market study was conducted by Hoover plc (1985). The aim of this particular exercise was to investigate the degree of interest in five new features to be incorporated in the redesigned

Figure 2.8 Subjective rating of vacuum cleaners.
(from Hawkins (1979)).

Turbopower model. Noise level was again of interest. The motor speed of a cleaner was reduced to obtain a 5 dB(A) reduction in its overall noise level and the modified cleaner was evaluated against the standard Turbopower model. In a room that was comparable to an average lounge, most respondents thought that the standard cleaner (having a sound power level of 87.5 dB(A)) was not considered to be particularly noisy. The majority thought that the "quieter" cleaner was only a "little less noisy" at 82.5 dB(A). It was concluded that low noise was not ranked highly and suggests that in the UK noise does not seem to be really important to most of the users of upright cleaners. It follows that any reduction in noise has to be quite marked before it becomes noticeable, confirming the subjective ratings given in section 2.2.3. 70% of the respondents were willing to pay up to £5 extra for a quiet cleaner but this percentage dropped sharply when asked to pay more than this.

In a survey of any kind it is difficult to guarantee that results are free from partiality. As mentioned earlier, arousal to noise is judged against past experience: positive or negative. In addition, however, the perception of high noise may be influenced by the possibility of a claim for damages: if there is a chance of a valid claim then a negative reaction can result. People are normally "defensive" of their purchases, that is, only a few would admit to having purchased an appliance with an excessive noise level. Consequently, such factors lead to biased opinions.

2.5 Past noise control experience at Hoover plc

2.5.1 Sound measurement facilities

The sound room at North Canton was designed in 1929 by the University of Cincinnati. The room is bigger than the acoustic room at Perivale and has the benefit of better acoustic treatment: absorptive lining on the surfaces and revolving diffuser vanes. Although sound level meters have been available at both sites for a long time, it has proved difficult to identify exactly when rigorous sound investigations were first carried out at Hoover.

Hoover plc engineers realised that noise measurement was a complex problem which could lead to confusion arising from the many different units and methods available. Therefore, a great number of the early internal publications were intended to give guidelines for uniformity within the company. An engineering report by Wallbank (1978) reviewed the state-of-the-art of acoustics and recommended that;

- noise measurements of the sound pressure level in dB(A) are adequate for internal comparison and development work,
- use of sound power level, sones and phon should be understood because these are used generally,
- determination of a correction factor to correlate Perivale results with semi-anechoic or other sound fields,
- all necessary information to accompany the quoted noise measurement.

While measuring the sound power level, simultaneous investigations on both sides of the Atlantic led to an appraisal of a reference sound source (RSS) and the juxtaposition method. Trials were then initiated at Perivale, to determine noise measurement consistency in the acoustic room and performance room (of much larger size than the acoustic room and with humidity and temperature control) and the anechoic chamber at NEL, East Kilbride. Use of the RSS put renewed confidence in noise measurements. Familiarity with sound measurements exposed three important factors: directivity of source and the position and distance of microphone. Hoover engineers believed then that any measurement "standard" should consider these points. Despite this, a standard procedure was not written for inter-company noise measurements.

When the IEC Draft 59 for the determination of airborne noise of household appliances appeared in 1980, it was reviewed both at Perivale and North Canton. As a result, inter-laboratory correlation of sound measurements became possible and in 1981, laboratory specifications were written for the determination of vacuum cleaner noise at North Canton. Summary of the IEC 704 test code and the procedure adopted for sound measurement at Perivale are described in Chapter 4.

2.5.2 Noise control experience

Details of noise reduction work at Hoover plc prior to 1979 are not clear, but some experience is obviously evident because there was noise specification for the Turbopower cleaner at its design stage. The cleaner was designed about this time for launch in 1983.

Since 1979, much of the noise reduction has been problem orientated towards the following areas:

- fan design modifications
- exhaust silencing
- balance of rotating parts
- agitator redesign
- transmission path analyses
- assessing the competition.

Fan design

Blade passage tone was recognised a long time ago and was thought to radiate from the exhaust and the inlet. No further work was done until 1985 when three alternative designs were evaluated: these were highly swept back blades, taller blades and reduced fan diameter. Results showed that there was no distinct reduction in the passage tone or the overall noise level. During the author's visit to North Canton in July 1986, a 12 blade fan was being evaluated there. This had shown some promise but no results or further information is available.

Production quality of the suction fan in a Turbopower cleaner has always been in doubt in that its as-moulded imbalance is influenced by many slight changes in manufacturing conditions. The 4 cavity tool produces only 75% fans within the design specification.

Armature balance

The motor has been considered to be the biggest contributer to the overall noise but its actual contribution has been rather varied. In 1979, motor imbalance was thought to contribute 4-5 dB(A) to the overall noise level (not a Turbopower) whilst tests on the Turbopower model by Ogilvie (1984) gave the value as 1 dB(A). An analytical study of the rotating elements of the motor was done in 1982 at North Canton using a computer model. The suction fan was seen to be the major source of motor unbalance although the variance of armature imbalance in production motors was also of concern (as shown by trials on pre-balanced and through-bore interference fit fans by Ogilvie (1984)). Post-assembly trim balance on the suction fan was proposed as the only effective and viable production method to check motor balance. A two-plane dynamically balanced armature-fan assembly did not necessarily improve cleaner noise level except for the octave band centred on 250 Hz. Hence, a single-plane trim balance of the suction fan end plane was proposed because this bearing plane was believed to be more critical than the vent fan end.

Agitator

The agitator which comprises "beater bars" and bristles to sweep the carpet, was previously made of steel, and in 1979 narrow-band frequency analysis illustrated that plastic agitators had better balance.

Noise transmission path

In 1979 the foot of the cleaner was identified as the major noise source. The transmission paths were the hood vibration and the gap between floor and nozzle for blade passage tones and air turbulence.

Assessment of competition

The noise levels for Hoover cleaners are regularly compared with the competition. The results have already been shown in Table 2.2.

2.6 Review

This chapter has outlined the fundamentals of acoustics and the terminology appropriate to the scope of this project together with the circumstances which prevailed at its outset. The effects of excessive noise on people have been highlighted. It follows that even low noise levels cause annoyance. These circumstances have led to an increase in consumer awareness of the environmental issues concerning noise pollution and subsequently led to pressure on the domestic appliance industry to reduce product noise levels. As the appliance industry braced itself to meet demands for low noise, the problems posed by noise labelling and statutory noise limits became apparent. This has led to the development of standards for a unified approach to noise measurement. A brief survey of the development of standards for vacuum cleaners and the possible developments in regulations has also been presented.

Although an active interest in noise measurement at Hoover plc dates back to around 1978, the company has failed to improve its noise control knowledge and experience. It is a step towards remedying this situation that this project was undertaken.

CHAPTER THREE: NOISE REDUCTION METHODOLOGY: A LITERATURE REVIEW

Contents

3.1 Introduction

An international survey by Myncke (1985) revealed a rapid increase in institutes and societies in the field of acoustics since "noise abatement" became a much used expression. Since 1950, 10 institutions have been founded for such diverse purposes as research, the establishment of standards, noise control and measurement of the effect of noise on humans. As a result, there are many institutional gatherings for presentation and discussion of their work. Clearly, documentation and publication of papers has mushroomed, as shown in Figure 3.1. An on-line search of the ESA Information Retrieval Service Data in July 1985 disclosed 34636 citations for the word "noise".

The literature review of noise reduction principles presented in this chapter will be structured as follows:

3.2 techniques for identification of noise sources

3.3 methods of noise reduction and control

3.4 noise prediction techniques.

Noise and vibration problems can vary sustantially in their nature and severity, yet even the most complex can be represented as a block diagram thus,

SOURCE — PATH — RECEIVER

and the noise reduction treatments can be categorised as,

- source modification
- path or structural modification, which includes
  damping, vibration isolation, enclosures and barriers
- noise curtailment at reception.

Aston University

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Figure 3.1 Survey of the variation in the number of participants and papers of ICA and I-INCE as a function of time.
(from Myncke (1985))

Researchers generally agree that the first step of a noise control strategy is to determine the dominant noise sources. Crocker (1977), Brito (1979) and Nashif (1983) also state that the initial strategy should identify, rank and quantify these sources.

As noise solutions differ for each application, only those relevant to the key issues in a vacuum cleaner will be considered. The key areas of interest are the noise and vibration due to the bearings, electric motor and fan, the aerodynamic noise generated by the air flow through ducts and the exhaust, and the sound radiated by the outer surfaces of the structure.

3.2 Identification of noise sources

Ebbing and Hodgson (1974) and Crocker (1983) reviewed papers on identifying sources of noise. The former reviewed "classical" methods that do not require a thorough theoretical knowledge of sound propagation, and hence these methods prevent a proper evaluation. Crocker discussed modern techniques which have been discovered with the advent of digital computers. These techniques overcome the limitations associated with the "classical" methods.

3.2.1 Classical techniques

Human senses have always played an important role in identifying noise sources. Ever since the first machine was built, noise and vibration could be sensed directly by the designer's ears and fingers. Even today when sophisticated equipment is available, many automobile engines are inspected by a highly trained experienced mechanic who detects major problems simply by listening. This combination of human sensitivity and discrimination is not easily duplicated and hence, earlier attempts were aimed at improving sensitivity by using a stethoscope or a microphone-amplifier-earphone system. However, human perception does have serious limitations.

The senses are non-quantitative and cannot separate sound sources which are close together. Kryter and Pearson (1965) found that the ear is better able to distinguish and understand discrete tones than broad-band noise. Methods to examine broad-band noise were devised because there are many sources of this type, for example turbulent flow, wind and traffic at a distance. If part of a machine can be disconnected without affecting its operation, then the probable contribution of that part to the overall noise can be determined. "Selective operation" can be helpful in a complicated machine. In many cases, however, disconnecting some parts may alter the operation of the remaining parts and give misleading results. For example, removal of the fan from a vacuum cleaner motor alters the load on the motor which affects its dynamic characteristics.

"Selective wrapping" or enclosing different parts of a machine can obviate changes in loading, because the method does not require any parts to be disconnected. The enclosures can then be removed sequentially to determine the contribution of the known exposed area. Priede (1984) used this technique on a diesel engine and found it was effective in highlighting the relative contributions of the tappet cover, sump, inlet, etc. There are drawbacks however, the complex source may be too small or the enclosure seal may, by way of damping or stiffness, alter the structural characteristics. Electronic instruments are now available which overcome these limitations.

3.2.2 Fast Fourier Analysers

Because most signals are not pure tones but a complex combination, ie. broad-band, it is easier to evaluate these signals in the frequency domain using frequency analysis techniques. The usual method of obtaining the frequency spectrum of a signal is to pass it through a number of filters each with different centre frequency. Octave bands are common and have bandwidths such that the upper limiting frequency is twice the lower limiting frequency. The "third octave filter" is created by dividing the octave band into three equal logarithmic sub-divisions. Modern analysers are digital instruments which employ the Fast Fourier Transform (Fast is a relative term denoting the use of algorithms that minimise the computation necessary). The review of real-life problems by Bickel (1977) and Hundal (1985) reveal that high-resolution narrow-band frequency spectra are essential to pin-point discrete frequency components such as motor imbalance and blade passing tones.

The initial application of Fourier analysers was as extensions to the familiar classical methods. By changing the speed of a machine, frequency information can identify the resonances and critical speeds of a system accurately. Thien (1973) used Fourier analysis to extend the performance of acoustic ducts previously used in "selective wrapping". He placed one end of the duct in the vicinity of the complex source under test and a microphone to register the signal at the other end. The use of a single channel FFT analyser in sound level mapping around a machine can provide additional frequency information to identify the directivity of a source. "Sound level mapping" is particularly suited for graphic illustration of a noise problem in a large area such as a workshop. Lord et al (1980) found that sound measurements in a large room were influenced by the sound field because free-field or reverberant conditions effect different rates of fall-off.

<u>Near-field sound measurements</u>

This is a convenient and quick method of determining and ranking the major noise sources. However, the results should be treated as a guide because this technique has inaccuracies and limitations. The near field can be contaminated by noise from other nearby sources or, according to Crocker (1979), the sound measurement may not relate to the intensity because the particle velocity and sound pressure are only in-phase for idealised point sources and not complex sources. Nevertheless, near-field method was used effectively by Brito(1979) who demonstrated that results agreed well with sophisticated two-channel measurements. Although both near-field and mapping techniques can be undertaken with sound levels alone, the frequency information enhances source recognition.

Signature analysis

Not all sound and vibration sources produce a signal whose character is constant in time. Measuring variations in time is helpful because they highlight the structural activity with, for example, a change in the speed of rotating machinery during coast-down or start-up. A succession of frequency spectra presented one on top of an another, often called a waterfall display or signature map, clearly illustrates any critical speeds and structural resonances. Smiley (1983) found that this analysis has virtues for use as a condition monitoring tool. Signature analysis is normally associated with a multi-channel analyser because of the extra speed dependent sampling required. However, a cost-effective method was proposed by the writer (1986b) which utilised a single channel FFT analyser connected to a plotter. This paper is attached in Appendix 2.

Digital FFT analysers manipulate the data very quickly, and hence data capture of more than one channel was possible simultaneously. Bendat and Piersol (1980) contributed to the corresponding advance in theory, and proposed engineering applications. Most of the initial research was in the high technology industries such as aerospace, turbine and automotive, mainly because of the high initial cost of dual channel analysers. Hence a number of new techniques have since been devised which have their roots in these industries.

Surface intensity

This technique utilises simultaneous measurements of two signals: the sound pressure level measured close to this point and the particle velocity normal to the surface (by integrating the acceleration signal).

The cross-spectral density between the two signals is a measure of the surface intensity. This technique provides a means to identify, rank and quantify the source because the parts of the structure most responsible for the radiated noise are highlighted. Brito (1979) applied the technique to reduce noise of a sewing machine and found that sound power levels can be determined directly and as accurately as in a reverberation room using a precision method. Brito also found this method confirmed the ranking obtained by the near-field technique.

Acoustic intensity, (AI)

AI was mentioned as early as 1932 by Olsen. The recent interest in this technique is owed to the advent of digital signal processing. Sound intensity utilises a pair of phase-matched microphones which, with analytical algorithms, indicates the amount and direction of net flow of acoustic energy at a given point. Theoretical aspects are documented by Fahy (1977) and Chung (1978), and details of a commercially available device are given by Gade (1985). Many examples of use of this technique can be quoted, such as a bottle labelling machine by Gade et al (1983) and escalator drive by Kendig and Coakley (1986). Most researchers value the ease of this method because it can be applied in-situ to determine the sound power level and identify sound sources.

Coherence technique

Coherence is a measure of the cause and effect information from a system showing what part of the response is due to a particular source. This facility is common on two-channel analysers and can be applied to either sound or vibration measurements. Nicolas et al (1983) used the coherence between the vibration and noise signals to give a better understanding of the causes of noise generation in a chocolate plant while Priede (1984) used the technique to identify the noise sources in a diesel engine. Wang (1983) noticed that the physical limitations and the practical value of this technique were not well understood, and conducted experiments with known simple sound sources in an anechoic room. Wang, and later Hundal (1985), found that the limitations, related to the noise environment and the contamination of measurement, can render the coherence function inapplicable, especially over large distances.

Frequency response functions, FRF

FRF's are ratios in the frequency domain of motion parameters and the force. Usually, these are used when the system can be excited with a known forcing function. The system response is then measured and the ratio obtained. The ratios are,

dynamic mass	=	force / acceleration
impedance	=	force / velocity
dynamic stiffness	=	force / displacement
receptance	=	displacement / force
mobility	=	velocity / force
inertance	=	acceleration / force

The theory of vibrations using the mobility function is given by Bishop and Johnson (1980). A frequency response function can illustrate system resonances and give indication of damping. Practical applications include the analysis of tyres by Dunn (1972) and the car body by Desanghere (1985). The frequency response functions are normally used to determine system parameters and then enable analytical modelling, for example modal analysis which will be discussed later.

For all of the analyses, the conclusions - where the noise and vibration is coming from - depend upon the interpretation of the results. The more the analyst knows about the suspected sources the greater will be the confidence with which a final inference can be made. This review of papers has confirmed the writer's opinion that a number of source identification techniques should be used because each gives information on a different aspect of the source.

3.3 Factors influencing noise generation and reduction

Having discovered the major noise sources and ranked them in an order of significance or perhaps of priority, then the process of noise abatement can proceed. This process involves firstly, an understanding of the factors that influence noise generation, and secondly, the application of noise reduction principles to these factors. The most beneficial approach, and often the most cost-effective in the long term, is to reduce noise at its source. In the event that this approach is not effective or possible then other means must be sought. For example, modification of the noise path, its propagation and possibly protection of the receiver.

The study of major noise sources will be confined to the aspects of a vacuum cleaner, that is the,

- electric motor
- centrifugal fan
- air-flow and ducts
- airborne noise
- vibration and its transmission
- domestic appliances case studies.

3.3.1 The electric motor

According to Crocker (1973), research on motor noise and vibration is thought to have begun in the 1920s. It would seem that the principles that govern noise would presently be well understood. Indeed, the main parameters are known. Even so, as many of the noise generating parameters affect performance as well, noise has been given a lower priority in the design process. Noise sources in electric motors are broadly categorised as ventilation, mechanical and magnetic as shown in Figure 3.2.

Ventilation or windage

Ventilation noise is broad-band because it is due to air turbulence although some discrete frequency components are present due to the nature

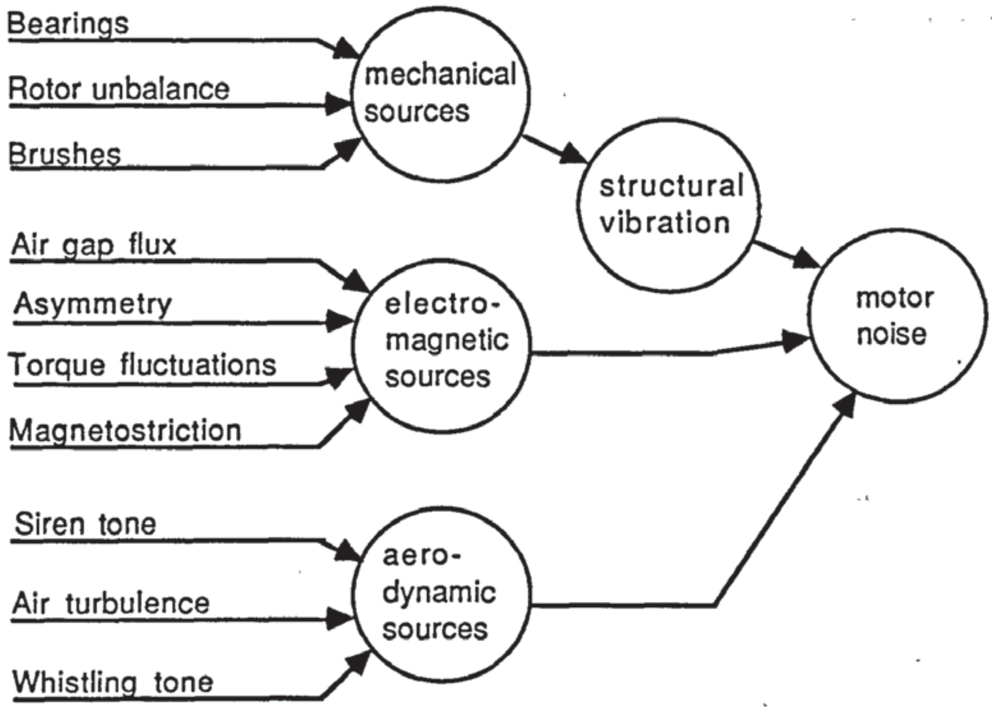


Figure 3.2 Elements of electric motor noise.

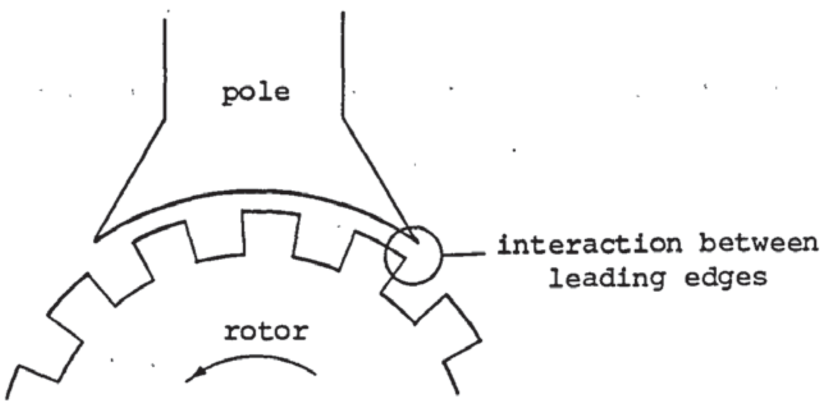


Figure 3.3 Interaction between rotor and pole.

of fan noise. It is generated by the cooling air flow which is necessary to maintain the rotor/stator temperature within design limits. Higher working temperatures could adversely affect winding insulation (special insulation is available but it would add to the unit cost of the motor). Ventilation noise is normally significant for high speed motors and because the development of motors has concentrated on achieving higher outputs from smaller and faster motors, ventilation noise has become more significant. Guinter (1979) mentioned that until recently this noise had masked noise from most other sources. Binks (1981) stated that ventilation noise is virtually independent of load. Drastic noise reduction of a broad-band nature have been reported by Teague (1975) who used efficient air inlet and outlet mufflers on large motors.

A "siren" is caused by the "chopping" effect of the air in the rotor/stator gap due to the leading edges of rotor teeth as shown in Figure 3.3. This sudden interruption of the radial flow of air as stator and rotor slots change their relative positions can be reduced by increasing the radial gap. This solution however, conflicts with the performance of the motor. This "chopping" effect can also be reduced by skewing either the rotor or the stator slots, but skewing increases costs.

Mechanical

The chief sources of mechanical noise are the bearings and the rotor out-of-balance. These two are inter-related since any unbalance will increase the cyclic load on the bearing and hence the vibration transmitted by them.

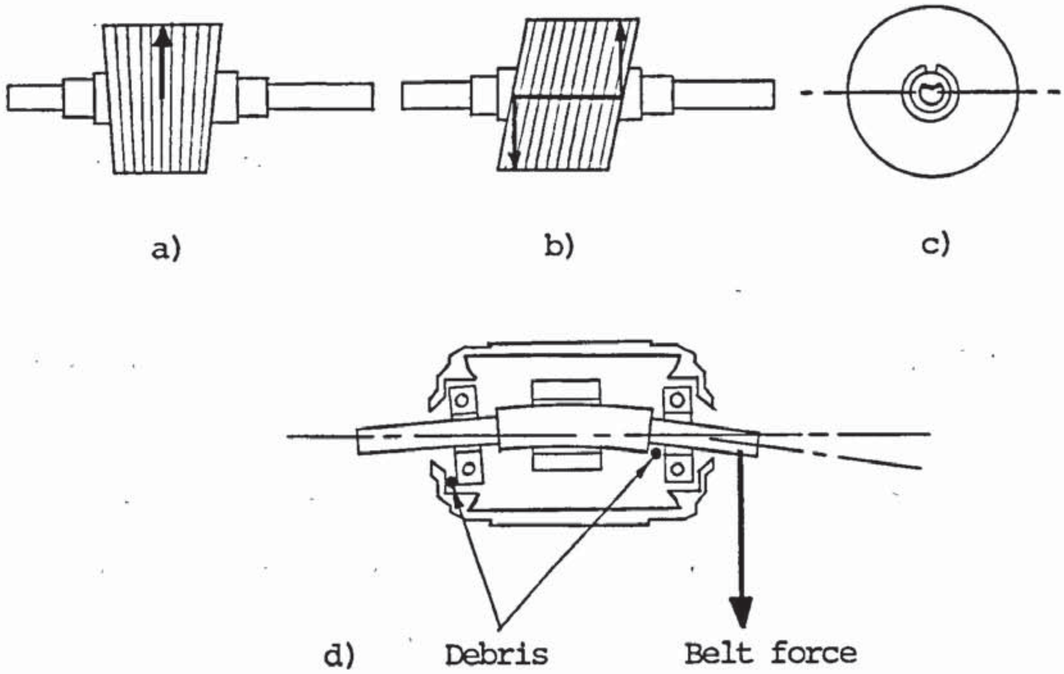
Bearing noise is not usually the major source of motor noise, but as noise generation depends on the type, quality, method of lubrication and installation, there comes a time in the life of a bearing when it can become prominent. Ball bearings are most common as they are generally quieter than the needle or roller type. Many theoretical studies have been undertaken to understand noise generation in a bearing. Rolling elements exert a train of concentrated loads on the raceways and any defects in the ball element or either race are known to generate vibration of a particular frequency. The relevant equations are given by Diehl (1973). From the equations, it follows that noise preventive measures would include good quality bearings and low rotor unbalance.

Rahnejat and Gohar (1985) found that ball passing frequency becomes significant under increasing out-of-balance of the shaft and Sunnersjo (1985) discovered that variation in ball diameter and waviness in the inner race causes vibration which can be significant. Some of these factors can be controlled, for example by a radial electro-magnetic bearing designed by Gondhalekar and Holmes (1984), but these would be restricted to special applications because of cost.

The second category of mechanical noises arises from the rotor shaft being out-of-balance, misaligned or bent. Although the armature shaft is usually dynamically balanced as a sub-assembly before it is built into a motor, there is always residual unbalance present due to some play in the bearing, production tolerances, shaft installation or shaft bending due to the magnetic pull as pointed out by Guinter (1979). Dynamic balancing of the shaft cures mechanical assymmetries of the type shown in Figure 3.4.

Electro-magnetic

Pomfret (1973) cited that the major source of magnetic noise is due to the vibration of the laminations which is set up by two separate phenomena. Firstly, the rotating field due to the magnetic flux produces radial forces which distorts the stator as shown in Figure 3.5. The rotor will effectively run unbalanced due to bending. Secondly, a phenomenon called magneto-striction causes the core-plates to expand and contract along the magnetisation axis as the flux rotates. If the exciting frequency coincides with a natural frequency of the stator or other element then dynamic amplification will take place and increase noise



a) unrandomised tapered lamina

b) skewed lamina

c) asymmetrical shaft stiffness

d) distortion of assembly

Figure 3.4 Mechanical dissymmetry as a source of vibration.

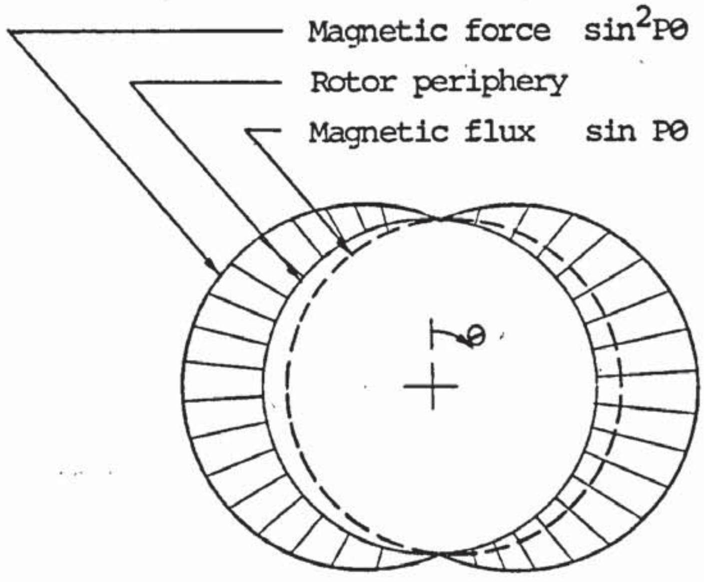


Figure 3.5 Magnetic flux and force distributions in the air gap of an electric motor.

generation.

A study by Watanabe et al (1983) enabled good approximations to be made from the assessment of Young's modulus of insulating layers of the impregnated windings. Cashmore (1983) analysed the magnetic forces and found that torsional vibration due to oscillation of the stator and rotor is caused by the tangential component of the magnetic force which sets up torque oscillations at twice line frequency. This vibration does not directly cause airborne noise, since no air is displaced, but noise is generated indirectly via the transmission of vibrational energy which excites the motor carcass. The principle of action/reaction prompted Fujimoto et al (1983) to introduce a second motor on the extended shaft of the first. The poles were arranged to counter-act the forces of the first motor thereby reducing vibration 100%. Applications are limited by cost to precision uses such as up-market turntables.

Nobody has attempted to predict accurately the noise of a motor. Binks (1981) recognised that although assessment of certain factors can be made, the vibration paths are numerous and therefore cannot be identified accurately. Consequently the noise radiated by the motor carcass cannot be predicted.

Many manufacturers produce quiet motors for specialist use. Typical modifications included,

a) liquid cooled motors which are reputed to reduce motor noise by almost 10-12 dB(A) by lowering the cooling medium velocity
b) reducing the quantity of cooling air, hence the ventilation noise, the resultant rise in winding temperature is maintained within tolerance limits by adding more copper and more core length
c) providing various types of enclosures and silencers. Although this solution adds cost, in many instances where a particular

cause cannot be identified, this is the only remedial action possible and it is known from experience to be effective.

Summary

There is no simple solution to reduce motor noise and there is no simple means to turn a standard motor into a "quiet motor". Moreover, Teague (1975 p47) argues that,

"it would be pointless to specify low noise levels for a motor which operates infrequently and for short periods of time".

Nevertheless, bearing quality, rotor out-of-balance, rotor/stator gap and prevention of sharp edges in the cooling air flow passages are important factors.

3.3.2 Centrifugal fan

A fan is a device for moving air. There are many types of fan but these can be divided into two categories: centrifugal and axial. Each type can be sub-divided depending upon the construction and blade configuration details. Eck (1975) provides a comprehensive textbook on fans and presents both the theoretical and practical aspects of fan design. There are three types of centrifugal fans as shown in Figure 3.6: these are backward curved, forward curved and radial blades. The suction fan in a Turbopower cleaner is a cross-flow centrifugal type with backward swept blades, and the review will be confined to this type. The aerodynamic noise from centrifugal fans comprises two components: harmonic and broad-band as shown in Figure 3.7.

Harmonic noise

The volute cut-off, the region of the fan casing closest to the rotating impeller blades, was identified by Neise and Koopmann (1984) as an active and significant region. The velocity profile at the impeller exit exhibits sharp minima and maxima. According to Finkelstein (1974), the maxima occurs between the blades and the minima at the blade edges.

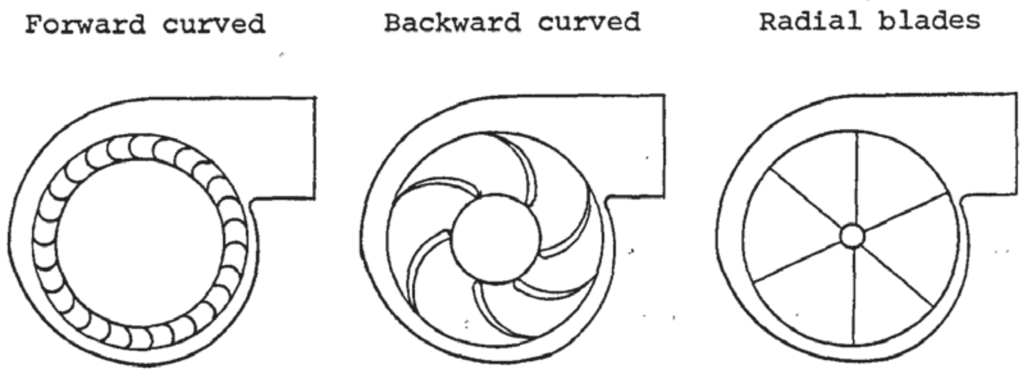


Figure 3.6 Examples of centrifugal fans showing types of blades.

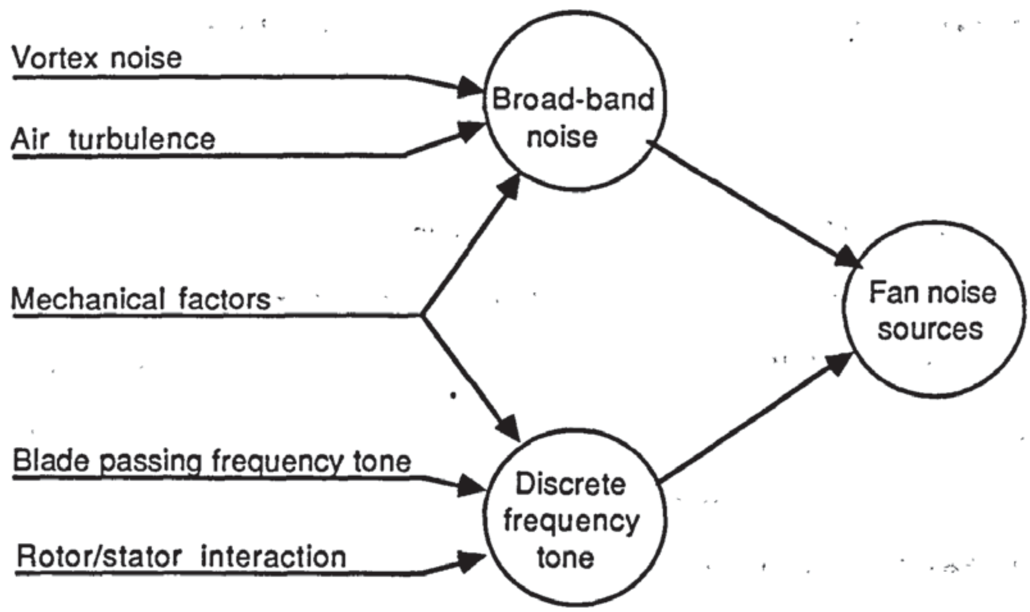


Figure 3.7 Elements of fan noise.

This non-uniform velocity profile produces strong pressure variations and generate sound at the blade passage frequency, BPF. Madison (1949) and Harris (1957) suggested that the velocity profile may become flatter with an increase in the cut-off clearance. The first experimental results were reported by Eck (1975) who also found that skewing the cut-off or the blades also reduced the noise related to the BPF.

Methods to reduce noise which involve modification to the fan have been presented in reviews by Neise (1976, 1982). Slots cut in the blades, Figure 3.8, enabled high-energy flow from the pressure side of the blade to the suction side in order to displace the point of boundary layer separation further downstream. However, the benefit was small; 1-2 dB(A). Evaluation of the variation in cut-off radius and clearance, by Neise and Koopmann (1980), revealed that these factors strongly influenced the sound power of the BPF tone. At the optimum value, fan efficiency increased and some reduction in the broad-band noise was obtained. Neise and Koopman (1980) replaced the cut-off region with the mouth of a tuned resonator tube. They reported a reduction in the BPF tone of upto 29 dB when the resonator length was a quarter-wavelength of the BPF.

The technique of unevenly spaced fan blades applies modulation principles to reduce the strength of the BPF tone by spreading the energy to the neighbouring sidebands as shown in Figure 3.9. Mellin and Sovran (1970) and Ewald et al (1971) explain the background and the theory which is based upon Bessel functions. Mellin and Sovran discussed the criteria for evaluating minimum tonal annoyance, and Ewald et al considered three methods for predicting the noise spectra resulting from non-uniformly spaced blades. Experimental results of asymmetrical spacing were reported by Krishnappa (1980) who concluded that although the BPF tone was reduced, in extreme flow conditions, that is away from the points of maximum fan efficiency, some of the sideband tones became higher than the initial BPF tone. The aerodynamic performance of these fans was unchanged when

Figure 3.8 Principle of slots in the impeller blades, as suggested by Embleton.

(from Neise (1976))

Figure 3.9 a) Bessel coefficients vs. maximum phase deviation ($\Delta\phi$).

b) Frequency spectra for the phase-modulated fan with $\Delta\phi = 3.85$.

(from Ewald, Pavlovic and Bollinger (1971)) compared with evenly spaced fans.

Broad-band noise

Broad-band noise is caused by turbulent airflow. According to EcK (1975), the principal sources are the turbulence of the incoming air, the turbulent boundary layers and vortex detachments from edges of the blades. If flow distortions due to the mainstream turbulence are on a large scale so that several blades pass through each eddy, then pure tones can be produced, and as these distortions become smaller the tones vary in frequency. Blade boundary layers produce high frequency components whereas mainstream boundary layer variations give rise to low frequency noise.

Modification of both boundary layers has been reported in order to reduce the noise. Moore (1973) experimented with the mainstream annulus layer. Moore considered this layer the most important source of distortion because it acted on the blade tip region where any distortion is most efficiently converted to noise. Moore bled off this annulus layer just upstream of the rotor tip so that the tip only saw a thin, even layer. Unfortunately, only limited noise reduction was achieved by this method, probably because the bleed system introduced new distortions.

Blade trailing edges give rise to eddies and vortices and Chanaud et al (1973) hyphothesised that if the edge was "removed" then turbulence may be reduced. Ideally the fan needs to be 100% solid on the leading edge and 0% at the trailing. This being impractical, they tried to eliminate the boundary layer by a porous fan blade and obtained a 5 dB(A) reduction but was accompanied with a drop in flow efficiency.

Blevins (1984) stated that predictions of vortex induced vibration were subject to considerable uncertainty but the influence of non-uniform inflow conditions which gives rise to broad-band fan noise were better understood. Barsikow and Neise (1979) presented experimental results which infer that excess noise is produced by a non-uniform velocity profile. The effect was an increase in the BPF tone and its harmonics.

Further attempts to reduce fan noise involved modifications to the fan scroll. Bartenwerfer et al (1977) packed acoustic lining on the outside of the scroll and reported noise attenuation up to 12 dB in the fan outlet duct. Both harmonic and random components of the noise were reduced. Yeager (1983) used a thin porous plastic material as a sound absorber and installed it as an integral part of the scroll (of a double-inlet centrifugal blower). A reduction of 2-3 dB(A) was reported. Another method utilising a series of Helmholtz resonators/absorbers tuned at low frequency was reported by Challis (1976). He built these absorbers into the periphery of the scroll and claimed 8 dB reduction at 63 Hz octave band and 10 dB at 125 Hz (suitable for large ventilation systems).

Summary

The significant factors in fan noise control are the volute cut-off radius and clearance between blade tip, speed and diameter of fan and the velocity profile at fan inlet. Noise can be reduced by Helmholtz resonators planted in the scroll, sound absorption on outside, and by optimal shape, angle and asymmetric spacing of fan blades.

3.3.3 The air-flow and duct noise

Air turbulence generally causes broad-band noise. Discrete tones appear due to fan characteristics and interaction with obstructions in the flow of air. The interest here is with the sources of noise within the air carrier systems, classified as follows;

- air collection
- air flow in ducts
- air exhaust.

3.3.3.1 Air collection

A vacuum cleaner has a great deal in common with a localised exhaust ventilation system. The latter aims to provide an effective means of collecting dust and hazardous fumes close to the source so that pollution of the general atmosphere is prevented. Moreover, sufficiently high airflow velocity must be produced in each case at the source of the pollutant to capture it.

Much research has been done to determine the important aspects of exhaust ventilation system design. Alden and Kane (1970) give detailed information on the design of exhaust systems. Some important factors were discussed by Socha (1979) such as minimising captive distance, adequate captive velocity and an even air flow distribution. Dust collection is therefore critical.

Dust collection is influenced by the velocity profile at the inlet. Fletcher (1977) and Garrison (1981) proposed formulae for the centreline velocity along the duct opening. Typical velocity profiles for plain and flanged inlet are shown in Figure 3.10. According to Fletcher (1977), the effect of the flange on the profile in front of the inlet is threefold: it provides a larger effective flow area, a lower entry pressure loss and hence lower power requirements and better velocity distribution at the front. Garrison and Byers (1980) discovered a pronounced vena contracta for a plain inlet and only a slight one for a flanged inlet. This implies that for an equal air-flow rate the plain nozzle would require more energy; approximately 50% more in the form of a negative static pressure. The latter is particularly important factor as it affects nozzle noise.

Garrison and Byers (1982-2) considered noise generation at the nozzle inlet. They concluded that a flange increased the noise levels in front of the nozzle and that the variation in the length of the inlet section of the nozzle had little effect on noise characteristics. They also suggested the overall noise levels may be reduced by aerodynamic "choking" because it inhibits the propagation of high frequency turbulence noise within the nozzle flow. (Choking refers to the maximum flowrate obtainable for the nozzle).

Figure 3.10 Velocity profile for air entering a round duct a) with and b) without a flange.

(from Socha (1979))

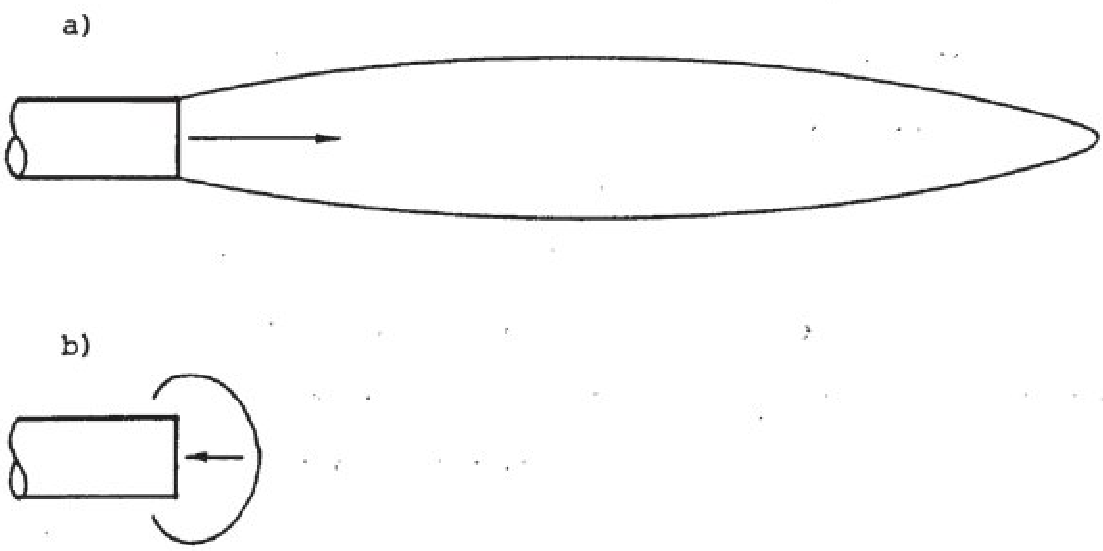


Figure 3.11 Velocity profile for air flowing a) outward and b) inward for a round duct.

The effectiveness of a nozzle, as regards flow distribution, can be investigated with a trace aerosol as reported by Burgess and Murrow (1976). This technique highlights the flow weakness around the inlet nozzle of a vacuum cleaner. Nozzle performance is important because the efficiency of the cleaner is dependent upon both the dust arousal (or agitation) and dust capture. It is interesting to note that Alden and Kane (1970) believe that for similar flowrate, the velocity profile of an air jet, that is air flowing out, is forty times that of suction as indicated in Figure 3.11. So, a jet of air may be better in disturbing the dust in the carpet.

3.3.3.2 Airflow in ducts

Sound is transmitted effectively along a duct, but is attenuated in its passage. General principles of sound attenuation in ducts will be reviewed as these are thought to be significant even though the air ducts in a vacuum cleaner are of relatively short length.

Mason (1969) concluded that sound is generated by the vortices shed from obstructions, sharp edges etc., in the air flow and that sound pressure levels show marked variations with their position in the duct. The latter is a result of standing wave interference, like "organ pipe resonances". Corrugated walls of a duct, similar to the flexible corrugated tube used on vacuum cleaners also generate sound. Petrie and Huntley (1980) had some success in identifying the controlling factors but they could not predict the occurrence or the intensity of the "whistle" in the corrugated section. Nevertheless, they believed that an axial mode was being excited and could possibly be cured by changing the turbulence structure. No suggestion was made as to how this might be accomplished.

Changes in the cross-section of the duct are deterimental to noise reduction. Morfey (1969) stated that in an irrotational flow there is no production (or dissipation) of acoustical noise. So, a duct of a constant cross-section without bends is best. Changes in the cross-section of a duct introduces turbulence which gives rise to rotational flow and eddys. The air-flow in the Turbopower cleaner duct is expected to be turbulent because the Reynolds number is greater than 70000. According to Cartwright (1985) turbulence exists if this number is in the range 10000-80000. Szumowski and Piechna (1983) illustrated that a sudden expansion within a duct can also cause self-excited oscillations which give rise to acoustic waves and then lead to an increase in the overall noise of a jet by upto 30 dB; similar dramatic effects can be expected at contraction. A sharp bend was shown by Peistrop and Wesler (1953) to lower the attenuation at particular frequencies.

Some noise in ducts is attenuated by the natural absorption due to the walls. This attenuation can be further improved by adding absorption medium on the inside walls of the duct. Duct resonances can actually increase the sound level within the duct. Dean (1975) suggested using circular ducts wherever possible because their fundamental resonant frequencies are much higher than rectangular ducts.

3.3.3.3 Air exhaust

Duct noise can be reduced by silencers of which there are two types: absorptive and reactive. The fcrmer are straight-through devices and comprise flow channels lined with absorptive material. A silencer system can be represented as in Figure 3.12. A review by Ericksson et al (1983) identified a recent change in silencer design due to the advent of computers. Bender (1973) analysed silencer design by transfer matrices, but his results of the insertion loss (the term used to evaluate the noise attenuation of a silencer) exhibited a great deal of scatter. Similar findings were later reported by Doige and Thawani (1979) who applied transmission matrices. These researchers found that the characteristics of a silencer were dependent upon the sound spectrum of the source. For this

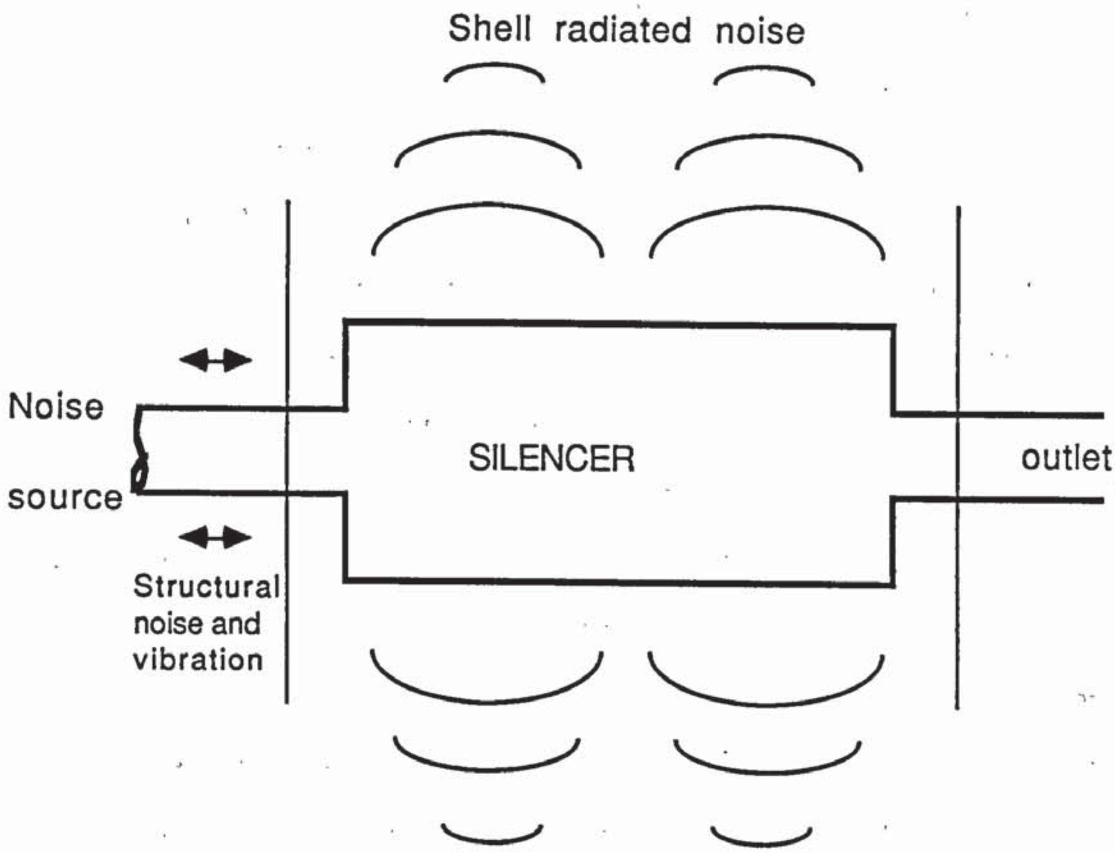


Figure 3.12 Noise propagation from a noise source to an outlet via a silencer.

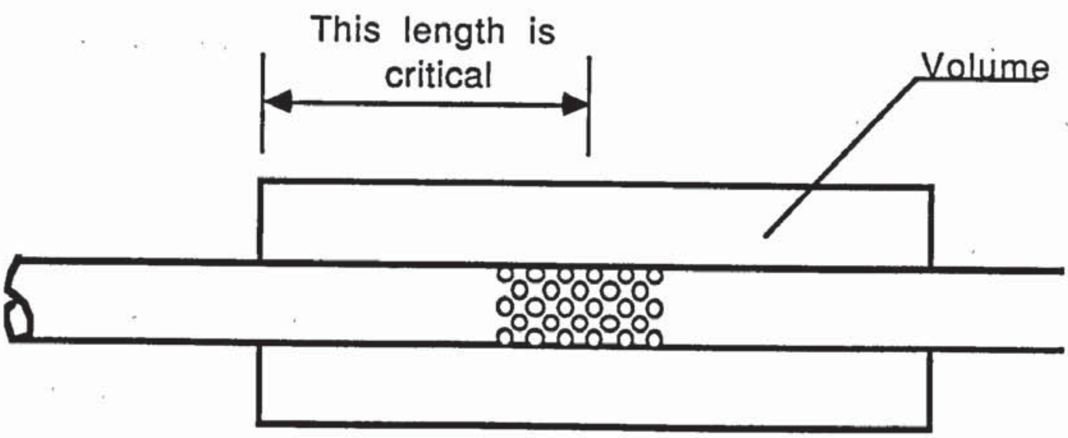


Figure 3.13 A simple resonator silencer.

reason the optimal silencer has to be "tuned". So, although a silencer can be designed using a computer, a practical evaluation is still required. A made-to-measure silencer is not yet possible.

Absorptive silencers are usually for attenuation of high frequency noise because of the limitations of the materials used. Alternative methods to extend this frequency range have been tried. Abdelhamid (1978) and Soderman (1981) evaluated parallel baffles and baffles incorporating resonant cavities. These solutions, however, are only suitable for large ducts which do not carry dust. The straight-through variety, shown in Figure 3.13, is considered to be suitable for dusty air flow. Shenoda (1982) evaluated such a silencer and obtained 16-20 dB attenuation.

The second type of silencer is the reactive and utilises a change of direction of the air flow to reflect some sound waves back and forth within an expansion chamber. Reactive silencers are unsuitable for vacuum cleaners because the dust particles will accumulate in the "dead flow" regions of the chamber and clog the silencer.

Summary

The inlet velocity profile is important for an efficient suction "nozzle" for a vacuum cleaner and regions of either high or low velocity around the nozzle should be avoided. The cross-section of the duct should be constant: unavoidable changes in direction or obstructions should be few, streamlined and gentle not sudden. A silencer can be fitted to improve noise attenuation. Although there are many types of silencers, the straight-through variety comprising holes and absorbent material is best suited for a vacuum cleaner.

3.3.4 Airborne noise

The airborne sound can be attenuated by,

- cancellation
- insulation

- absorption

Active cancellation works by the principle of superposition of two signals to cancel one another out. Mangiante (1977) and Warnaka (1982) reviewed the work on active cancellation. This technique, however, is not suitable for reducing the noise of a vacuum cleaner primarily because of cost.

Sound insulation, measured by the sound transmission loss, is achieved by preventing the escape of sound waves. Sound absorption, measured by the absorption coefficient, occurs where a proportion of the sound incident on a material is absorbed within that material. Therefore, the level of sound transmitted or reflected is reduced. Insulating materials are non-porous, dense and usually have structural properties whereas materials for the absorption of noise are porous and relatively lightweight.

Harris (1957) stated that effective noise reduction is commonly obtained by a combination of these functions. For example, Venghaus et al (1985) developed absorber combinations using metallic foils while Byrne (1984) used porous blankets. The influence of damping on acoustically excited panels was researched by Jones and Trapp (1971) and Hanson and Hampel (1984). Their results showed that damping reduced the vibration amplitude of the panel and therefore reduced the radiated noise.

Acoustical details are available for ceramic and building materials but there is a lack of such data for plastic materials. The problem is that there are a vast number of different types of plastic, and their trade names which run into hundreds, do not always convey the type of material. According to Graham (1986), although sound transmission loss data has been determined for some plastics, many manufacturers and suppliers are unco-operative because of commercial confidentiality.

The sound transmission loss (STL) of a material can be measured but requires special facilities. Guy et al (1985) discussed the effects upon the STL of physical parameters such as material properties, panel size, room parameters and the mounting orientation. They reported of conflicting results achieved by various researchers and these findings were confirmed by Nilsson (1985) in his summary of a Round-Robin test series. Warnock (1982) evaluated the ISO 140 standard and concluded that a support frame used for mounting the test material was critical, and the effects of the particular frame should be established. Experiments to adapt the ISO standard to a small, low-cost test rig suitable for small panels were done by Papanikolaou and Trochides (1985). They found that the STL obtained by using a small test rig were unreliable at low frequencies.

Orifices in a panel, reduce the STL of a panel considerably. Formulae to predict the effect of circular and slit shaped apertures were derived and evaluated experimentally by Gomperts and Kihlman (1967). They found that an opening lowered the potential of noise reduction due a panel as indicated in the Figure 3.14. Crucq (1981) considered the noise escape through openings in the body of a washing machine and found that high frequency noise had been hardly attenuated.

Summary

Sound attenuation due to partitions is effective and optimum when no apertures are present and may be enhanced by the addition of sound absorbing material.

3.3.5 Vibration and its transmission

Beranek (1960) stated that vibration is caused in two ways: structurally as a result of fluctuating mechanical forces and acoustically by the impinging sound waves. The response of structures to vibration is a complicated process to analyse, theoretically or experimentally. Many types of vibration such as flexural, torsional and axial may interact to produce this complex problem. Consequently, like noise, vibration should

Figure 3.14 Effect of enclosure openings on potential noise reduction.

(from Ghering W'L,'Reference data for acoustic noise control',Ann Arbor Science, 1978, USA) be prevented at source or very close to it.

Motor vibration is transmitted to the rest of the structure via numerous paths. Typical transmission paths for the Turbopower cleaner are,

- motor armature to bearing housing in the motor casing
- motor casing to motor mounting in the cleaner chassis
- agitator thread guard to its mounting in the chassis
- chassis to the attached sub-assemblies, especially the bag housing covers and the hood.

A system for dissipating or absorbing vibrational energy, known as an isolator, may be inserted in the path to reduce vibration transmission. Many textbooks, such as Den Hartog (1956) and Bishop and Johnson (1980), present vibration isolation theory for classical problems in the form of "free" and "forced" vibration response of single and multiple degree-of-freedom systems to usually one main variable forcing frequency. A fixed foundation is usually assumed. The simplest solution is to avoid operating a machine close to its major resonant frequencies.

In many applications however, both the "machine" and "foundation" are non-rigid. The process of designing an optimal isolator is then rather complex. Muster and Plunkett (1971) considered such a situation but only for small oscillations, that is within linear boundary conditions. Non-linearity in a structure can be identified and allowed for in transfer function measurements using the Hilbert transform, see Okubo (1982). Lyon (1983) extended the transfer function concept to represent the total system, that is machine-vibration-mounts-foundation-structure-noise. Such a transfer function comprises several components but with the overall transmission being expressed as a simple product. Thus, sub-structuring of a system can be used to "tune" the isolator.

Different approaches for sub-structuring have been reported. Lyons (1983) applied statistical energy analysis to predict vibration in a steady state situation. Analytical treatment using mobility equations was first demonstrated by Sykes (1971). These studies did not include any experimental verification. This was later presented by Plunt (1981) and DeJong (1983) who analysed the vibration of a sewing machine. Their results found some disagreement between theory and practice. Lyons (1983) stated that mechanically induced vibration cannot be predicted with accuracy because the excitation takes place in many different parts. Therefore, the isolator cannot be optimised by design alone and requires measured data. Subsequently, better correlation was shown by Hemmingway (1986) who analysed vehicle sub-systems using mobility matrices and measured data.

Bolton-Knight (1971) found that the point of attachment of the motor to a chassis is important if optimal isolation is to be achieved. The transmission of vibration is low when the motor is mounted at nodal points, that is points of least oscillation in a resonant mode. He also suggested that further reduction in vibration can be achieved if the mounting points on the chassis are located at areas of low mobility. In the case of the Turbopower cleaner the mounting points were predetermined by other considerations.

Elastomeric materials, especially rubber, are commonly used because these are more readily mounted and have superior inherent damping properties. Rubber elasticity is due to its deformation under application of load but rubber, like water, is virtually incompressible. Consequently, it is normal to use rubber in shear, which results in greater deformation than in compression. Properties of synthetic rubbers and polymers can be modified by compounding, ie the addition of fillers like carbon black and oil, or changing the "cure" temperature. The dynamic characteristics of these materials can differ appreciably from the static values and depend upon temperature, frequency and amplitude of the force of excitation. Although transfer functions have been used in theoretical analyses, for example Sykes (1971), there is no reference to their use in practice, for example in measurements of the dynamic properties reported by Ianniello and Maffei (1982).

Summary

An isolator can reduce the transmission of vibration. The isolator can be designed easily but requires dynamic properties of the motor and chassis for optimisation. Nodal mounting points will enhance reduction in the transmitted vibration.

3.3.6 Sound reduction at the receiver

Ear-muffs and ear-plugs have proved successful in protecting the operator from high levels of noise. However, in the case of using a vacuum cleaner, for which the duration of use is relatively short, ear-muffs or ear-plugs are considered to be inconvenient and prone to mis-use and therefore not a proper solution.

3.3.7 Case studies of household appliance noise reduction

There have been very few case studies reported in the literature. Jackson and Leventhal (1975) presented third octave noise measurements for a variety of household appliances without any analysis, while Betzhold and Gahlau (1981) summarised, very briefly, the application of intake and exhaust silencers and sound barrier mats to reduce the noise level of a suction type cleaner.

An important project was conducted by Tree and Uffman (1973) who described noise reduction of a cannister-type vacuum cleaner. The methodology adopted by Tree and Uffman is worth describing because it was similar to the one planned for this project (which was arranged prior to reading their paper). Briefly, the methodology was,

1. use of existing performance specifications, as baseline values
2. identify noise sources,

Sound measurements were made in the anechoic and reverberation rooms. Experimental procedures included;

- directivity pattern in anechoic room,
- near-field noise measurements at distance of 50mm.,
- vibration analysis of the cover of the cleaner,
- narrow-band frequency spectra,

3. reduce noise levels in two parts,

   a) attempt to reduce noise at source, and

   b) to alter the path along which the noise travels.

   Areas considered were,

   - vibration isolation, four rubber isolators reduced noise in the range 300-2000 Hz,
   - fan-rotor design, the blade passing frequency tone was reduced by irregular spaced blades,
   - addition of sound absorbing material reduced noise in the range 1-5 kHz,

4. noise test to evaluate the combined benefits.

3.4 Noise prediction or estimation techniques

The sound power generated by some common sound sources, such as the electric motor and fan, can be estimated. From the total sound power, the sound pressure level at any point in the medium through which the sound waves propagate can be estimated. The effect of sound attenuation due to partitions, barriers or enclosures can be approximated. However, the sound that "travels" as vibration through structures to re-appear elsewhere as noise, cannot be estimated with any certainty at the design stage.

3.4.1.1 Electric motor noise prediction

Lance (1977) collated the sound levels for industrial motors. However, there is a lack of similar information for fractional horsepower motors of the type used in household appliances. In this case, the sound level in third octaves of existing motors should be used as a starting point. The frequency of discrete tones can be predicted, but their magnitude, however, cannot be estimated accurately.

3.4.1.2 Fan noise prediction

The following empirical equations, derived from experimental data by Beranek (1971), are helpful to estimate fan noise. The appropriate equation is selected depending upon the known parameters of the fan.

$$SWL = 77 + 10 \log Kw + 10 \log P \quad \text{dB}$$

$$SWL = 25 + 10 \log Q + 20 \log P \quad \text{dB}$$

$$SWL = 130 + 20 \log Kw - 10 \log Q \quad \text{dB}$$

where, Kw - rated motor power, kWatts

P - static pressure developed by the fan, mm water gauge,

Q - volumetric air flow rate, cubic metres/hour.

Graham (1972) and Erskine and Brunt (1975) proposed specific equations which, under certain circumstances give better results. A single value of the sound power level is obtained by using the above equation which is then converted into octave bands. This conversion, which is added to the overall SWL value, is different for each type of fan as shown in the table on the next page. Also shown is the contribution of the blade passing frequency increment, BFI, which can be the most prominent source of annoyance. Consequently, the BFI is added to the octave band that contains the blade passing frequency.

3.4.2 Prediction of sound radiation and its attenuation

Sound radiation depends upon many factors. For example, whether the source is hung in space or placed on a floor, and the characteristics of

Type of fan	BFI	Octave band centre frequency (Hz) 63	125	250	500	1000	2000	4000	8000
Centrifugal fan;									
radial blades	5 to 8	-3	-5	-11	-12	-15	-20	-23	-26
backward curved	3	-4	-6	-9	-11	-13	-16	-19	-22
forward curved	2	-2	-6	-13	-18	-19	-22	-25	-30
Axial fan	4 to 6	-7	-9	-7	-7	-8	-11	-16	-18
Mixed flow	6 to 7	0	-3	-6	-6	-10	-15	-21	-27

All values in dB

the sound field. These factors are considered in the 'Handbook of Noise Control Guidelines for Hoover Design Engineers', see Appendix 4. The handbook includes and discusses the mass law which is important in determining the sound transmission loss of a panel. Noise attenuation due to partitions, full or partial enclosures are also discussed in the handbook together with the detrimental effect of openings in the enclosure walls.

The procedure for noise prediction can be summarised step-by-step and tabulated as shown in Table 3.1.

3.4.3 Vibration related sound

Recently, as a result of new analytical techniques and powerful computers, there have been great strides towards the estimation of the dynamic properties of a system. There are many finite element packages which can determine the stresses, displacements and modes of vibration of a structure at its design stage. The advantage of an analytical model is that results can be predicted before building a prototype. Therefore, various design alternatives can be evaluated quickly. Schmidtberg and Pal (1986), among others, concluded that the main disadvantage was damping

	Octave band centre frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000
Predicted or measured sound power level of the fan.	....	....	....	....	....	....	....	....
Fan noise adjustments eg. due to fan type	....	....	....	....	....	....	....	....
BF Increment	....	....	....	....	....	....	....	....
Result; noise of fan	....	....	....	....	....	....	....	....
Predicted or measured sound power level of the motor.	....	....	....	....	....	....	....	....
Result; noise of fan and motor	....	....	....	....	....	....	....	....
Attenuation due to motor assembly, etc	....	....	....	....	....	....	....	....
Result;	....	....	....	....	....	....	....	....
Attenuation due to cover or an enclosure Effect of any orifices	....	....	....	....	....	....	....	....
Result;	....	....	....	....	....	....	....	....
Attenuation due to a known muffler, duct, dust-bag, etc.	....	....	....	....	....	....	....	....
Result;	....	....	....	....	....	....	....	....
Attenuation due to distance.	....	....	....	....	....	....	....	....
Result;	....	....	....	....	....	....	....	....
........other treatments........								
Final estimated noise level.	....	....	....	....	....	....	....	....
	All values in dB							

Table 3.1 Procedure for noise level estimation.

because damping was very difficult to estimate. Consequently, the predicted response differed substantially from the actual.

Modal analysis enables an "animated" illustration of structural vibration modes. The results of modal analysis are dependent upon the reliability of the measured data. Nevertheless the technique is cost-effective because it can be applied to structures which are too difficult or costly to model analytically. Modal analysis is an experimental technique and Ewins (1984) and Patrick (1985) described the practicalities of acquiring valid data.

Ibrahim (1985) reported that modal analysis had limitations when a structure has a number of closely coupled modes. Ewins (1984) highlighted another limitation. For lightweight structures, the accelerometer causes a change in the mass and the stiffness in the vicinity of the attachment point which subsequently alienates the response. A modified form of modal analysis was presented by Brown and Allemang (1978) whereby a microphone was used for the response and an instrumented hammer for the excitation. They also describe another version using a small known mass that is attached and the corresponding shift in the frequency spectra is used in the analysis to determine the mode shape. There were however, some limitations with regard to the point chosen for the response.

As modal analysis is experimental, it has limitations in that it only analyses an existing structure. Recent advances have been in the integration of analytical and experimental methods. This has improved the accuracy of predictions. Structural modification system, SMS, is an analytical extension of modal analysis and determines the effect on the dynamic response due to hypothetical changes. Holmer and Lilley (1983) used modal analysis and SMS to predict accurately the vibrational response to minor modifications. With the advent of cheaper, faster and more powerful computers, Ramsey (1983) envisaged the introduction of these facilities to the desk top computer.

Figure 3.15 categorises the state of knowledge of a system with respect to the appropriate analysis technique. The above mentioned analyses can be used to optimise product design as illustrated in Figure 3.16. Potential noise and vibration problems can be highlighted and treated within this test cycle.

3.5 Review

This chapter reviewed the noise source identification techniques, discussed the main factors of sound generation for primary noise sources in a vacuum cleaner, and highlighted the principles and the application of noise reduction treatments. It was found that sound emission can be coarsely estimated for the motor and the fan but an accurate prediction of vibration related sound is not yet possible.

From the case studies which were reviewed, it was found that the approach to a noise problem is standard. There was, however, no universal remedy, and each problem demands an individual solution.

Figure 3.15 Catagorising the state of knowledge with respect to analysis technique. (from Holmer and Lilley (1983))

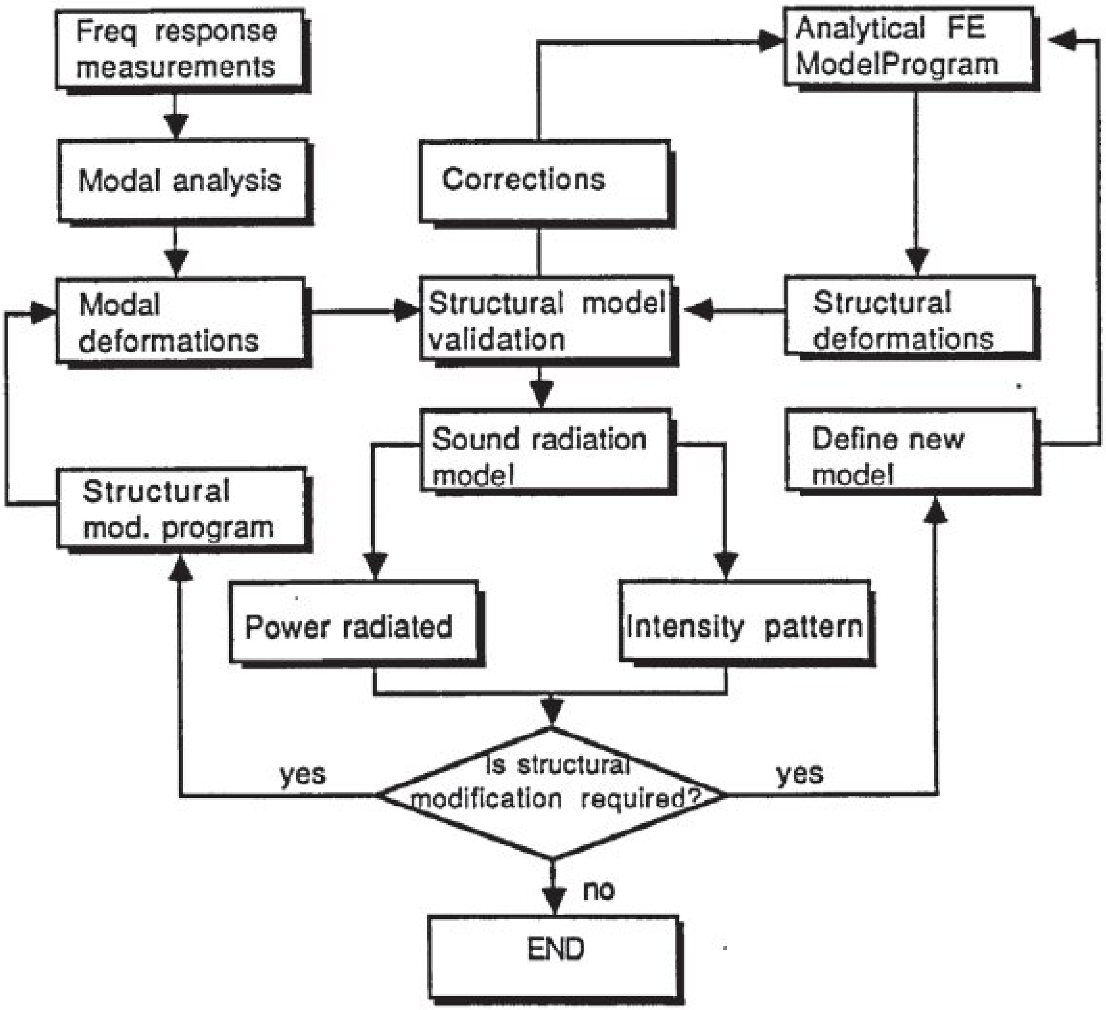


Figure 3.16 Optimisation of noise control by modal and finite element anlyses with structural modification program.

CHAPTER FOUR: THE INSTRUMENTATION AND MEASUREMENT PROCEDURE

Contents

4.1 Introduction.

From the noise source identification techniques reviewed in Chapter 3 the following basic properties required measurement;

- general sound level
- vibration response characteristics of the major components
- frequency characteristics of the sound and vibration levels
- out-of-balance of rotating elements
- general parameters of cleaner performance.

The instrumentation and the measurement procedures adopted to investigate the noise sources of the cleaner are discussed in this chapter. Before making a measurement, it was necessary to calibrate each instrument. Calibration ensured that the complete measuring system was checked for any inaccuracy or malfunction. The sound measuring equipment was calibrated quite frequently because the sound levels were found to be dependent upon ambient conditions. This chapter commences with a brief description of the Turbopower cleaner and its operation and closes with the outline of the programme for experimental work.

4.2 Description and operation of the Turbopower cleaner

The subject of this research project was the Hoover Turbopower hardbag upright vacuum cleaner. Photographs of the cleaner, the motor, armature and the suction fan are presented in Appendix 1.

4.2.1 Description

The Turbopower cleaner is known as a "dirty-air" type because the air with the dust particles and debris passes over the suction fan. The air then deposits the dust in the paper dust bag and is finally exhausted to the atmosphere. (In the clean-air system the dust is deposited in the dustbag before the air passes over the suction fan). The Turbopower cleaner consists of a box section upright, which comprises the dust bag housings and the handle, and a foot or base which houses the agitator, motor and the airflow inlet. The upright box section is connected by hinges and bellows to the base.

A small a.c. universal motor, the EURO, is fitted as the prime mover for this cleaner. It is located transversely in the cleaner base and mounted on two rubber bushes. The larger rubber bush on the suction fan chamber also acts as a seal for the air flow passage; a smaller bush is mounted on the belt end. The agitator is driven via a rubber belt which runs on one end of the armature shaft.

The motor comprises two fan systems. One fan generates sufficient air-flow to cool and maintain the temperature of the windings within tolerance (18-70 degrees Centigrade) and the other fan generates the air-flow and suction needed for dust pick-up.

The inlet duct envelopes the agitator, except for an opening underneath the cleaner base where the agitator bristles brush and beat the carpet for improved dust-pick-up. The outlet from the suction fan is connected to a corrugated type of bellows which allows angular movement of the handle, then to an air duct or top fill tube which is finally connected to the dust bag. The air duct and dust bag are housed in the cleaner upright section. The foot of the cleaner has a nozzle height adjustment lever which facilitates the cleaning of different heights of carpet pile.

4.2.2 Cleaner operation

When the Turbopower cleaner is in operation, the dust in the carpet is disturbed by the action of the agitator bristles and sucked up with the suction and the air-flow into the inlet duct. The air-flow rate is sufficient to ensure a speedy transport of the dust particles along the ducts into the dust-bag. The bag is fabricated from a paper material having fine pores to allow the air to pass with little resistance. The air then flows out via a gap all around the periphery of the front and back bag housing covers.

4.3 Measurement of sound pressure and power levels

4.3.1 Measurement of the sound pressure level, SPL

The pressure variation due to a sound wave is the basic quantity to be measured. BS 4197 (1967) requires precision condenser microphones to measure sound in a laboratory. At Perivale, two condenser microphones, B & K type 4165, were available.

One microphone was fitted to a portable sound level meter (SLM), B & K type 2203, which met the requirements of IEC 651 (1979) giving an analog reading of the overall noise level in decibels in direct (linear) or A-weighted scales. Octave frequency analysis of the measured sound was obtained by using an octave filter set, B & K type 1613, which enabled sound pressure levels to be measured for each octave band from a centre frequency of 63 Hz to 8000 Hz. The second microphone was connected to the input of B & K type 2107 frequency analyser. The output, linear or A-weighted, from this analyser was input to the Nicolet FFT analyser. With this set- up, the overall sound level was read on the B & K analyser whilst the narrow-band and third octave values were obtained from the Nicolet unit.

Sound measurements at the Advanced Dynamics laboratory at Aston University were mainly for narrow-band frequency analysis using a dual channel analyser. The analyser, a HP 3582A, enabled correlations to be established between the noise of the cleaner and the vibration response of a panel. The noise was detected by a B & K type 4165 microphone.

4.3.2 Measurement of the sound power level, SWL

The sound power level is determined from the measurement of the sound pressure level. A method for determining the sound power of household appliances is given in IEC 704.1 (1982), which is summarised in section 4.3.5.

4.3.3 Calibration of sound measurements

Calibration was required for both the sound pressure and sound power measurements. Microphones were easily and conveniently calibrated by using a hand held calibrator. The microphone was inserted into the pistonphone, which produced a known sound level. The reading on the sound measuring device was adjusted accordingly to calibrate the system.

To calibrate the measurement of the sound power level, a source of a known sound level is required. Two standard sound sources were available at Perivale. A Reference Sound Source, B & K type 4204, emitted a constant sound level of 92 dB(A) re 1 pW, with components over a wide frequency range and a B & K Sound Power Source type 4205, which had the facility to vary the sound power output from 40 to 100 dB(A) re 1 pW. Both sound sources were precision manufactured to ISO 3741. The variable sound power facility was ideally suited to determine the sound power level of a source by the comparison method.

4.3.4 Acoustic environments

Perivale acoustic room

The internal surfaces of the acoustic room had sound absorbing tiles except for the floor, which was plain wooden chipboard raised on joists. The acoustic treatment was complemented by a well fitted double-panel door. Power leads and the signal cables passed through a hole in one wall. The size of the acoustic room is rather small, being 2.37m x 2.38m x 2.335m high giving a total volume of 13.2 m^3. As a result of the small size, the temperature within the room rose appreciably during 20 minute running of an appliance, so an extractor fan was fitted in the ceiling. All sound measurements, however, were done with the fan switched off.

Clause 4.2 of ISO 3743 requires the sound measuring room to be between 70 m^3 and 300 m^3. The Perivale acoustic room is therefore too small. According to Kristensen (1976) the size of the room influences the frequency characteristics of the sound measurements. The absorption due to the air in a large room, greater than 300m^3, becomes large at frequencies above 4 KHz, whereas a small room, below 70 m^3, shows low accuracy for measurements of low frequency noise. Details of construction, size, length-width ratio and shape are discussed by Kristensen. All these parameters affect the reverberation time which needs to be adjusted to the values specified in ISO 3743.

Anechoic room, National Engineering Laboratory

Hoover plc, through their membership of NEL's Noise Control Club, was entitled to use noise measuring and signal analysis facilities at the NEL. The anechoic room was made to the specifications quoted above. It was equipped with precision equipment and facilities to make measurements of the sound power using the direct method, which will be discussed later. These were used to calibrate the sound measured in the acoustic room at Perivale.

4.3.5 Summary of the IEC Test Code 704.1

This standard was first drafted and discussed in 1974. Part 1 was published in 1982 and comprised general requirements and part 2.1, which was published in 1984 as an uncorrected proof, covered the particular requirements for vacuum cleaners.

The IEC 704 test code applies to all electric appliances for household and similar use and their accessories, run on batteries or mains. It outlines the methods of engineering accuracy for determining sound power levels of the airborne acoustical noise. Other factors such as frequencies of interest, operating conditions, location of the appliance during test and units of sound power are stipulated. Two measurement methods are described in the test code, namely the direct and the comparative. These methods are different in the context of the required acoustic environment and the number of measurements required.

Direct method

This method can only be used for measurements in a standard test environment. In the free-field case, the sound power level of a source is determined from the time-averaged sound pressure levels recorded on the measurement surface and the area of this surface. The measurement surface which applies to an upright cleaner placed on a horizontal reflecting plane is indicated in part 2 of this test code as a hemisphere of radius 1.5m with ten microphone positions as shown in Figure 4.1. The facility to adopt the direct method was only available at NEL, East Kilbride.

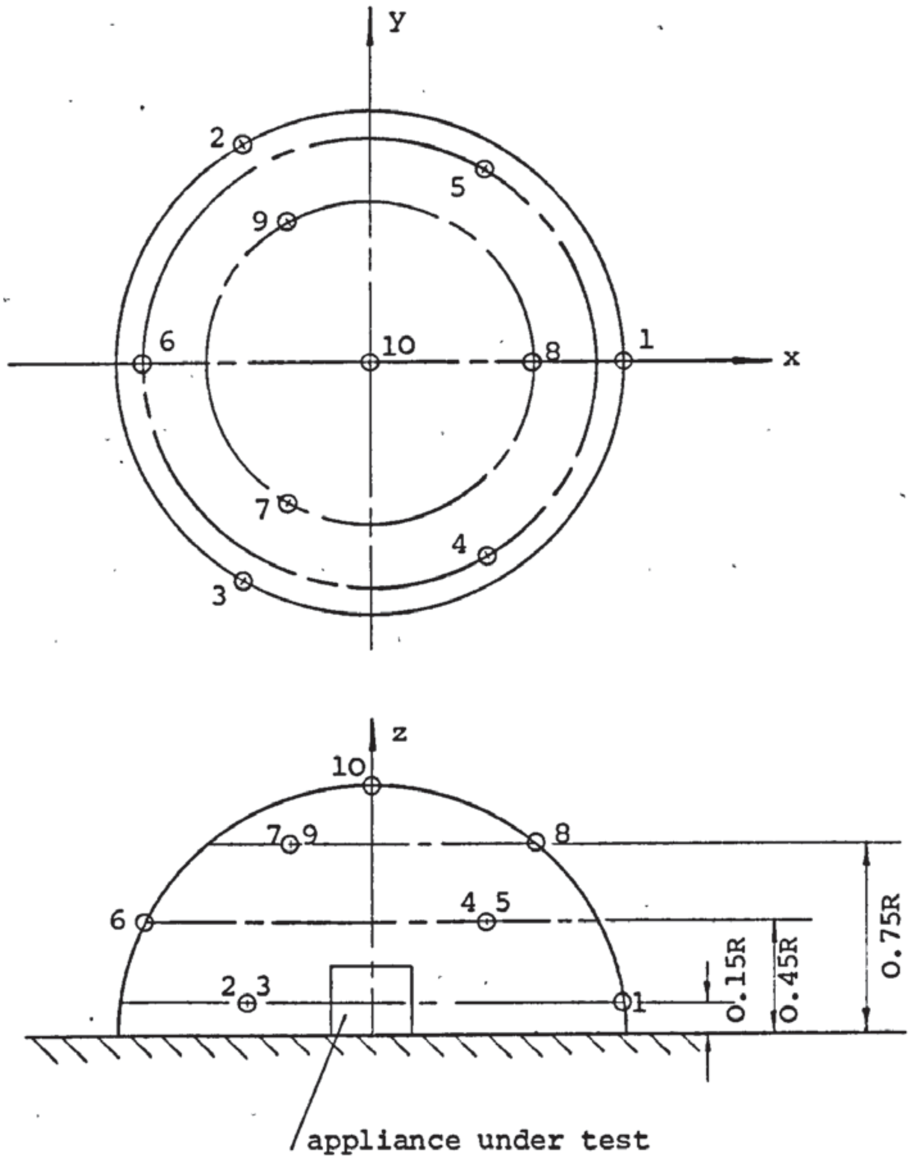
The sound pressure level in each octave band was noted for each of the 10 microphone positions. The procedure for obtaining the mean octave band pressure is given in the test code. The A-weighted sound power level (SWL) of the appliance under test is calculated from the mean value of the measured sound pressure levels, SPLm, using the equation:

$$SWL = SPLm - K + 10 \log [S/So]$$

where, S - area of the hemispherical surface, square metres,

So - reference area, 1 sq m.

K - is the environmental correction of the test room, determined during room qualification. A method is given for its derivation in the standard.



Co-ordinates of microphone positions:

No.	x/R	y/R	z/R
1	0.99	0	0.15
2	-0.5	0.86	0.15
3	-0.5	-0.86	0.15
4	0.45	-0.77	0.45
5	0.45	0.77	0.45
6	-0.89	0	0.45
7	-0.33	-0.57	0.75
8	0.66	0	0.75
9	-0.33	0.57	0.75
10	0	0	1.0

R - radius of hemisphere 1.5m.

Figure 4.1 Hemisphere measurement surface with ten microphone positions.

Comparative method

Using this method, the sound power level is determined by comparing the mean-square sound pressure level produced by the appliance under test with that produced in the same environment by a reference sound source. This method is recommended to check for the possibility of a systematic difference in the frequency characteristics in the results obtained in different environments.

The same hemispherical surface is suggested for the comparative as for the direct method. The IEC 704.1 also suggests that for a comparison of appliances of the same family, type and size (as is the case for sound measurements within this project) the number of microphone positions can be reduced to a single position. Position number 8 is suggested for the surface shown in Figure 4.1.

4.3.6 Procedure for sound measurements at Perivale

An accurate measure of the sound power level emitted by a cleaner was determined at the NEL, East Kilbride, using the direct method. For the evaluation of noise reduction modifications or comparison of cleaners however, this method was not required. As suggested in the IEC 704.1 test code, a single microphone position option was selected.

The single microphone was positioned in front of the cleaner at a distance of 1.5m from the centre of the base of the cleaner and a height of 1.0m from the floor. The centre of the base of the cleaner was placed to coincide with the centre of a Wilton carpet one metre square and the handle was lowered to the normal operating position. The carpet, cleaner and the microphone were placed centrally in the acoustic room. The sides of the room were approximately 250mm away from the microphone and cleaner handle.

Initially, the suction nozzle of the cleaner was adjusted to its highest setting and the cleaner was run-in for ten minutes at its normal running speed. At this nozzle setting the agitator bristles ran clear of the carpet and this helped to reduce carpet wear. After 10 minutes the nozzle height was set to its stated normal operating level (indicated in the instruction manual) at setting 2 and run for a further three minutes. During this time the cleaner achieved a steady-state condition. The overall sound pressure level was noted on the B & K type 2107 frequency analyser and the narrow-band and third octave values were obtained on the Nicolet unit.

4.4 Vibration measurements

The difference between vibration measurement and vibration response is simple: the former implies measurements under normal operation and the latter refers to a measurement of the response to external excitation. Vibration response is a characteristic property of a structure and gives insight to its properties; namely mass, stiffness and damping. These properties are determined by exciting the structure with a known input forcing function and then analysing the response.

4.4.1 Vibration transducers

System response can be described by one of three motion parameters; displacement, velocity or acceleration. Transducers are available to measure each of these parameters but the accelerometer, to measure the acceleration, is the most widely used. The principles of measurement are well described in Broch (1980). The instantaneous acceleration is measured directly and the velocity or displacement can be determined by integration or double integration of the measured signal, respectively. According to Ewins (1984), it is essential that the transducer weight is small compared with the body under test. Hence, miniature accelerometers made by B & K and D J Birchall were used throughout this project. Ewins also stated that the means of attachment of the accelerometer is very important because the response of the transducer is affected by poor installation. Therefore, the accelerometer was fixed with "beeswax" or double-sided adhesive tape.

4.4.2 Calibration of vibration measurements

A vibration measuring system, comprising an accelerometer, charge amplifier, signal conditioning system and the recording device (meter or frequency analyser) and the cables was calibrated by the following single procedure. The accelerometer was placed on an accelerometer-calibrator, B & K type 4291. This compensates for the trandsducer mass and produces a reference acceleration level of 10 m/s^2. The output from the accelerometer is then a known charge which is can be related to the gain of the charge amplifier and the sensitivity of the analyser. The reading on the analyser is thus related, by a calibration constant, to a reference value.

4.4.3 Excitation methods for vibration testing

There are many ways by which excitation can be achieved and the following methods were used in this project;

- sinusoidal excitation,
- random vibration excitation,
- impulse hammer for shock testing,
- the motor in a standard cleaner was also used to "excite" the cleaner structure by running it at various speeds to obtain the vibration characteristics of panels of interest.

Electromagnetic vibrators are suited for vibration testing because high levels of force of broad-band frequency can be obtained to excite a structure. For lightweight structures, low levels of excitation force are required so a medium sized Goodmans vibrator, type V390A, was used. A further advantage is the ability to excite the structure with a range of frequency functions such as "white noise" and pure sinusoidal wave form.

Sinusoidal excitation

A sine wave generator was used to drive the electomagnetic vibrator for a sinusoidal excitation. The test piece was mounted onto the vibrator with a force transducer sandwiched in between as shown in Figure 4.2. The frequency of excitation was increased gradually while the response was carefully noted.

Random frequency excitation

The test equipment for random vibration testing was the same as for sinusoidal except that a "white-noise" generator was used.

Impulse excitation

The instrumented hammer has the advantage that a component could be easily excited at any point without the need for an attachment. Therefore, the component, with an accelerometer placed on it, was freely hung. The force input to the structure is dependent upon the strength with which it is hit and the shape of the impulse due to the type of tip used, namely rubber, plastic or steel. Figure 4.3 shows the frequency spectra of the impulse force for different tips.

The hammer has advantages over the vibrator. It is portable and therefore required less time for setting up. The excitation point can be changed at will, and no structural fixing is required. The hammer kit does have disadvantages. It required correct selection of the tip to be used and skill in delivering the impulse in order that the force input is of a limited bandwidth: also when the hammer was used, it was essential to prevent double hits. The inevitable variation in the magnitude of each hit was compensated for by averaging over a number of hits, typically 32. Figure 4.4 shows the apparent difference in the response due to a hard plastic and a soft rubber hammer tips. The main difference concerned the magnitude of response, and not the frequency.

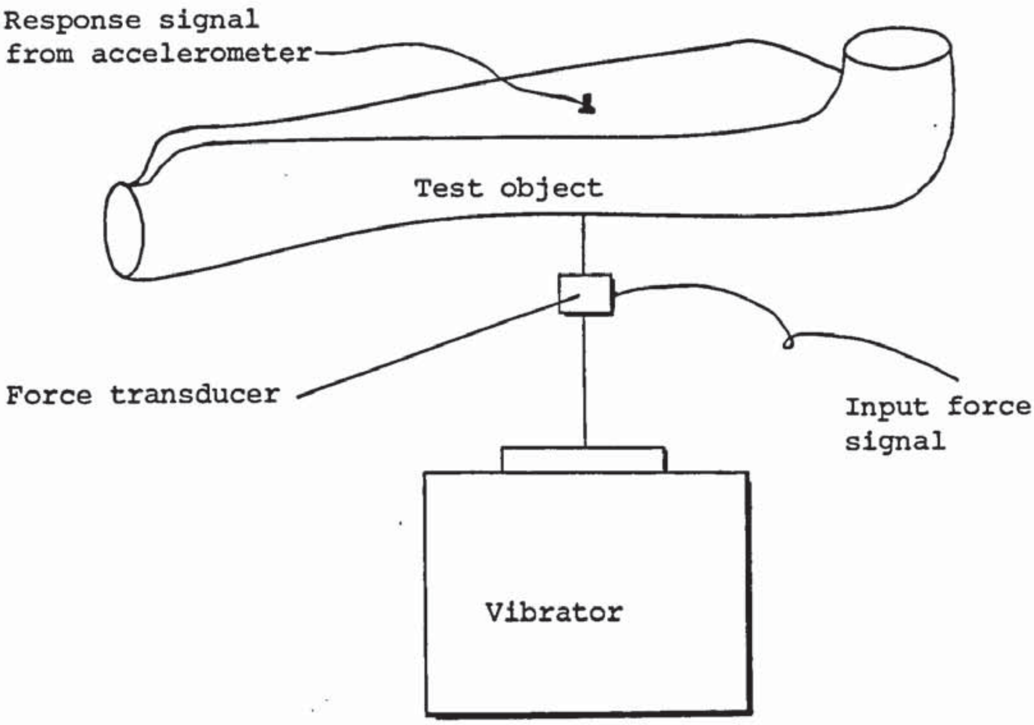


Figure 4.2 Set-up for vibration testing.

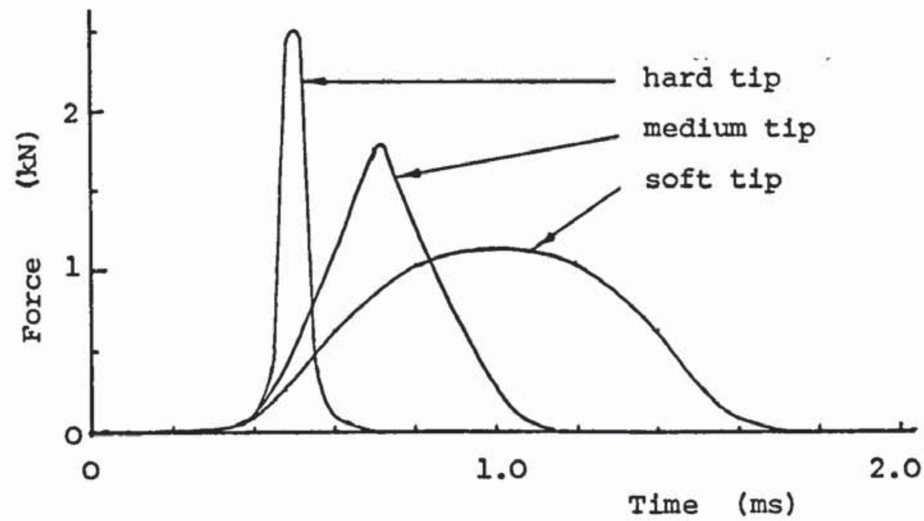


Figure 4.3 Impulse shape for various hardness of the hammer tip.

(from B&K's Instruction manual for impact hammer kit)

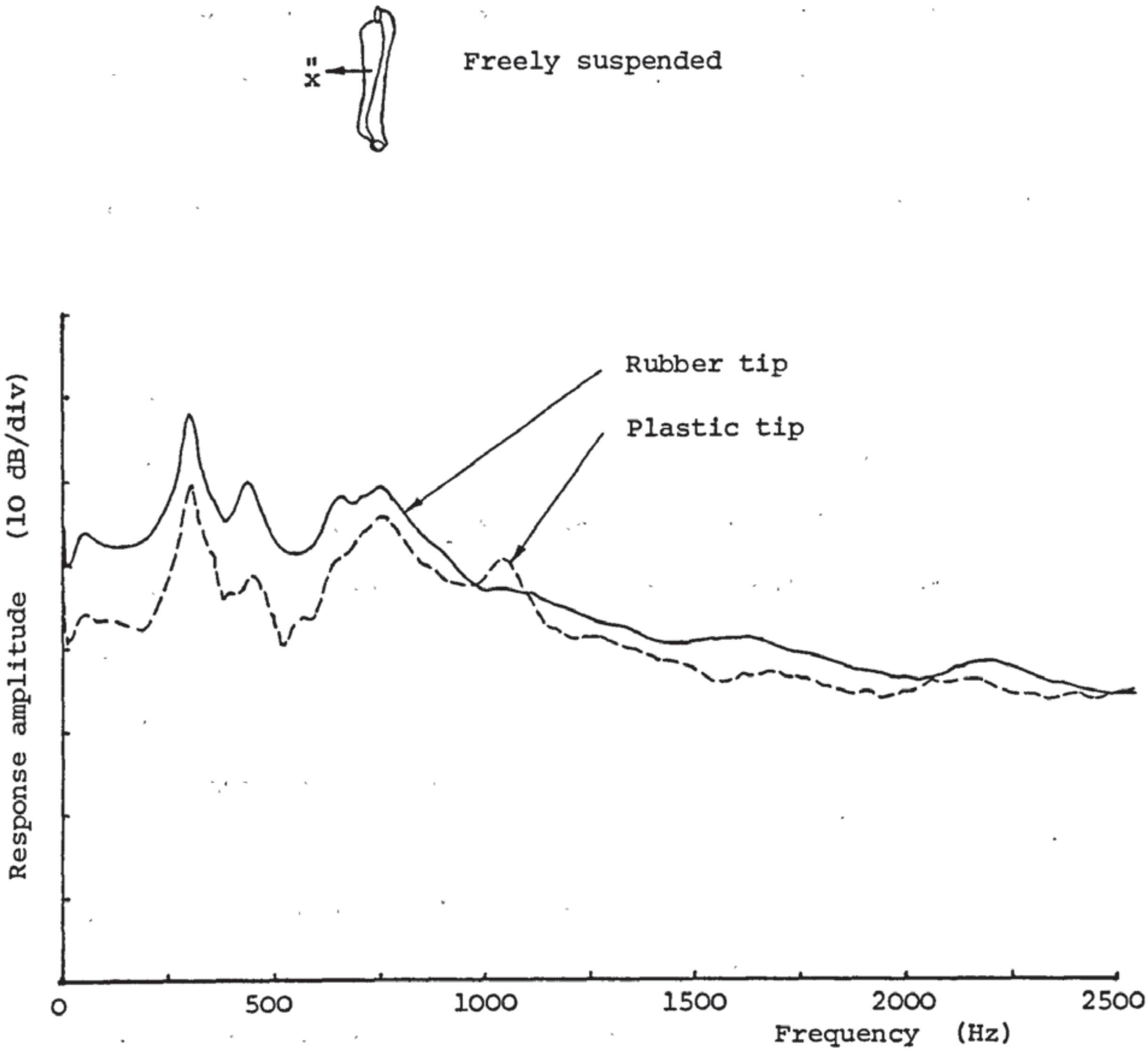


Figure 4.4 Response spectra obtained by impulse testing.

4.4.4 Transfer function measurements

A two-channel analyser is necessary for these measurements and this facility was only available at the University. The results were presented in the form of mechanical inertance; that is acceleration/force. An advantage of a transfer function measurement is that it takes account of the force level in the result. Hence, within limits of linearity, the inertance response is independent of the magnitude of the input forcing function.

4.5 Frequency analysis

For a detailed analysis, as required in an engineering investigation of noise sources, it is necessary to measure the sound level in frequency bands or narrow-band. Various types of frequency analysis were available from the equipment;

- octave band analysis,
- third octave band analysis,
- narrow-band,
- dual-channel narrow-band Fourier analysis.

Most of the initial sound measurements conducted at Perivale were on the Nicolet 444A frequency analyser using narrow-band. However, it was found that narrow-band frequency spectra were highly unrepeatable giving enormous variation. Third octave bands were used to overcome this problem, and narrow-band results were taken where appropriate.

4.6 Imbalance measurements

Imbalance of the universal a.c. motor of the Turbopower cleaner is the major source of vibration. These vibrations are transmitted to all parts of the cleaner. The following instruments were used to provide a measure of the imbalance of the motor armature.

i) Bruel and Kjaer vibration meter, type 9500, and

ii) Jackson-Bradwell dynamic balancing machine.

4.6.1 Vibration meter

The portable meter used in the Floorcare Laboratory consisted of an accelerometer installed on a probe having a selection of tips from a sharp to a rounded point. The response signal was input to the vibration meter on which the acceleration, velocity or displacement could be displayed.

The motor assembly under test was placed in a sponge-lined vee-block for stability. With the motor running at a specific speed, monitored by a hand-held digital tachometer, the probe tip was firmly placed at a point on the motor casing close to each bearing plane, in turn. The vibration level was noted. For an assembled motor, it is impossible to get access to the outer race of the bearing and the vibration level which was measured was that which was transmitted through the motor body structure. Hence the measured vibration was not the original generated at the bearing race, but which had been "conditioned" by the motor structure. Another disadvantage with this method was that the measured value was influenced by the pressure by which the probe was held on the motor body. It proved very difficult to hold the probe steady on a vibrating surface and hence impossible to ensure consistency, this has been noted previously at Hoover by Ogilvie (1984).

4.6.2 Jackson-Bradwell dynamic balancing machine

The machine consisted of a spring suspended cradle which supported the bearing housings of the armature in vee-blocks. The armature was rotated by means of a pulley drive in which a light elastic belt passed over the rotor lamination stack, and driven by a pulley wheel located under the cradle as shown in Figure 4.5. As the armature was rotated the electro-mechanical sensors connected to the cradle detected vibrations caused by the armature out-of-balance. The output of the sensors was

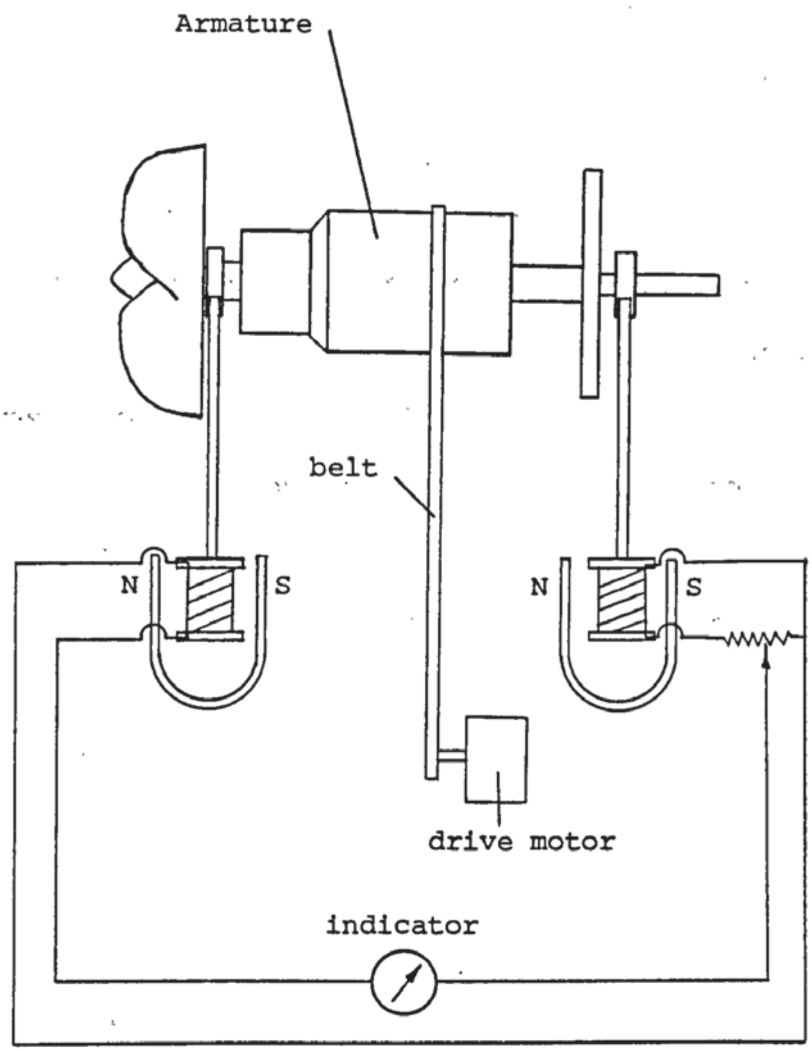


Figure 4.5 Measurement of the dynamic unbalance of an armature.

indicated on an analog type meter display, calibrated to read the armature imbalance in gm-cm, to read either the dynamic out-of-balance on the left or right hand bearing plane or give an indication of the static imbalance.

4.7 Vacuum cleaner performance measurements

A check on performance values is necessary to evaluate and compare one cleaner with another. Subjective tests involve people and, therefore, there can be a vast spread in the results or in their interpretation. Consequently, more objective or quantitative tests are contained in standards to reduce this ambiquity. This section outlines the test and equipment used by Hoover plc. The following parameters are checked;

- maximum suction
- maximum air-flow rate
- power rating
- overall noise level
- dust removal
- dimensional data
- electrical safety
- radio frequency interference, RFI

The most pertinent qualities by which vacuum cleaners can be graded are performance details such as the air-flow and suction and the air-flow and motor efficiencies. These parameters are obtained using a vortex flowmeter rig, which is described in the next section, and were reviewed when modifications to a cleaner were suspected of influencing these parameters.

The results of dust pick-up from carpet are obtained according to IEC Standard 312 (1981). Household debris such as fibre, kapoc and thread cleaning tests are simulated to determine the cleaning efficiency. The swept width and the areas to the front and sides of the cleaner left uncleaned along an adjacent wall are also noted.

The dimensional data is noted mainly for comparison with other types of cleaners. Data such as width, length, height of the handle and the hood (important for under furniture cleaning), weight, cord length, dust bag volume and the filter type are noted. These data were not monitored unless a major dimension was modified.

Domestic appliances have to meet radio and television interference suppression standards of BS 800 (1980), therefore a separate suppression kit is attached to the motor.

4.7.2 Measurement of the air-flow and suction

Vortex shedding rig

The suction and the air-flow were determined semi-automatically using a vortex shedding flowmeter. When testing a cleaner, the air being sucked by the cleaner passes through the "tunnel" comprising a hot wire anemometer to measure the airflow rate, which was controlled by a butterfly-valve. The rig was equipped with a manometer, voltmeter and a wattmeter. These instruments were connected to a Commodore Pet computer which controlled a sequential switching system and a data logger. The motor speed had to be determined externally using a digital tachometer.

Procedure for air-flow and suction measurements

The performance of the cleaner was measured while operating it at a specified voltage. Before commencing the actual measurement, the open and sealed suction levels were determined by installing the cleaner on the vortex rig, and noting the suction for the open and closed butterfly positions, respectively. The cleaner was run for 5 minutes in the open air-flow condition before the test. The measurement commenced at open airflow and the butterfly valve was closed fractionally to obtain the next level of suction. The cleaner was allowed one minute to achieve a steady state before the next set of readings was taken. A set of ten readings was then obtained in the suction range. As the airflow was restricted the motor speed increased which in turn raised the armature temperature. To counteract this temperature rise the open air-flow condition was restored between each reading and the cleaner was allowed two minutes to achieve normal operating condition.

Calibration of the airflow and suction.

The vortex flowmeter was calibrated by an external laboratory against reference instruments.

4.8 Noise source identification techniques

There are many techniques available to identify sources of noise and vibration. In Chapter 3, it was mentioned that many researchers had reported using these techniques with success. Recent techniques make use of data from two signals acquired simultaneously, namely coherence, acoustic intensity and the experimental modal analysis. Acoustic intensity and modal analysis generally require expensive hardware, complex computer software and extensive mass storage. Due to the lack of facilities at Perivale, acoustic intensity and modal analysis could not be tried. Techniques which were available are summarised below.

4.8.1 Narrow-band frequency spectra

Narrow-band facility was available on the Nicolet analyser and on the HP 3582A at the University.

4.8.2 Near-field sound measurements

These measurements were performed in the acoustic room. The microphone face was set parallel to and 15mm away from the surface of the component or the noise source. The distance between the face of the microphone and the component surface is very important because the sound pressure level is dependent upon the distance from a source. Hence a 15mm template was used to ensure this spacing. Calibration of the sound level at a distance of 15mm was unnecessary as the near-field sound levels were to be used for comparison purposes only.

4.8.3. Signature analysis

As described in Chapter 3, this technique is applicable to any machinery comprising rotating elements. The output of a signature analysis is an RPM spectral map, or sometimes referred to as a waterfall diagram. It comprises frequency spectra of the signal of interest which are stacked in an order of increasing motor speed. The measurements should have the same sensitivity and gain settings to enable a direct comparison of magnitude within the family of traces. This was achieved as follows. The motor speed was increased gradually through the range of interest whilst monitoring the response signal. The gain settings were adjusted to avoid the overload on the charge amplifier and the analyser at the cleaner's most active region. Away from this active region the signal would be safe from overloading. The noise levels were plotted in linear scale rather than logarithmic for improved clarity.

The motor speed was monitored very accurately by identifying the imbalance peak on the "zoomed" narrow-band frequency spectrum obtained on the Nicolet analyser. This enabled the motor speed to be measured to within 1.25 Hz. The speed was adjusted by a variac. The Turbopower cleaner is designed to operate at a motor speed of 17100 rpm (285 Hz). However, the speed fluctuates due to external factors such as the condition of the dust-bag, load on the motor and the agitator brushes due to the carpet, resistance to the air flow and variation in the domestic mains supply. The fluctuations in the speed can be significant. Therefore, the upper speed for this analysis was set to 19300 rpm (320 Hz). The lower limit was not considered to be especially important, so a speed of 9600 rpm (160 Hz) was chosen to enable an RPM spectral map of 16 noise traces to be generated.

A Tektronix 4662 digital plotter was used for the RPM spectral map plots. The results were plotted on paper on which the corresponding bottom-left and top-right corner positions were marked for each noise trace at the preselected motor speed. During the subsequent analysis it was a simple matter of resetting the bottom-left corner after each trace had been plotted. The basic RPM spectral map provides valuable information to highlight noise sources. It shows the,

- critical speeds of a machine,
- structural resonances,
- significant rotating elements.

4.8.4 Vibration testing of components

Major components of the Turbopower cleaner were tested to determine their resonant frequencies. The excitation was by an instrumented hammer or a vibrator and the response was analysed. Inertance, that is the ratio of acceleration divided by the force, was the parameter mainly used.

4.8.5 Coherence technique

Coherence is a measure of the cause and effect of a system's response. A coherence function value close to unity indicates a good correlation between the response and the excitation. In Chapter 3 the use of the coherence technique to locate complicated noise sources was discussed.

4.8.6 Sequential removal of components

This technique anticipates that some resonances or contribution to the overall noise generated by a particular component would disappear upon removal of that component. The Narrow-band and signature analyses were performed on the Turbopower cleaner for each stage as components were removed.

4.9 Programme for experimental test work

In a noise reduction project of any kind the first step is to determine the sources of noise and vibration. Once these sources have been identified only then can the principles of noise reduction be applied. A programme for the experimental test work was planned to,

1) evaluate the acoustic room, Perivale
2) determine cleaner performance
3) carry out preliminary sound measurements
4) identify noise sources
5) use noise reduction treatments to reduce noise.

4.9.1 Correlation of the acoustic room at Perivale

The aim of these measurements was to determine the calibration constants for the acoustic room which could then be used to obtain the sound power level (SWL) directly from the sound pressure level (SPL).

- To obtain the relationship between the SPL measured and the corresponding SWL given by the sound power source,
- to calibrate and correlate the acoustic room with the anechoic room at NEL, East Kilbride.

4.9.2 Measurement of cleaner performance

Current specific performance levels of a standard cleaner were used as reference values. These values were necessary for evaluation purposes.

4.9.3 Preliminary noise measurements

A knowledge of the influence of incidental factors upon the sound power level was considered to be critical. To determine the;

- variance in the sound power output upon cleaner start-up,
- variance in sound power level with motor speed,
- variance of sound measurements within normal experimental

limits. This may be due to environmental factors such as temperature, humidity and carpet wear,

- the repeatability of sound measurements.

4.9.4 Identification of noise sources

A list of the major noise sources was compiled using the techniques already outlined, and these sources were ranked in an order of severity.

4.9.5 Noise reduction

In this part of the programme, noise reduction ideas would be hypothesized, implemented and evaluated, then possibly modified and tried again. The approach is dependent upon the frequency and mode of vibration identified and also on space availability, and the manufacturing costs of the solution proposed.

4.10 Review

This chapter has presented the instrumentation and equipment which was available at three sites: that is Hoover plc, Perivale, Vibrations Laboratory at the University and NEL at East Kilbride. Basic measurement parameters were presented together with the techniques for identifying noise sources. The main conclusions of noise reduction principles reviewed in Chapter 3 were used to devise the programme for the experimental test work. This plan was implemented and results follow in the next chapter.

CHAPTER FIVE: NOISE REDUCTION METHODOLOGY: TEST RESULTS AND DISCUSSION

Contents

5.1 Introduction

This chapter presents the results of the experimental test programme which was detailed in the previous chapter. It commences with the calibration of the Perivale acoustic room, by comparing the sound levels with results from the anechoic room at NEL. The effects on the cleaner noise level of incidental factors such as time after switch-on, the motor speed and the ambient conditions are discussed because all these factors affected the repeatability of sound measurements.

The benefits and the limitations of noise source identifying techniques are discussed. The results are presented in a graphical form: logarithmic magnitude is plotted against third octave frequency bands wherever possible. The reason for choosing the third octave analysis were two-fold: firstly, the narrow-band results were erratic and hence lacked repeatability in measurements, and secondly, the results in third octave bands were consistent with those of other researchers. Nevertheless, narrow-band analysis is appropriate for source investigation and is presented on a linear frequency scale.

After identifying the major sources of noise and vibration, the process of ranking these sources is explained. The discussion continues with the noise reduction hypotheses and the final section of the chapter presents and considers the experimental results of noise reduction methods with regard to each noise source.

5.2 Sound measurements in the acoustic room at Perivale

5.2.1 Relationship between SWL and SPL

At present, the engineers of the Floorcare Laboratory measure the sound pressure level of the appliance under test, and then determine the sound power level by the comparative method using a reference sound source. This procedure to determine the sound power level was adequate for a small number of measurements but not for the number envisaged in this project and it was decided to derive a calibration constant for the acoustic room. To do this, the reference sound source was placed on the carpet in the position normally occupied by the cleaner under test. The output from the sound source was increased in steps and the corresponding sound pressure level given on the B & K type 2107 frequency analyser was noted. Figure 5.1 presents the calibration for the Perivale acoustic room. The relationship, as expected, is linear and given by the equation,

$$SWL = 1.011\ SPL + 6.25 \quad dB(A).$$

where; SWL - sound power level of a source, dB(A)

SPL - sound pressure level registered by a microphone at a distance of 1.5m away from the source in a particular enclosure,

The value of the constant "6.25" is significant. In a free-field environment, the value of the constant is 11.5 dB(A), and in contrast, for a reverberant type of environment the value is lower because the reflections add to the pressure which is measured. The constant indicates the absorption qualities of the enclosure.

5.2.2 Correlation of the acoustic room with the anechoic room

All noise work prior to this project was done within the acoustic room at Perivale. Ideally, therefore, all modifications resulting from this project would be evaluated at Perivale. Consequently, the acoustic room was thoroughly checked and the following results illustrate the comparisons of the acoustic and anechoic rooms.

5.2.2.1 Suitability of the acoustic room

ISO 3743 (1976) specified engineering methods for determining the

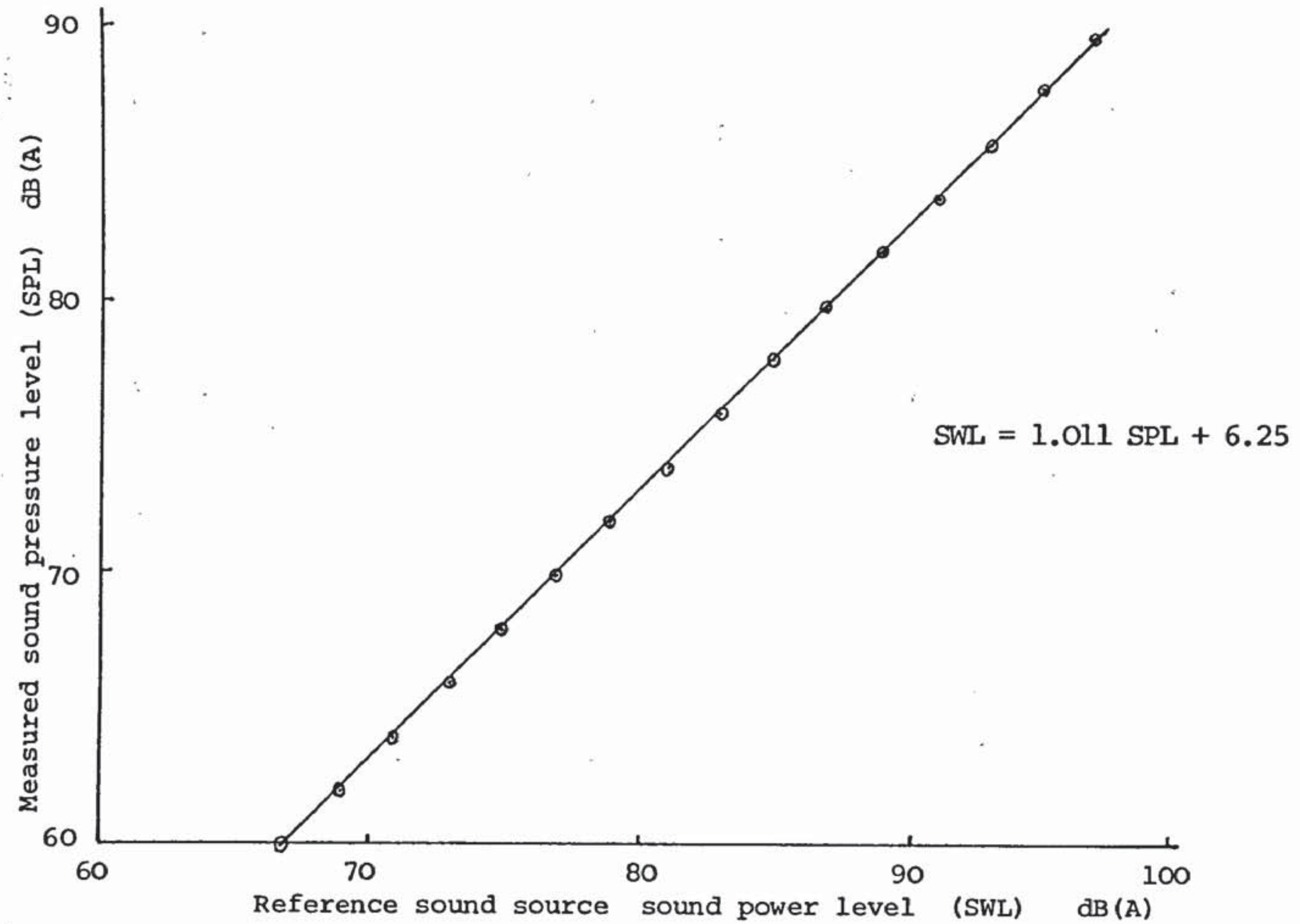


Figure 5.1 Calibration of the acoustic room at Perivale.

sound power for equipment and outlined the guidelines concerning the test room. The following briefly summarises the acoustic room environment.

- The volume of the acoustic room was 13.2 m^3, compared with the recommended minimum volume of the test room to be 70 m^3.
- The Turbopower cleaner occupied about 0.21% of the volume of the acoustic room, compared with the recommended maximum volume of the test unit to be 1% of the room volume.
- The requirements for instrumentation were met in full.
- The ambient SPL inside the acoustic room with the door shut was 28.6 dB(A) on one occasion and 30.3 dB(A) on another. The frequency spectrum was free from pure tones although there was presence of strong low frequency components as shown in Figure 5.2. These results demonstrate that the room was well isolated from the airborne noise from neighbouring Labs. The background sound pressure level is recommended to be at least 9 dB less than the A-weighted sound pressure level produced by the source.

The suitability of the room was checked by a broad-band reference sound source in the manner detailed in ISO 3743. The sound power levels (SWL) were measured in the acoustic room and again in the large anechoic room at NEL. Sound levels in third octave bands for both cases are given in Table 5.1 together with the numerical difference between the two measurements. The allowable difference according to ISO 3743 is given in Table 5.2. In the lower frequency bands the difference is greater than recommended. Consequently, the acoustic room, because of its construction and size, is not ideally suited to low frequency noise measurements. However, the room is adequate for measurements of high frequency noise.

5.2.2.2 Correlation of acoustic and anechoic rooms

A standard Turbopower cleaner was first examined and found to be

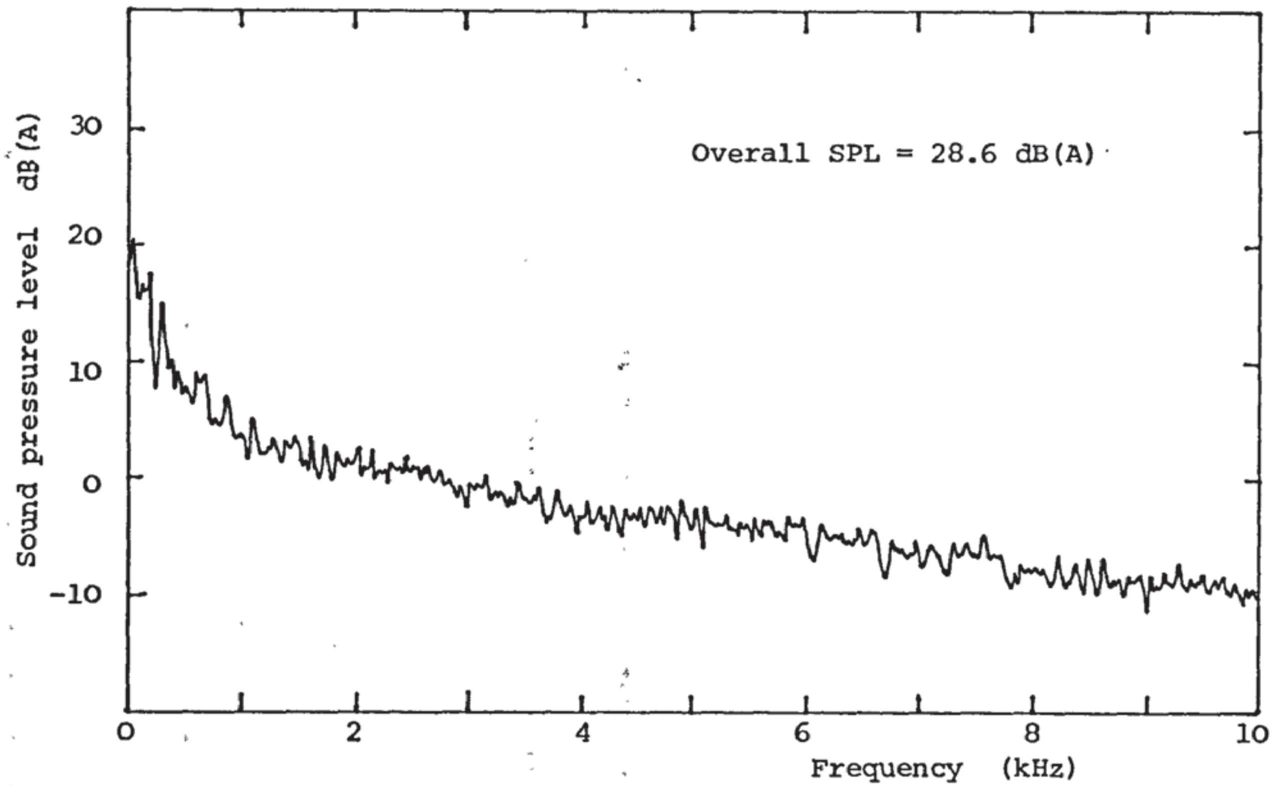


Figure 5.2 Background noise spectra in the acoustic room at Perivale.

Third octave band freq. Hz	Sound power level (not A-weighted) dB at NEL	at Perivale	Numerical difference dB
125	91.44	76.2	15.24
160	88.98	77.5	11.48
200	87.39	81.4	6.00
250	84.79	77.1	7.69
315	83.01	79.6	3.41
400	81.13	78.8	2.33
500	79.25	76.5	2.75
630	78.63	78.1	0.53
800	79.55	80.2	0.65
1000	79.63	81.2	1.50
1250	80.90	83.1	2.20
1600	80.65	81.2	0.55
2000	78.94	79.2	0.26
2500	76.00	76.8	0.80
3150	75.37	76.4	1.03
4000	75.96	76.4	1.24
5000	77.10	76.2	0.90
6300	75.37	75.7	0.33
8000	75.87	73.4	2.47

Table 5.1 Reference Sound Source measurements in the acoustic room and the anechoic room, NEL.

Octave band freq. (Hz)	Difference in band SWL (dB)
125	5
250 to 4000	3
8000	4

Table 5.2 Maximum differences between octave band power levels fo broad-band noise sources measured according to ISO 3743.

manufactured within production tolerances. Its sound power level was measured at both Perivale and NEL. At Perivale, the sound level was 87.1 dB(A) by using the comparison method, and at NEL the sound level was 86.9 dB(A) using the direct method. Whilst the correlation was good for the overall noise level, Figure 5.3 shows a lot of difference which is thought to be due to standing waves and the structural resonance characteristics of the acoustic room.

5.3 Measurement of cleaner performance

These measurements were made to provide the reference values for a standard Turbopower cleaner. Any noise reduction treatment which affected the cleaner performance could then be evaluated by comparing the new performance with the reference values. Results of the suction, air-flow and the related efficiencies are given in Figure 5.4. The speed of the motor was determined by a "tuning fork" type of tachometer known as a "cantilevered speed-box". Its accuracy plus the operator error in reading the motor speed was considered to be far inferior to the results deduced from the narrow-band frequency spectrum. Unfortunately the vortex rig and the Nicolet analyser were in separate laboratories and could never be used simultaneously.

5.4 Preliminary noise measurements of the cleaner

The following data were recorded:

1) the variation of sound power with cleaner start-up,
2) variation of sound power with the motor speed,
3) variation of sound measurements under normal experimental procedures and environmental factors,

and 4) the repeatability of noise measurements was explored.

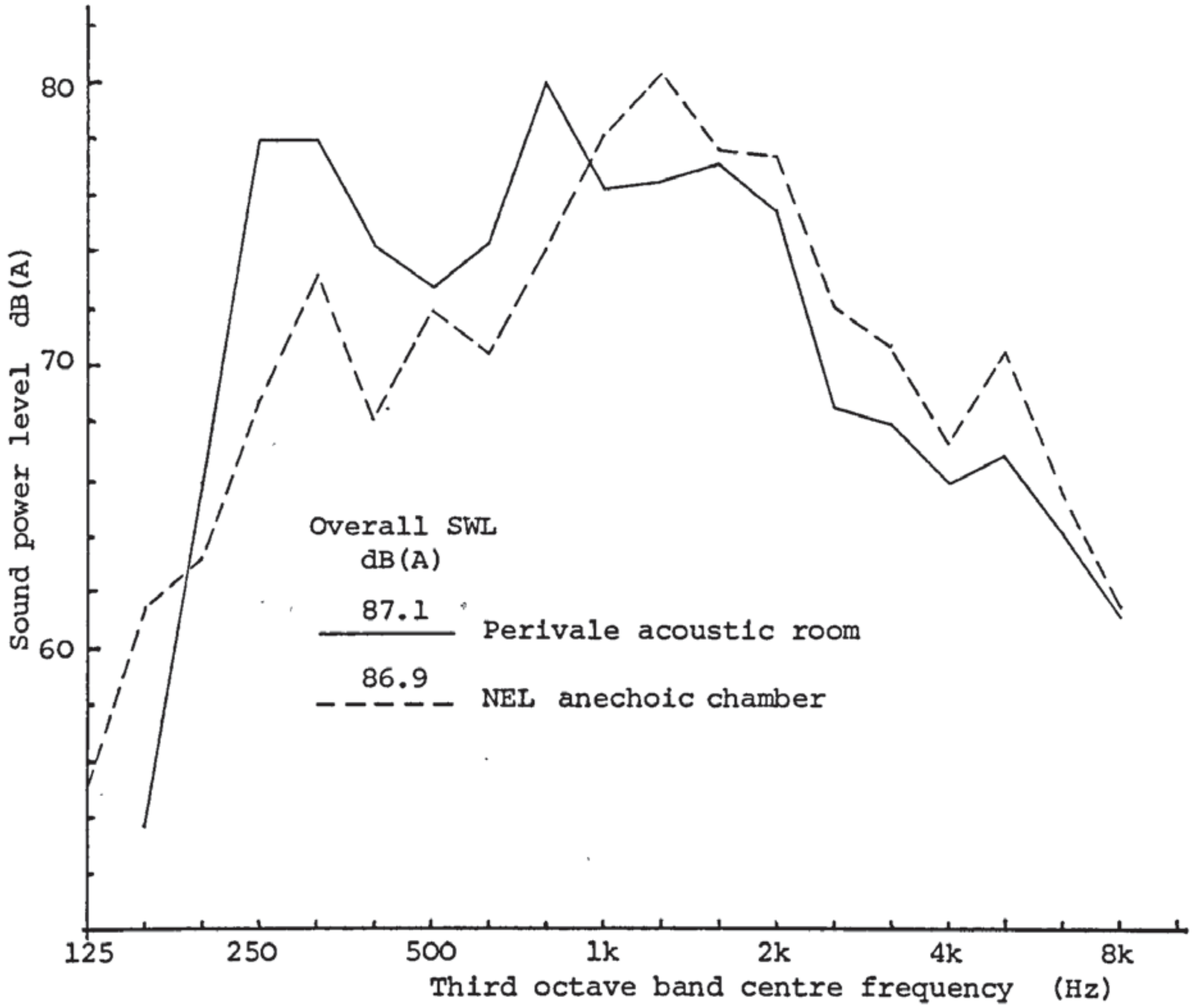


Figure 5.3 Correlation of Perivale acoustic room.

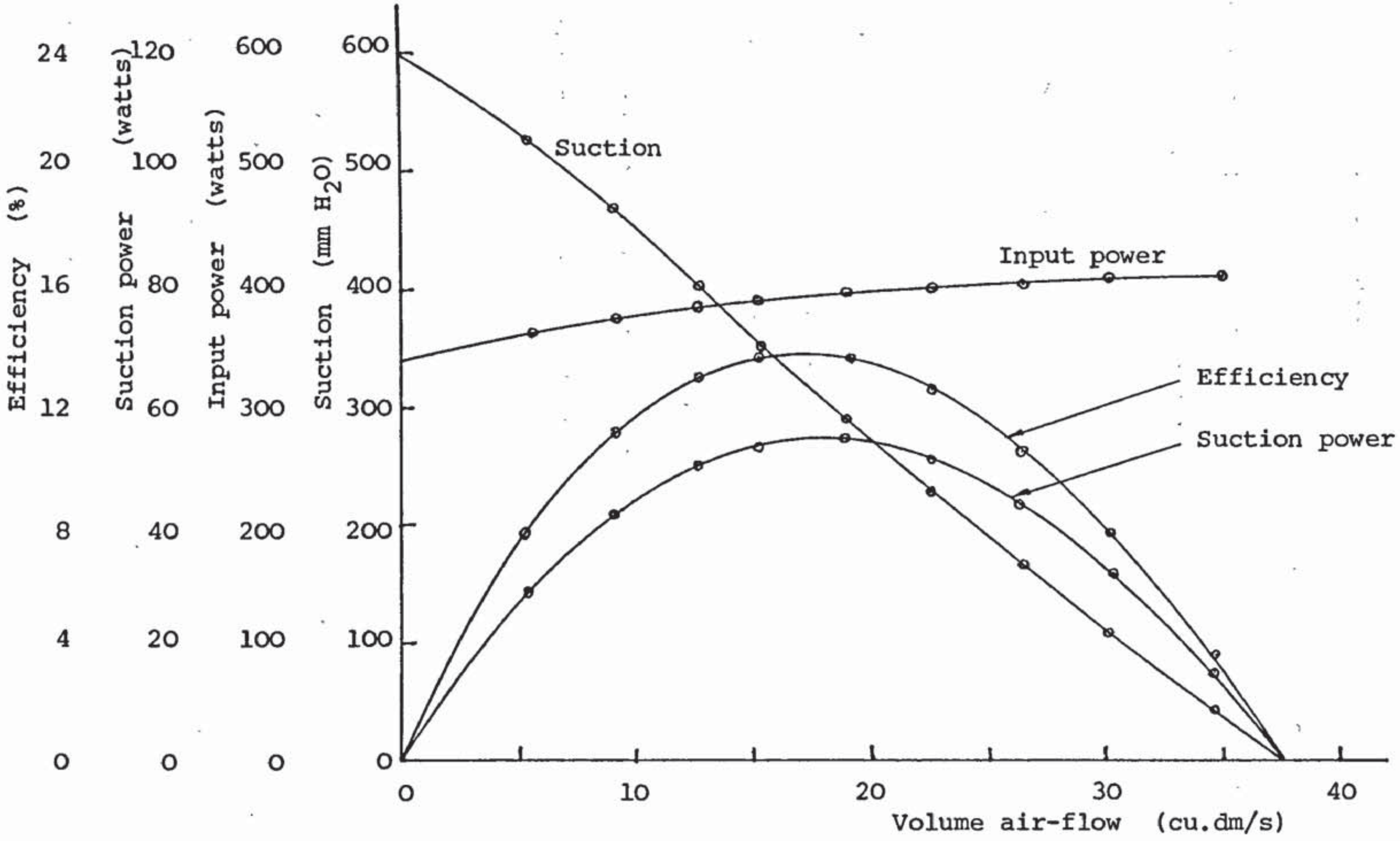


Figure 5.4 Suction and air-flow performance curves.

5.4.1 Sound power level vs. time

From the instant the cleaner was started the A-weighted sound power level was noted at 30 s intervals for 10 minutes. The result is presented in Figure 5.5. The sound level is seen to rise for a short period after start-up and then reduce to a steady value. According to Guinter (1979), this can be due to the extra axial forces on the bearings caused by the expansion of the armature shaft as its temperature rises, which justifies the requirement in the IEC 704.1 to run-in an appliance for 10 minutes.

5.4.2 Sound power level vs. motor speed

The sound level of the cleaner was noted with the motor speed. The objective was to determine the critical speeds, which can cause an increase in the overall noise level. Figure 5.6 highlights three such regions at motor speeds around 166 Hz, 223 Hz and 290 Hz. Resonances at speeds of 166 Hz and 223 Hz will not trouble the user because the motor speed will pass through these regions very quickly during start-up and coast-down. However, the resonance at 290 Hz is significant in that it may be met during normal cleaner operation. This resonance is also the most severe.

5.4.3 The effect of the test carpet on the sound level

The IEC 704.2.1 (1984) code for particular requirements for vacuum cleaners specified the size of the test carpet. The carpet which was used prior to this specification was 1.2m by 0.41m whereas the new size was 1m by 1m square. The type remained a Wilton, single colour of 2.9 Kg/m^2. Sound measurements following the transition revealed some consistent differences and subsequently, a calibration of the room was conducted for each carpet. Relationships between sound pressure and sound power were as follows:

old size oblong carpet : $SWL = 1.011\ SPL + 6.25$ dB(A)

new size square carpet : $SWL = 1.014\ SPL + 5.79$ dB(A)

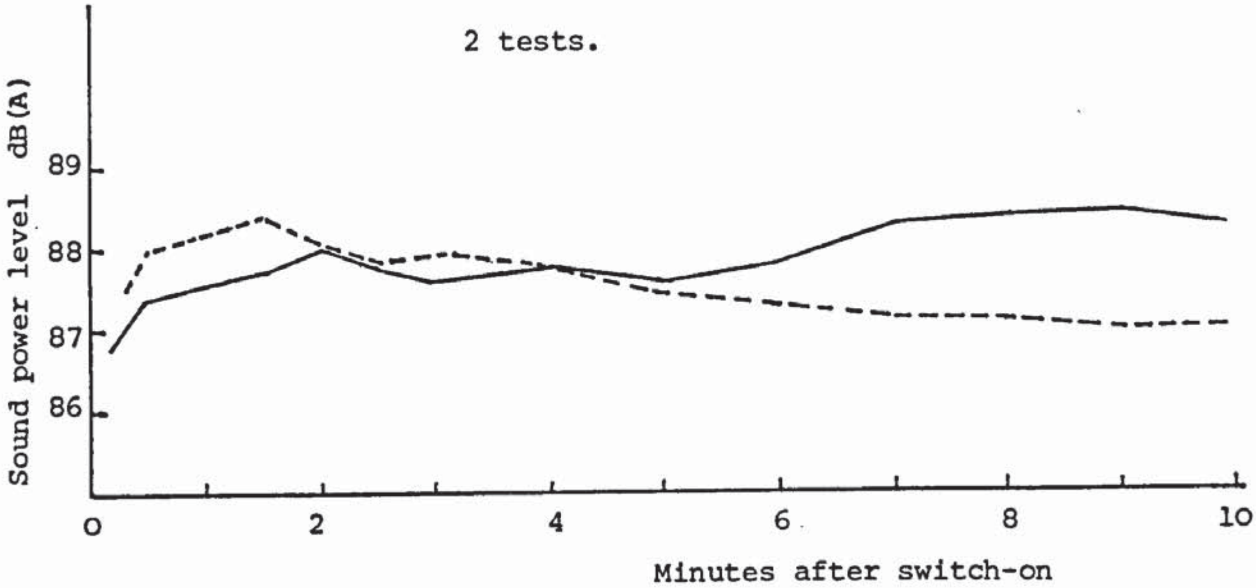


Figure 5.5 Fluctuations in sound power level after cleaner start-up.

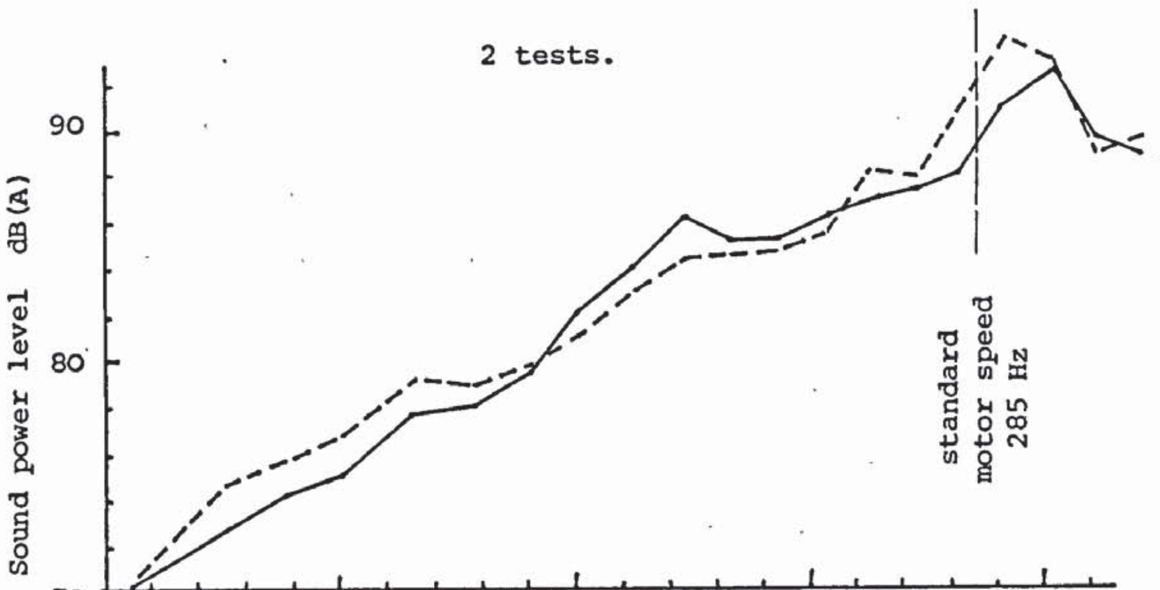


Figure 5.6 Sound power level vs. motor speed for standard Turbopower cleaner.

The results imply that for a given sound power level the sound pressure level for the square carpet is higher than the oblong. Following the discussion earlier, regarding the value of the constant, this is an absorption related problem and implies that the square carpet has lower absorption. Logically, this is surprising because the square carpet has a larger area of 1 m^2 compared with the oblong carpet area of 0.492 m^2, and if the carpet material is the same, then the square carpet should offer greater absorption.

To resolve this anomaly another test was done. The noise level of a cleaner was measured with and without each carpet in turn at each nozzle height. For all of the cases, raising the nozzle height was associated with more noise emission. With no carpet this increase was steady as illustrated in Figure 5.7. The carpetted floor also exhibited a rise, but there was a distinct break-off point or a discontinuity. For the square carpet the break-off point occured between nozzle heights 2 and 3 whereas for the oblong carpet it was between nozzle heights 3 and 4. This implied that the oblong carpet may have a deeper pile or different vibration characteristics. This would explain the increased absorption.

The above results highlight the importance of the properties of the test carpet and the accuracy of the nozzle heights. According to Stewart (1986), there has been some concern over the quality control aspects of nozzle height accuracy; this will be discussed in Chapter 7.

5.4.4 Exploring the repeatability of sound measurements

The narrow-band results were erratic in nature. It was decided to check the variation in the sound measurements which would result from dismantling and reassembling the cleaner.

Dismantle and refit results

Since, to evaluate any modification, certain parts of the cleaner had to be removed, it was decided to dismantle and then reassemble 90% of

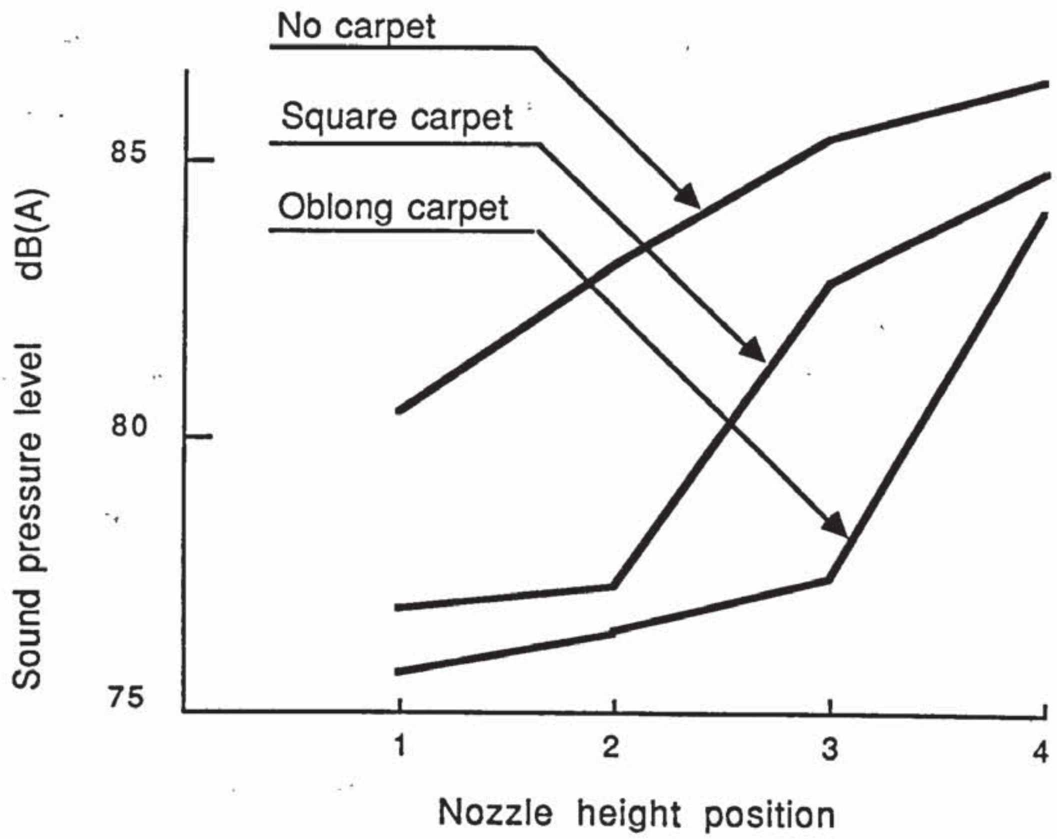


Figure 5.7 Characteristics of cleaner noise against nozzle height and size of test carpet.

the cleaner. 14 tests were done. The results are shown in Figure 5.8a. The solid lines envelop the spread in measurements. The greatest spread is shown for third octaves centred at 250 Hz, 315 Hz, 500 Hz and 630 Hz. The numerical difference is plotted in Figure 5.8b. The overall sound level varied from 88.3 dB(A) to 92.3 dB(A). Probable reasons are that,

1. the motor may not have been seated properly in its rubber mounts, or these may not have been fully relieved,
2. components may not have been seated in the same way after reassembly because of tolerances,
3. screws may have been tightened to different torques each time.

Subsequently, all normal engineering precautions were taken to ensure that the components were properly fitted and screws were tightened, as far as possible, to a similar limit.

Repeat measurement error

Sound measurements were repeated after 20 minutes while the cleaner remained in the same place. These results are shown in Figure 5.8b and show that a difference of almost 2 dB(A) in the overall noise of the cleaner can arise between successive tests! In practical terms, the above results suggest that any noise reduction due to a modification has to be greater than 2 dB for it to be credible. The success or otherwise of any noise reducing modification will therefore be judged against this finding.

5.5 Application of sound source identification techniques

It is desirable and advantageous in an exercise of this kind to use more than one source identifying technique since each gives information of the noise source from a different viewpoint. Each element of information helps in building an overall picture and thus ensures greater confidence in the identification of sources and successful methods for reducing their effort.

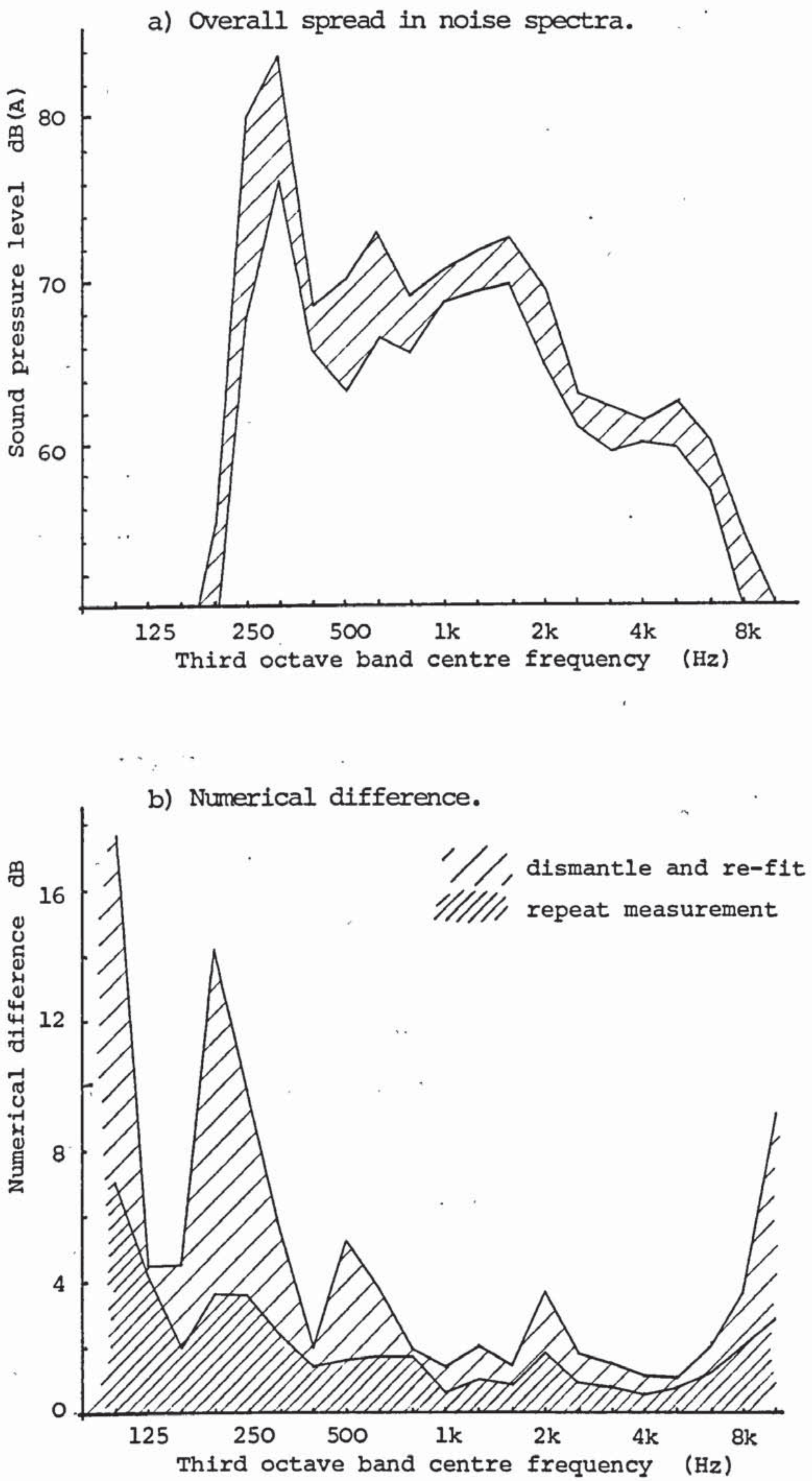


Figure 5.8 Spread in sound level for dismantle and re-fit tests.

5.5.1 Narrow-band frequency spectra of cleaner noise

A typical narrow-band frequency spectrum for the Turbopower cleaner running at a motor speed of 285 Hz is presented in Figure 5.9. The figure shows greater magnitude in the frequency range 0-1600 Hz and then falls steadily at higher frequency. It is dominated by many discrete peaks, some of which can be identified and attributed to the rotating elements of the cleaner motor. Some of the peaks are labelled with two numbers for example 1980(7). The first number indicates the frequency of the peak and the second (in brackets) represents the ratio of frequency divided by the motor speed. This ratio is called the "order" (ie. multiple of the rotational frequency).

5.5.2 Near-field noise measurements

The results given in Figure 5.10 are the sound pressure levels at a distance of 15mm, and these should not be compared or confused with the standard sound pressure levels which were measured at a distance of 1.5 m. The near-field technique was very quick and illustrated areas of high noise levels around the cleaner.

The sound radiation pattern was deduced from the sound pressure level measured at NEL using 10 microphones. The overall sound level at each location is summarised below. The results showed that the sound level is greatest immediately above the vacuum cleaner, the difference being 5 dB(A). The noise radiating in this direction comprised frequencies of the range 800-2500 Hz, as shown in Figure 5.11.

Microphone No:	1	2	3	4	5	6	7	8	9	10
Overall SWL dB(A)	75.3	75.4	75.0	72.5	74.1	73.8	74.5	74.6	74.9	80.0

5.5.3 Signature analysis results

Figure 5.12a shows the spectral map of the noise of the cleaner in

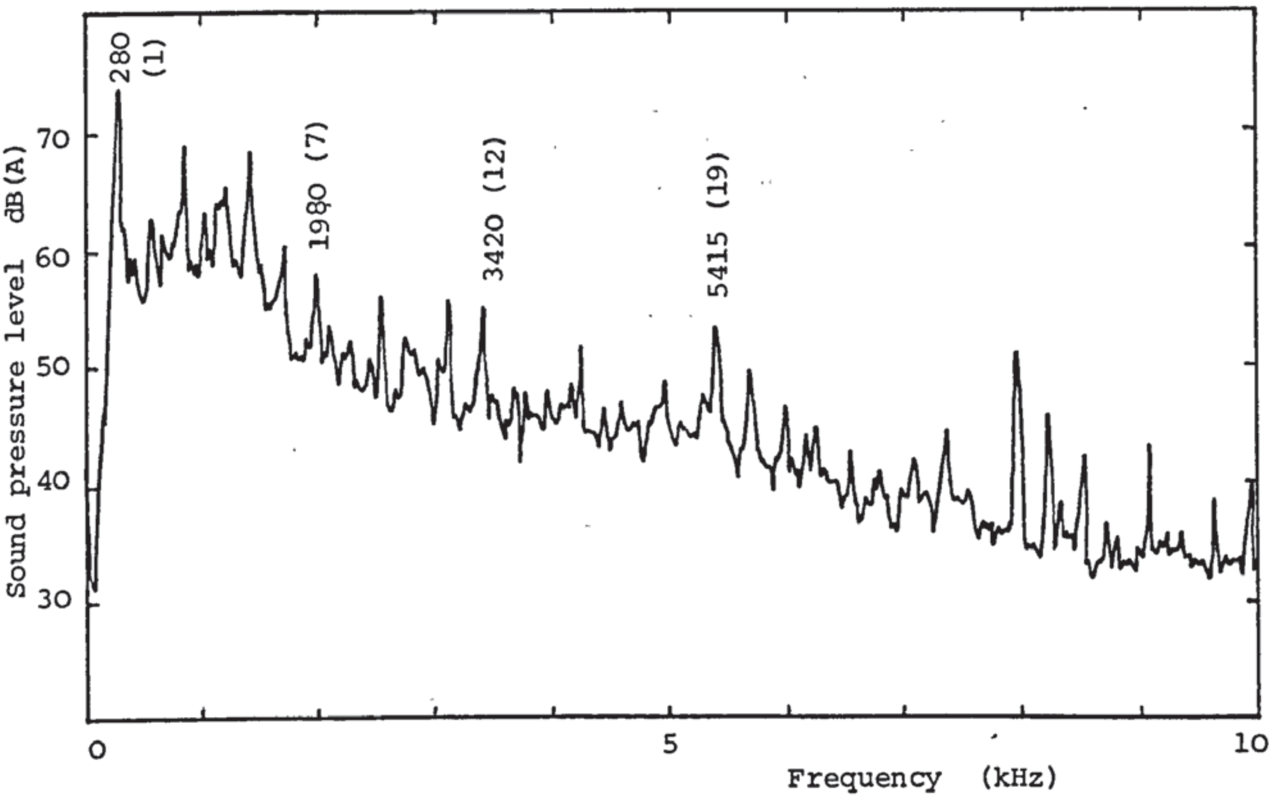
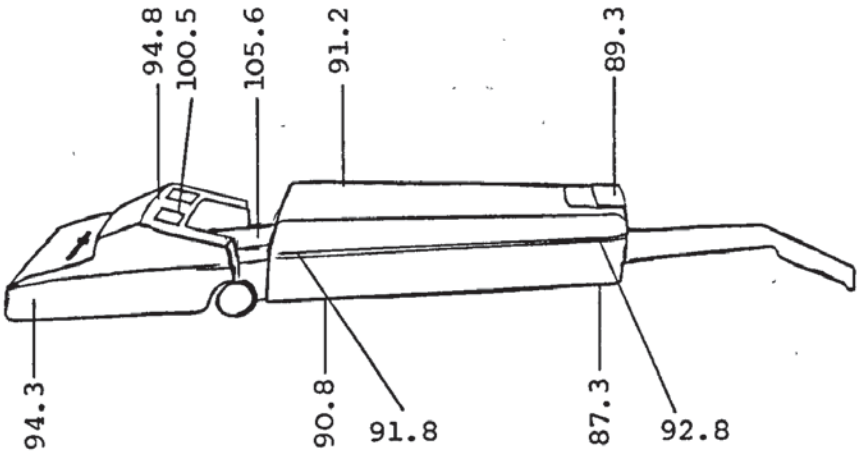


Figure 5.9 Typical noise spectra for a standard Turbopower.



All measurements in dB(A)

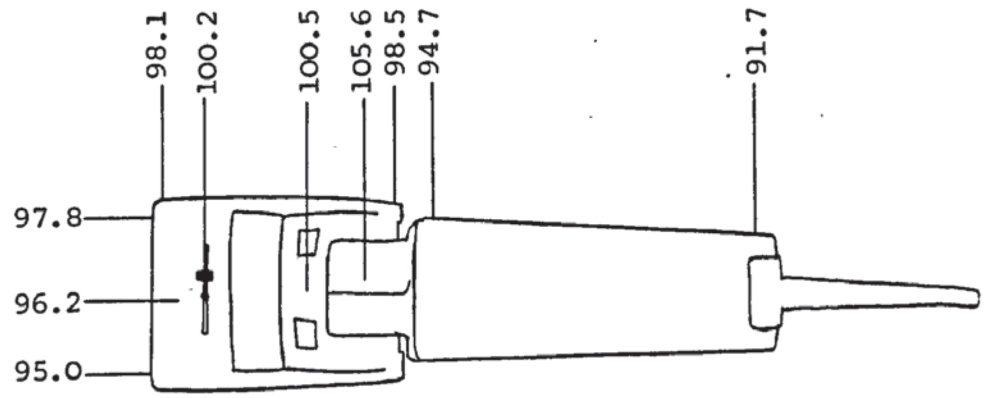


Figure 5.10 Near-field sound measurements.

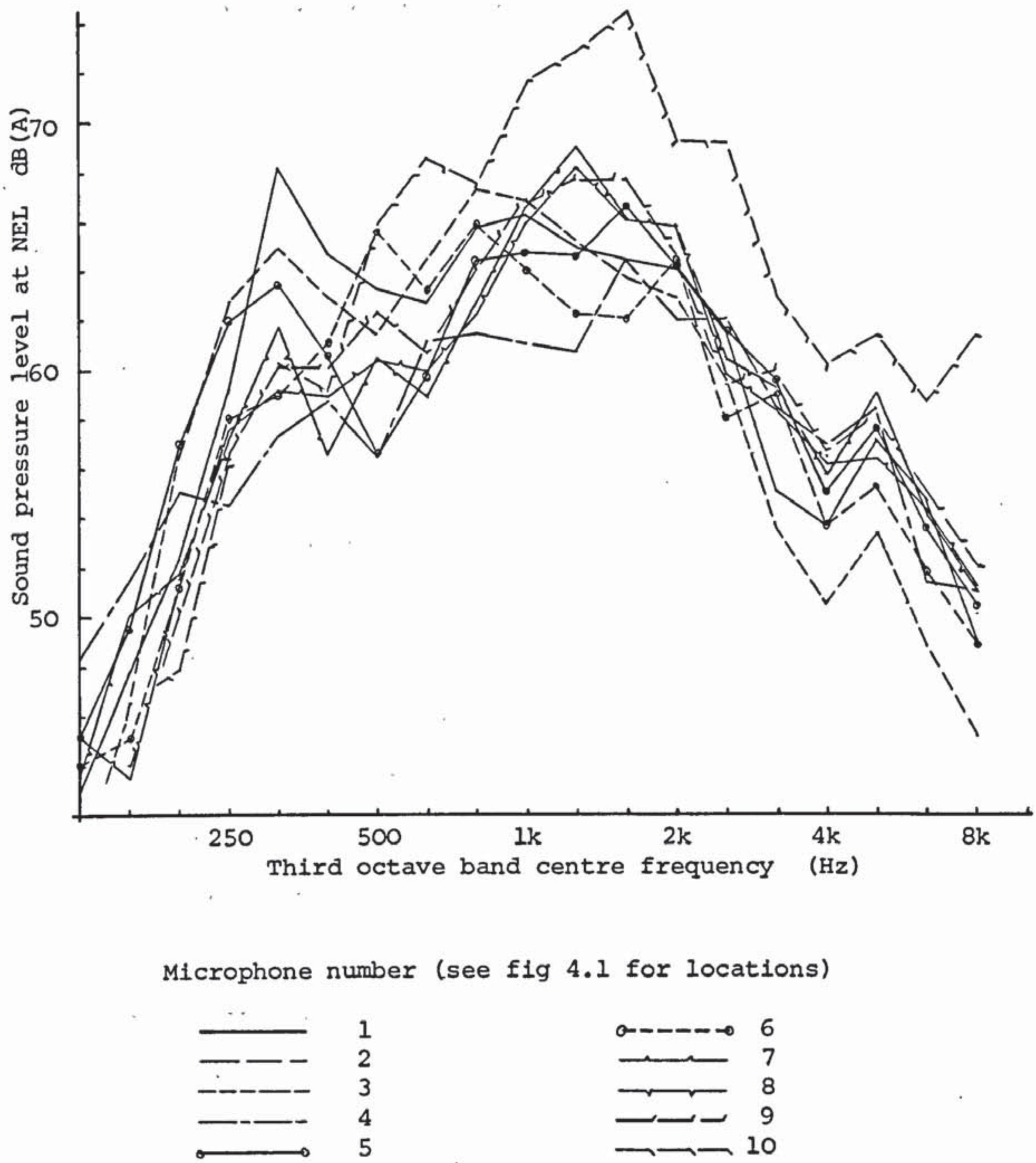


Figure 5.11 Directivity of sound radiating from a standard Turbopower.

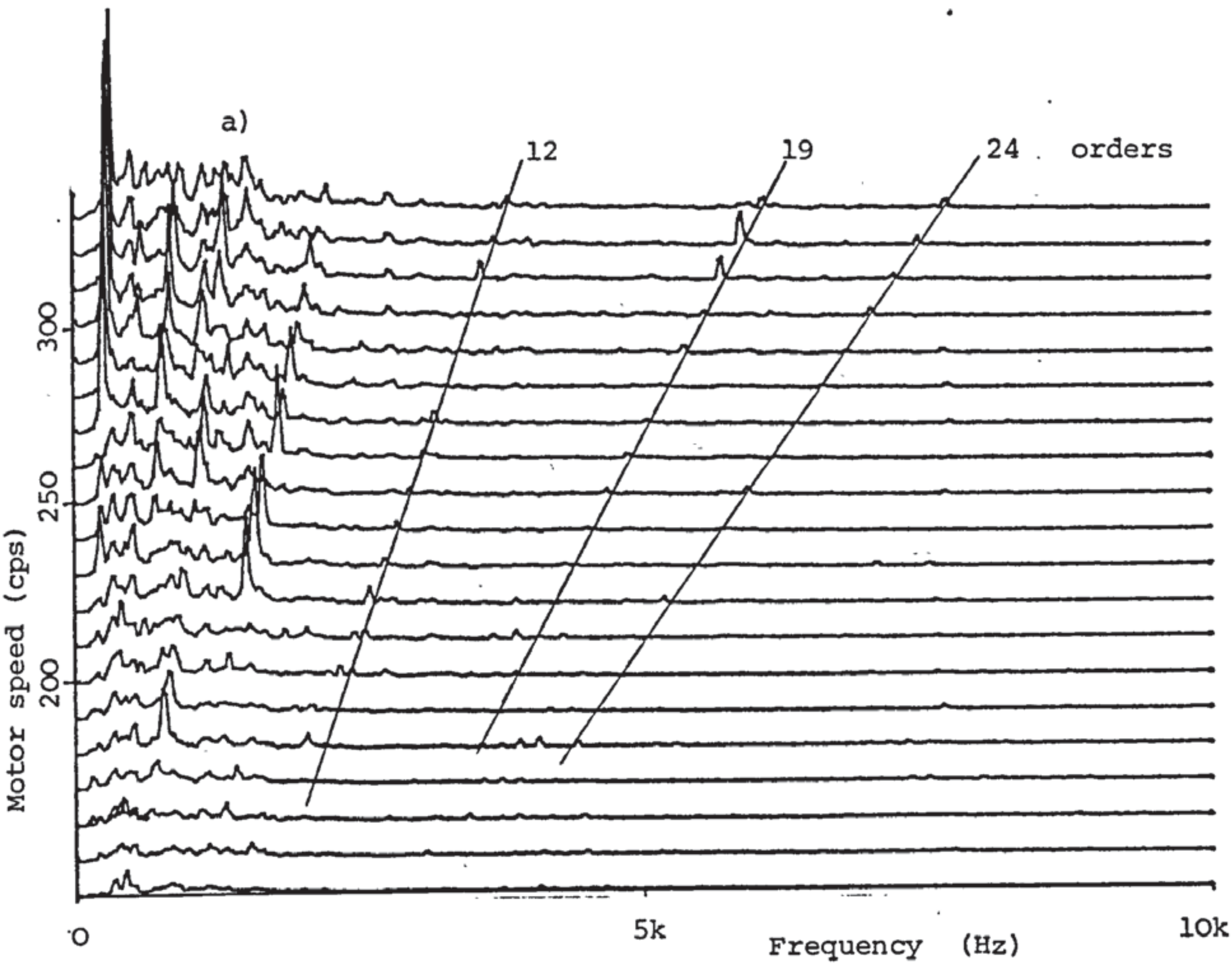
the frequency range 0-10 kHz. Those response peaks which are related to the motor speed lie on a straight line emanating from the origin, for example 12th, 19th and 24th orders are indicated. It follows that significant noise activity was concentrated below a frequency of 2 kHz. This fact agreed with the narrow-band logarithmic noise spectra given earlier. Consequently, Figure 5.12b presents the noise spectral map for 0-2 kHz. The orders which are significant are clearly seen. A structural resonance can also be identified at 520 Hz as an activity at a fixed frequency, independent of the motor speed. The major orders which appear correspond to the details of the motor, which are thus;

Commutator elements	24
Armature lamination slots	12
Suction fan blades	7
Cooling air fan blades	19
Armature bearings	NTN 608
rolling elements	7
Motor/agitator speed reduction	5:1

5.5.4 Sequential removal of cleaner components

This technique relies on matching a change in the response with the component which has been removed. That is, the characteristics in the response appertaining to a particular component would disappear upon its removal. Signature analyses were carried out as follows;

1. a standard cleaner, Figures 5.12a and b, referred to above,
2. a cleaner with the dust bag and the air duct removed, Figure 5.13. Notice the general increase in the broad-band noise and a new resonance at 1640 Hz.
3. cleaner without dust bag, air duct and front dust-bag cover, Figure 5.14. Notice a further increase in broad-band noise and a major resonance around 925 Hz.



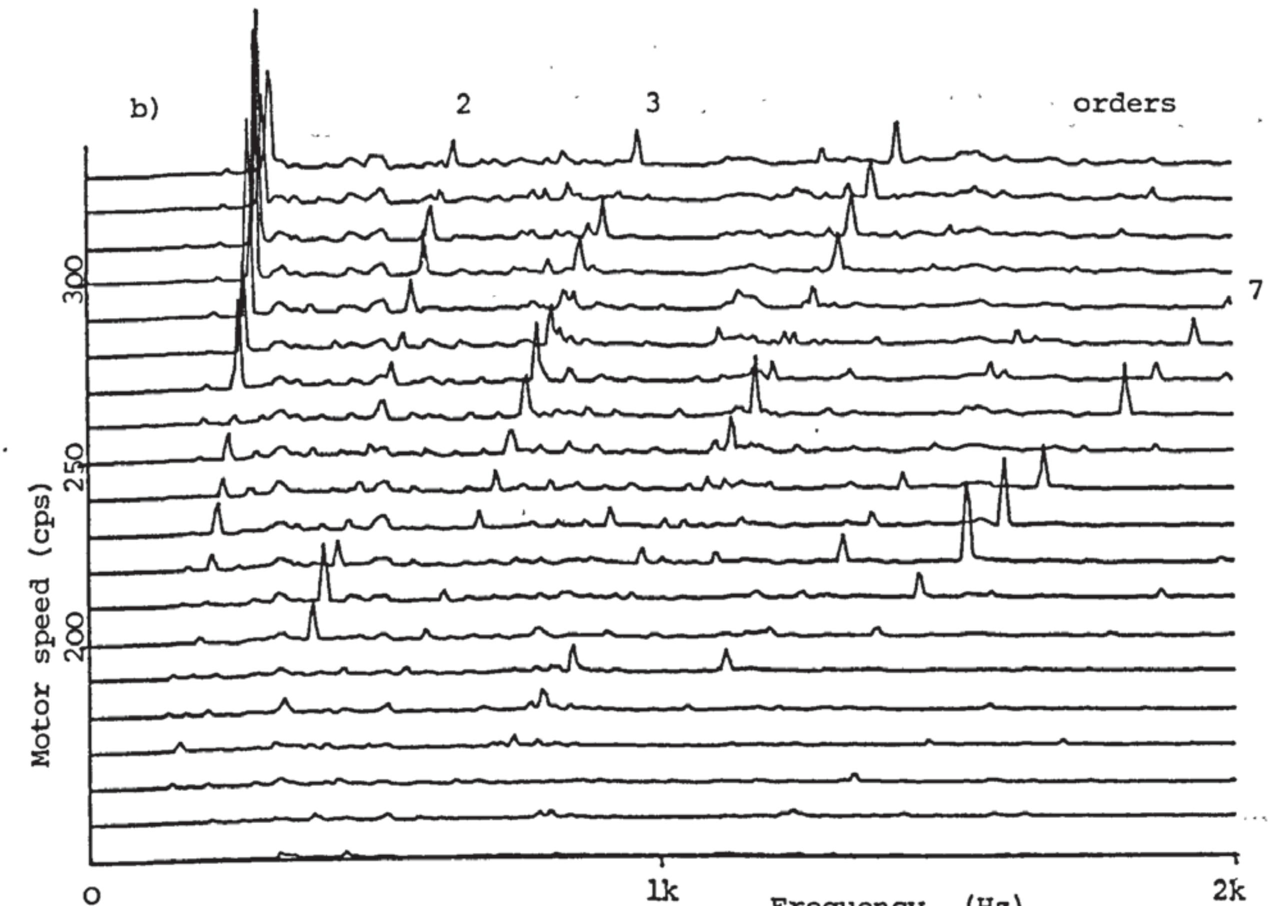


Figure 5.12 RPM noise spectral map for standard Turbopower.
a) 0 - 10kHz b) 0 - 2kHz

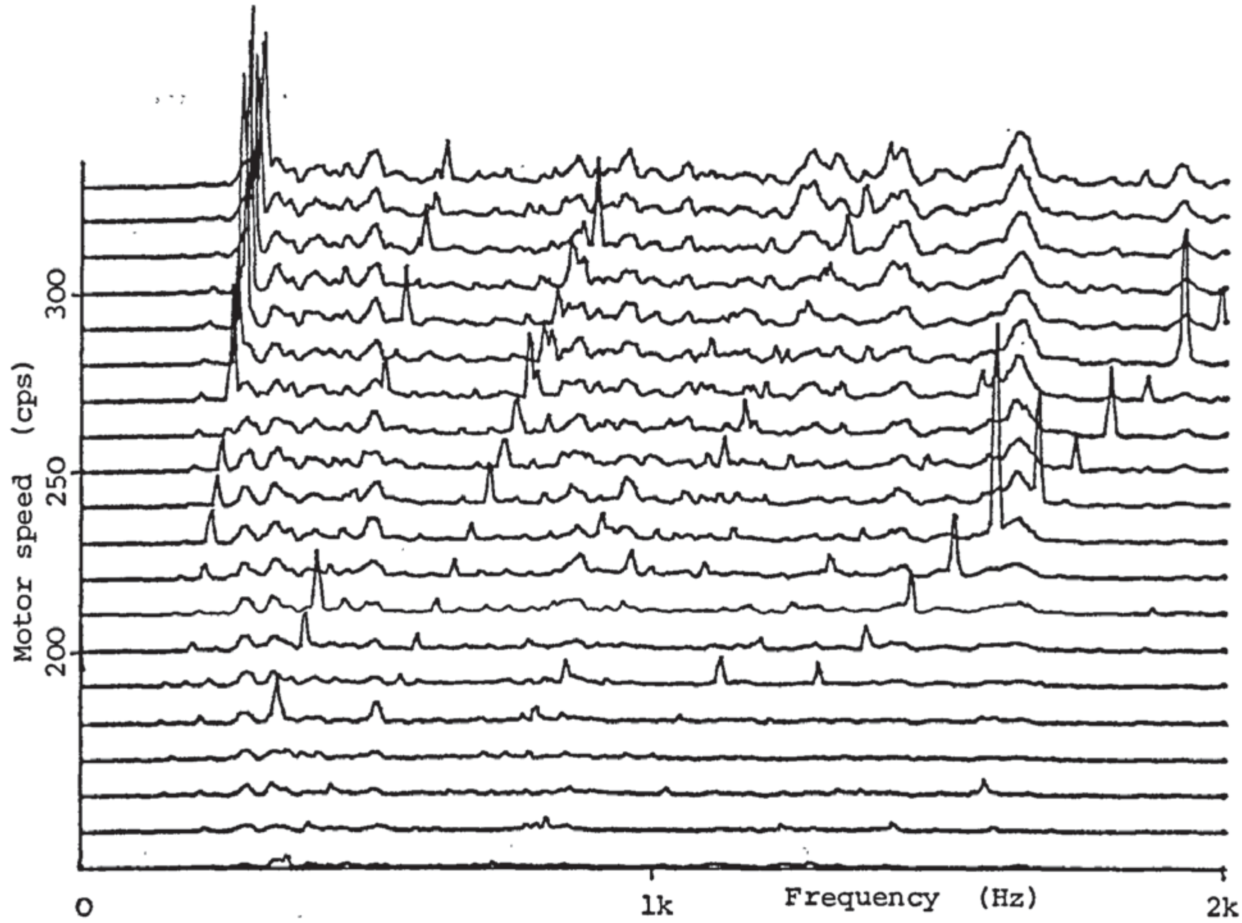


Figure 5.13 Noise spectral map 2; no dust bag, no air duct.

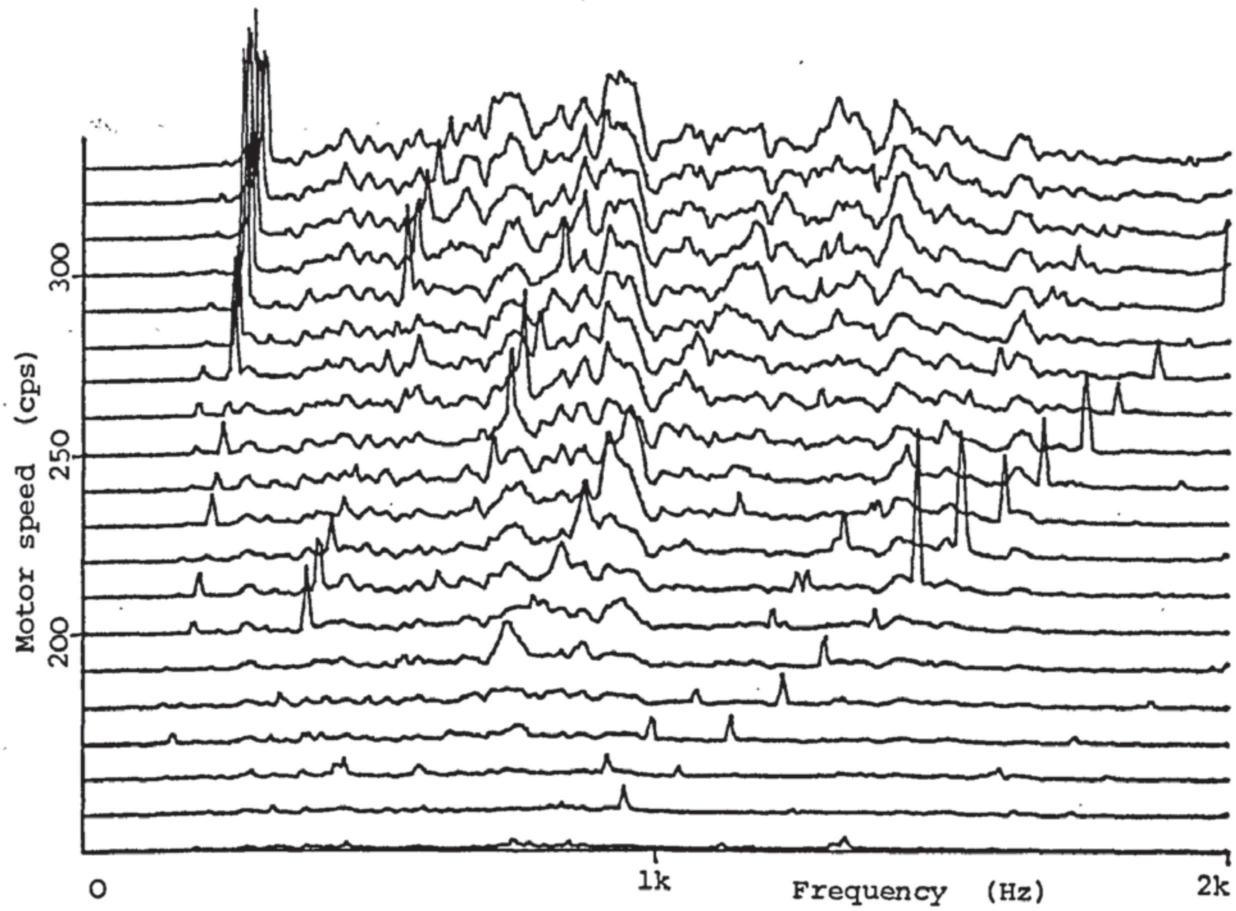


Figure 5.14 Noise spectral map 3; as 2 and without front bag housing.

4. a cleaner without the upright, Figure 5.15. It has the effect of eliminating some resonances and broad-band noise. There is general absence of low frequency response which is associated with large radiating surfaces.
5. a cleaner as in 4: with the agitator belt removed, Figure 5.16. This reduced the load on motor, hence raises its speed which is seen by an increase in the frequency at which the peaks occurred.
6. a cleaner as in 4: with the agitator guard removed, Figure 5.17. This increases the turbulence noise and the 7th order peak.
7. a cleaner as in 6: but without hood, Figure 5.18. Notice the virtual disappearance of the first order peak.
8. a cleaner as in 4: but without hood, (sound measurement only),
9. standard cleaner without hood, (sound measurement only).

The sound level variation with the motor speed in the region 100 Hz to 300 Hz is shown in Figure 5.19 for each of the above test conditions. The results showed that a great deal of noise, about 6 dB(A), propagates from the cleaner nozzle area, see curves 6 and 7. The results also show that the radiated noise became pronounced at high motor speed, that is 290-300 Hz. A drop of about 4 dB(A) was obtained by simply removing the hood at the motor speed of 295 Hz.

Cleaner "build-up" is similar to the removal technique. Some tests were done on the Euro motor. The results plotted in Figure 5.20 are for the noise of the motor as the suction fan, fan chamber and the corrugated duct were assembled. RPM spectral maps are given in Figures 5.21-24 and show; a resonance of the motor casing at 790 Hz, a significant resonance of the fan chamber at 660 Hz, and the enormous size of the 7th order peak compared with the first.

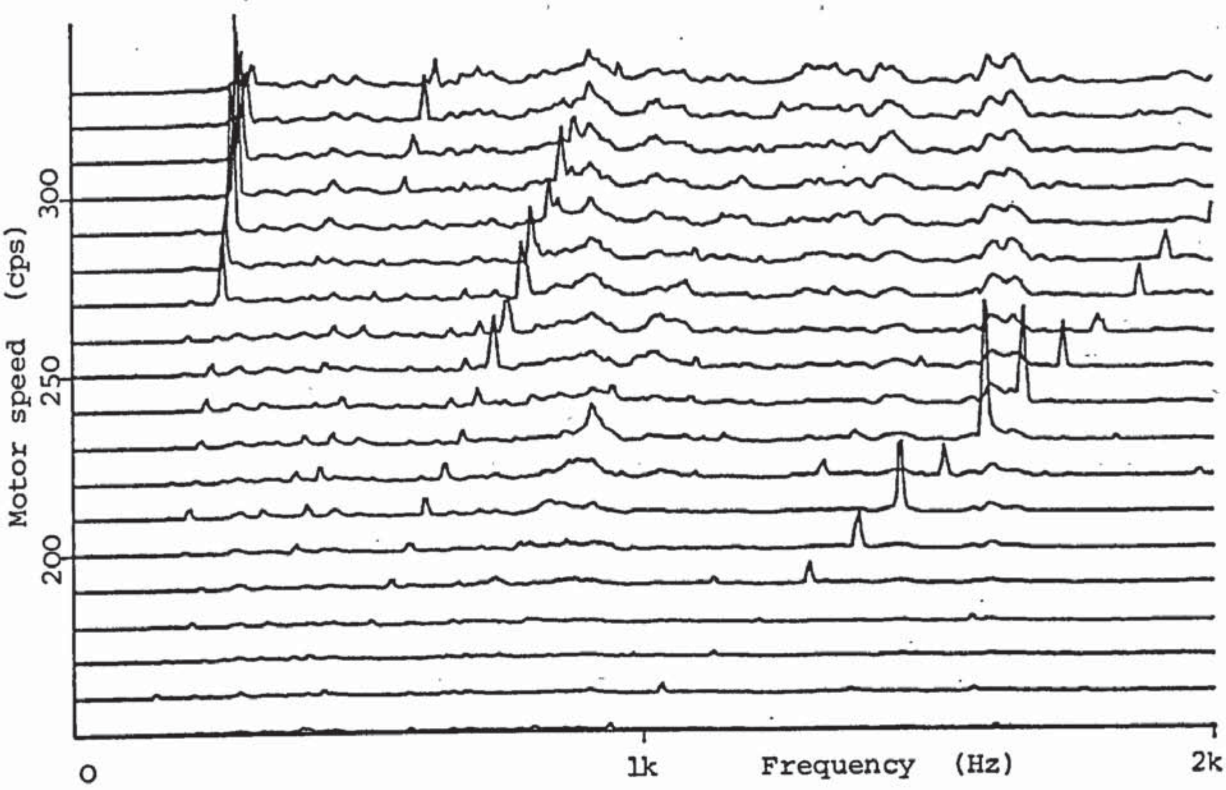


Figure 5.15 noise spectral map 4; without upright.

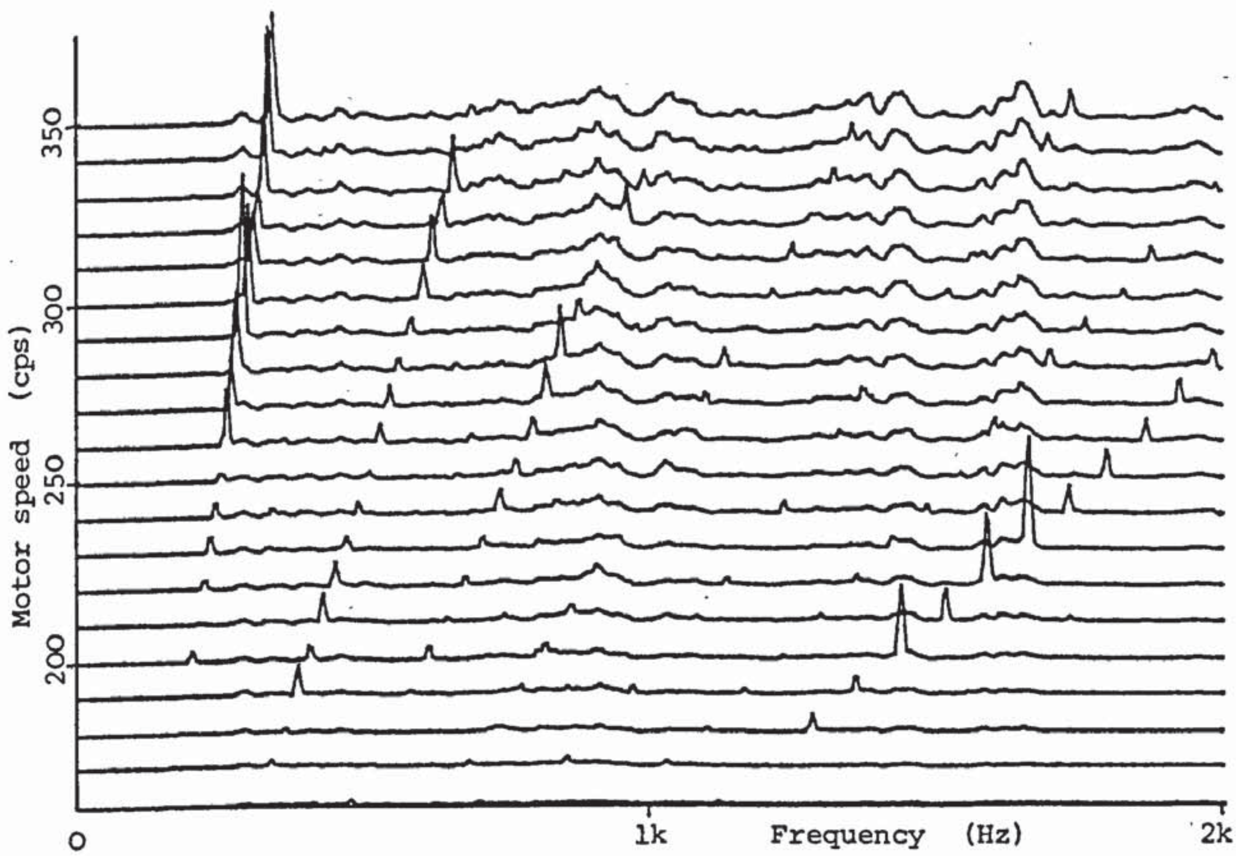


Figure 5.16 Noise spectral map 5; as 4 and without agitator belt.

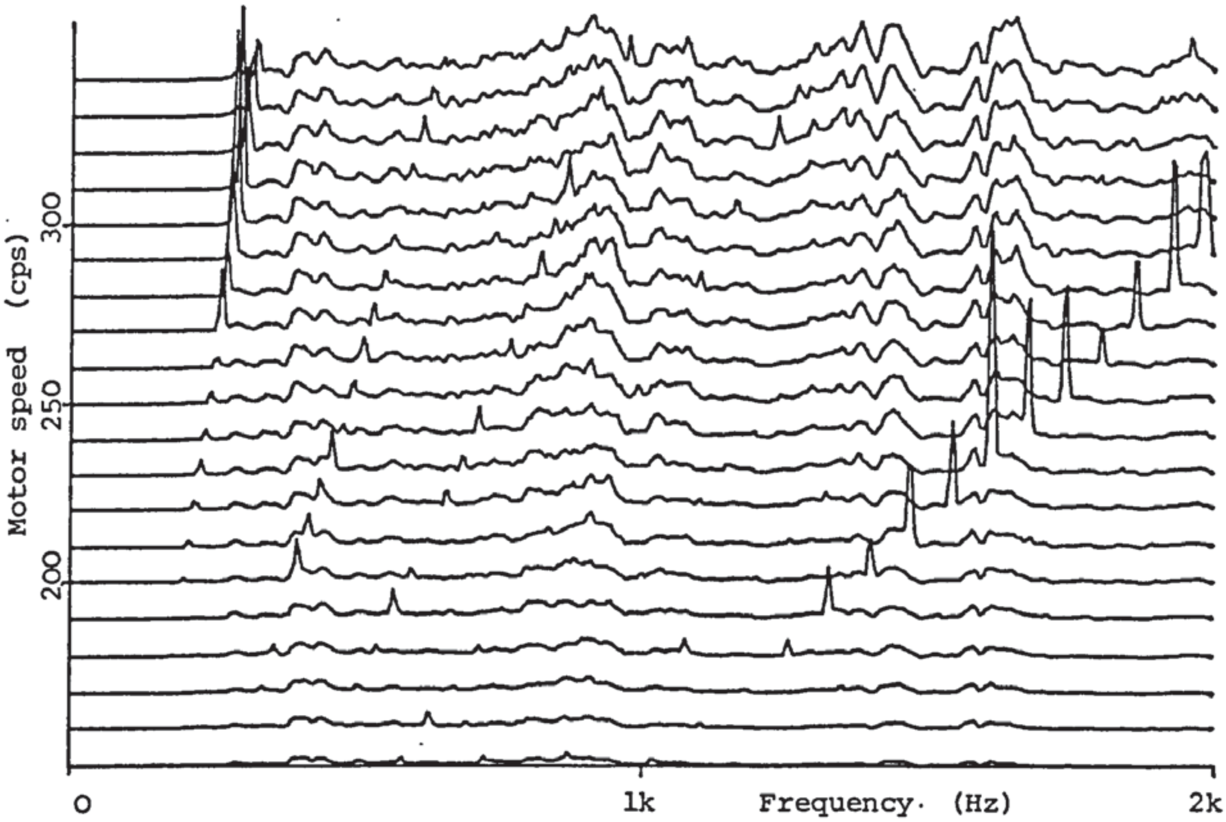


Figure 5.17 Noise spectral map 6; as 5 and without agitator guard.

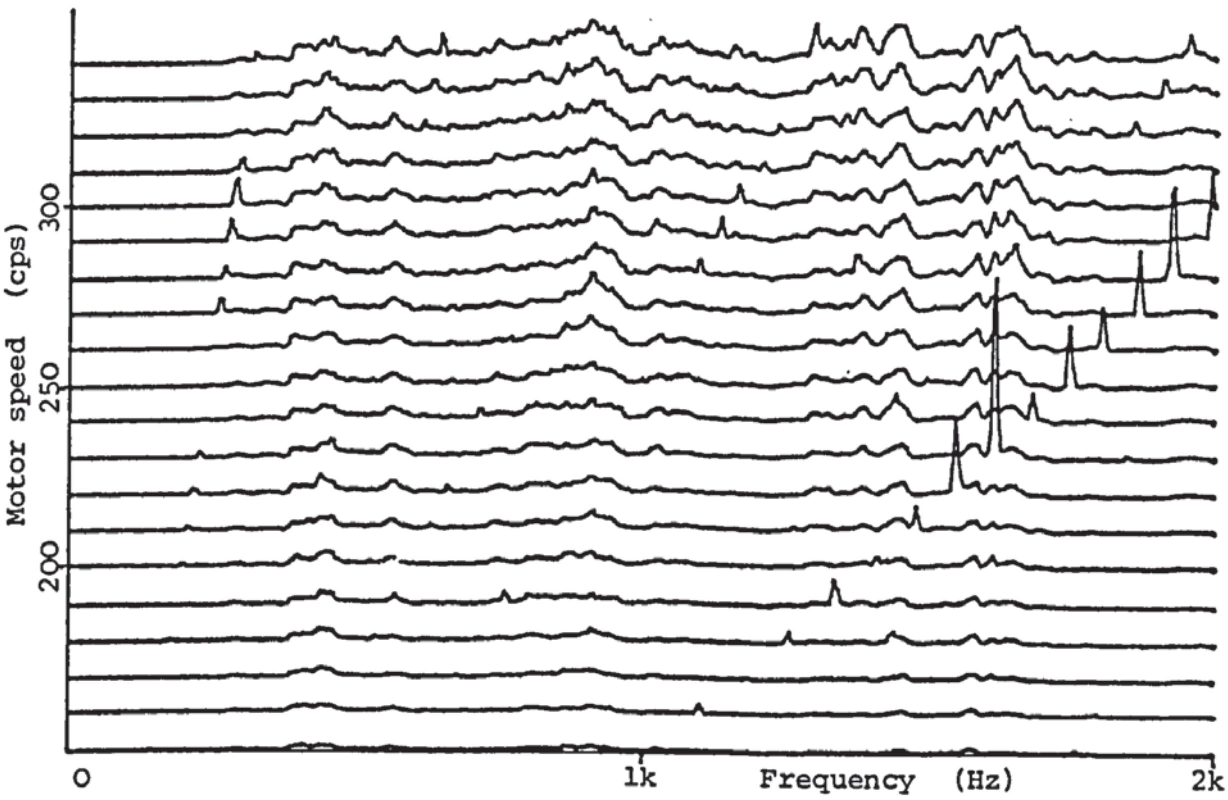


Figure 5.18 Noise spectral map 7; as 6 and without hood.

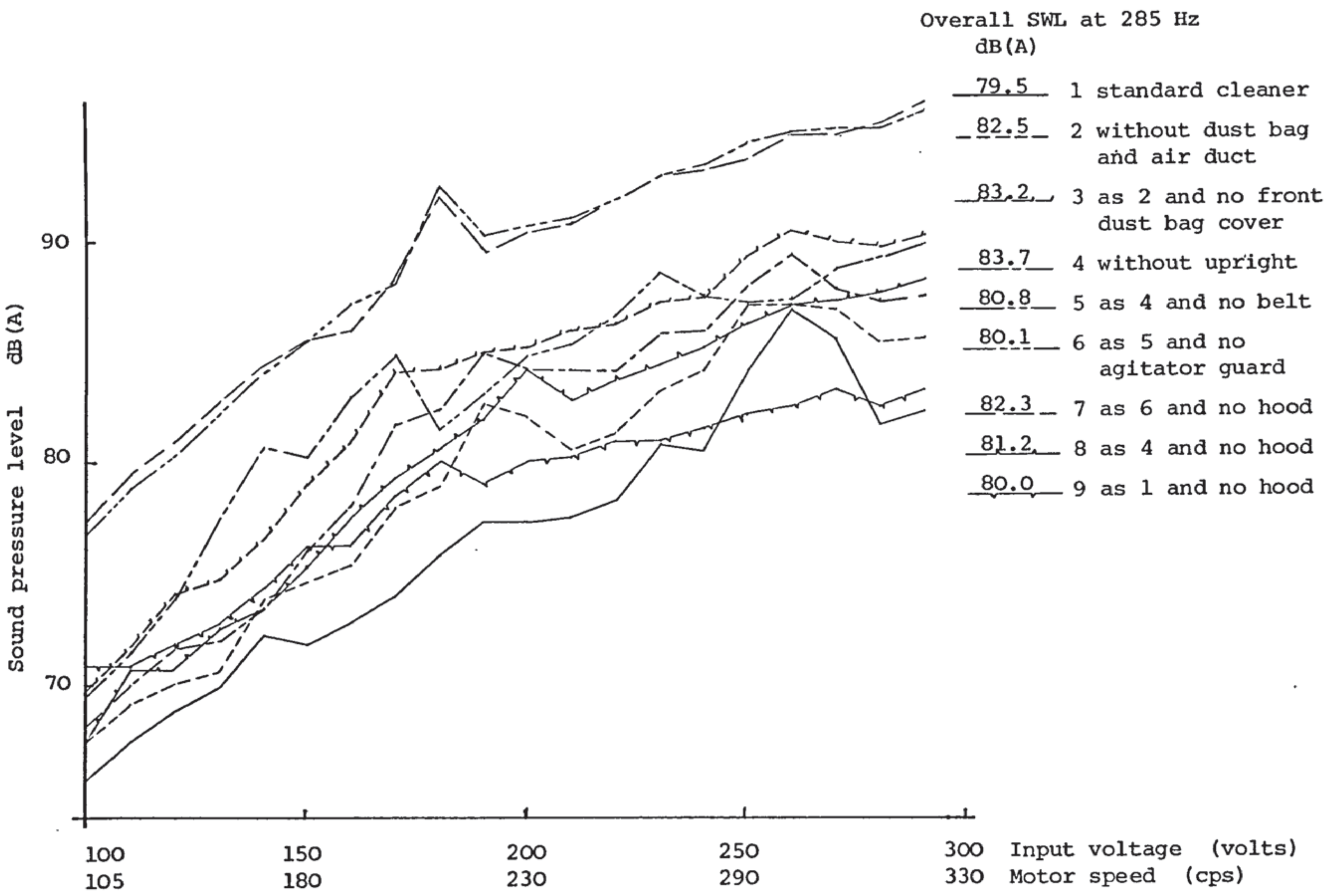


Figure 5.19 Variation in sound level with component removal.

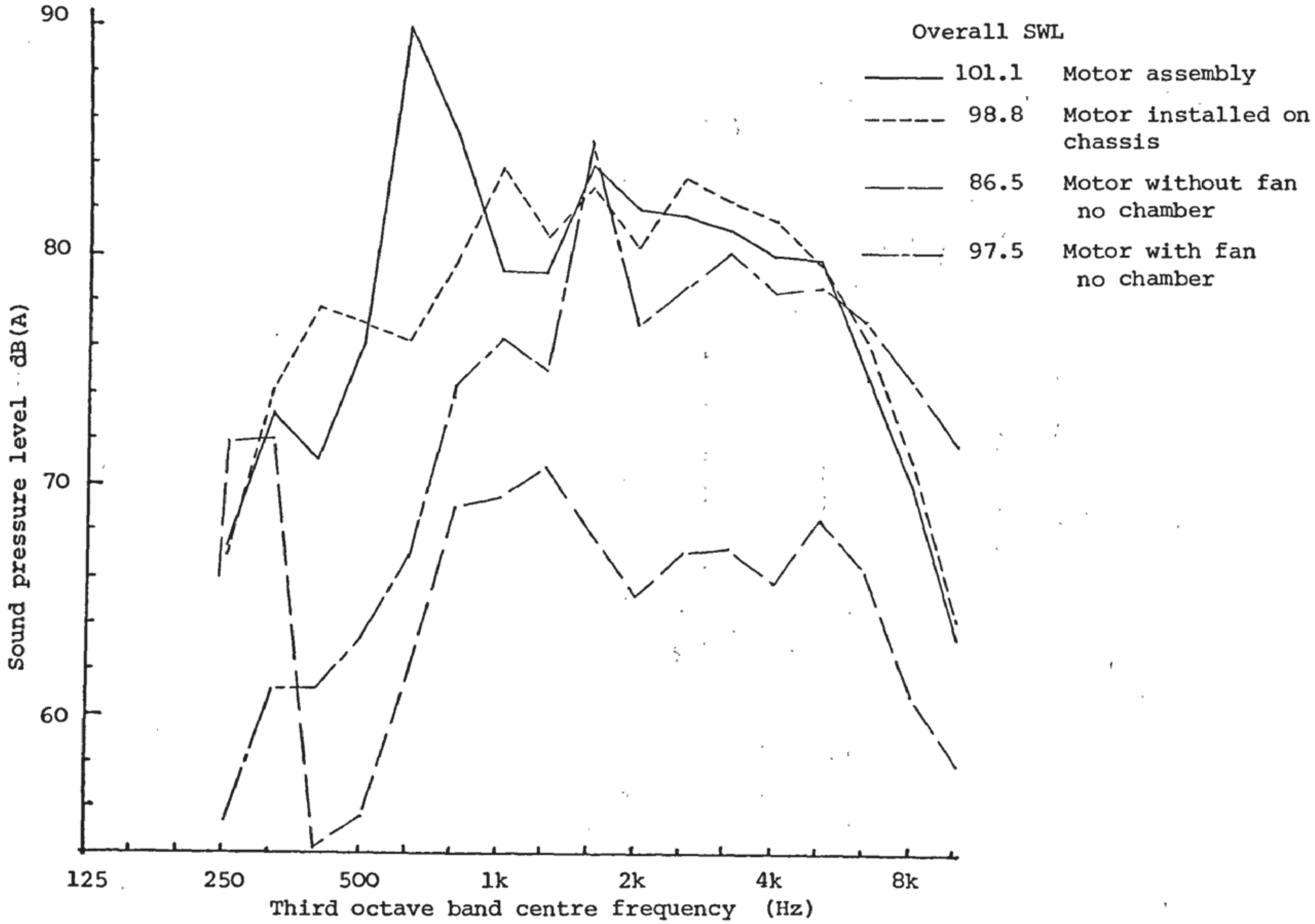


Figure 5.20 Noise spectra for the Turbopower motor.

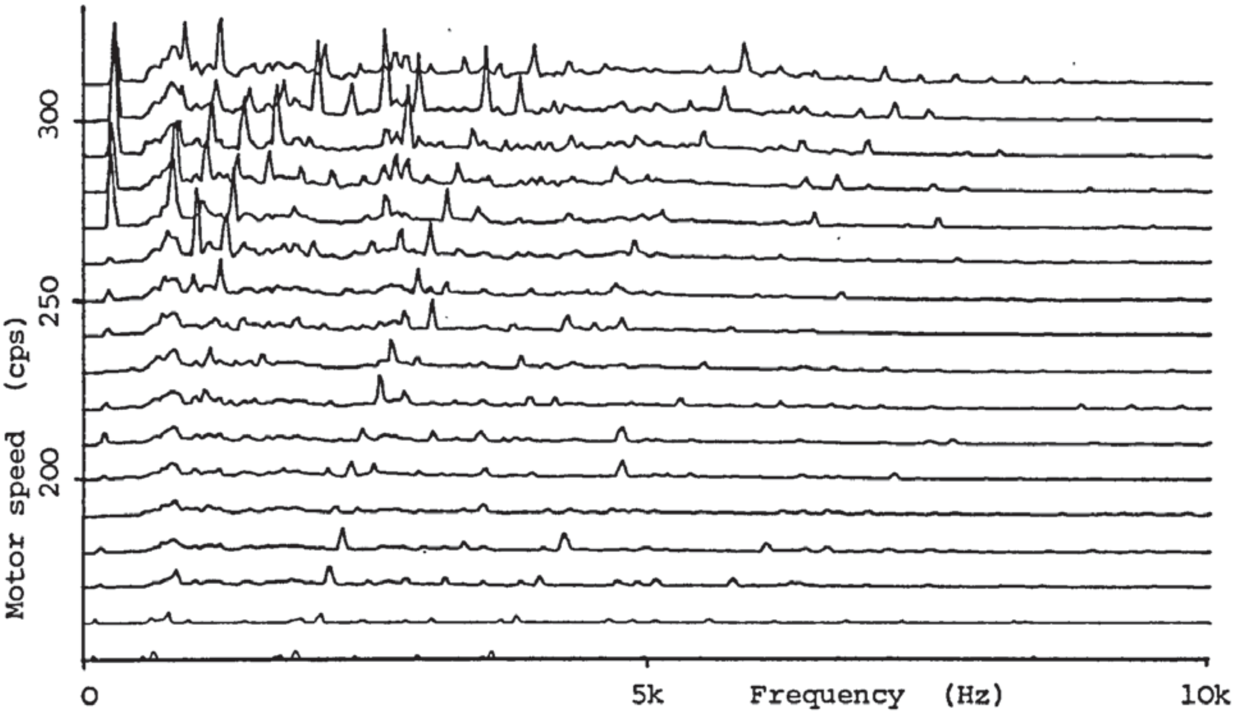


Figure 5.21 Noise spectral map. Motor without fan, without fan chamber.

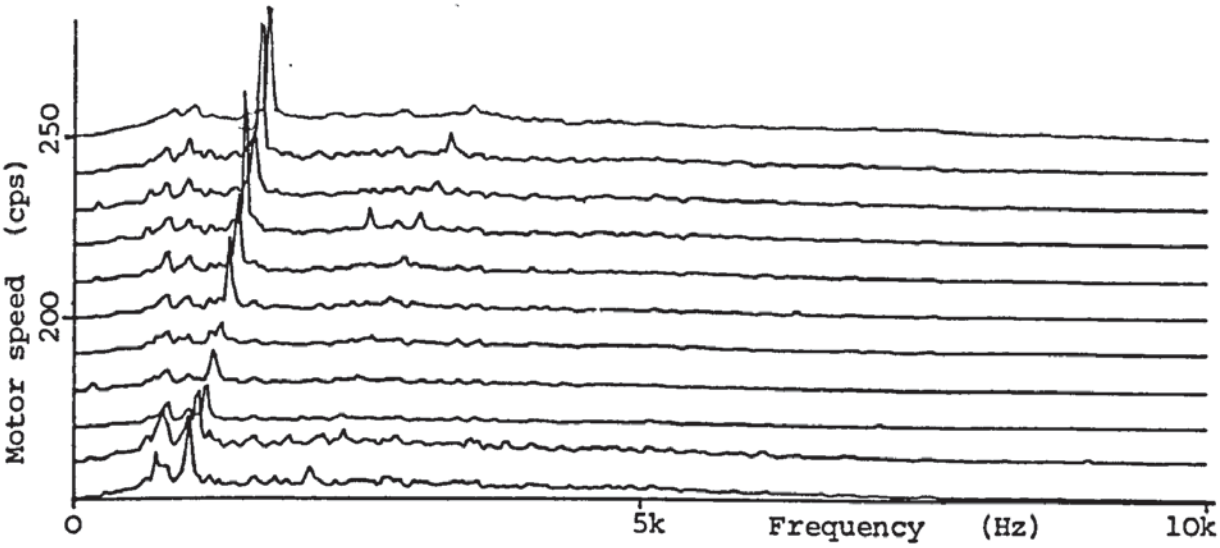


Figure 5.22 Noise spectral map. Motor with fan, without fan chamber.

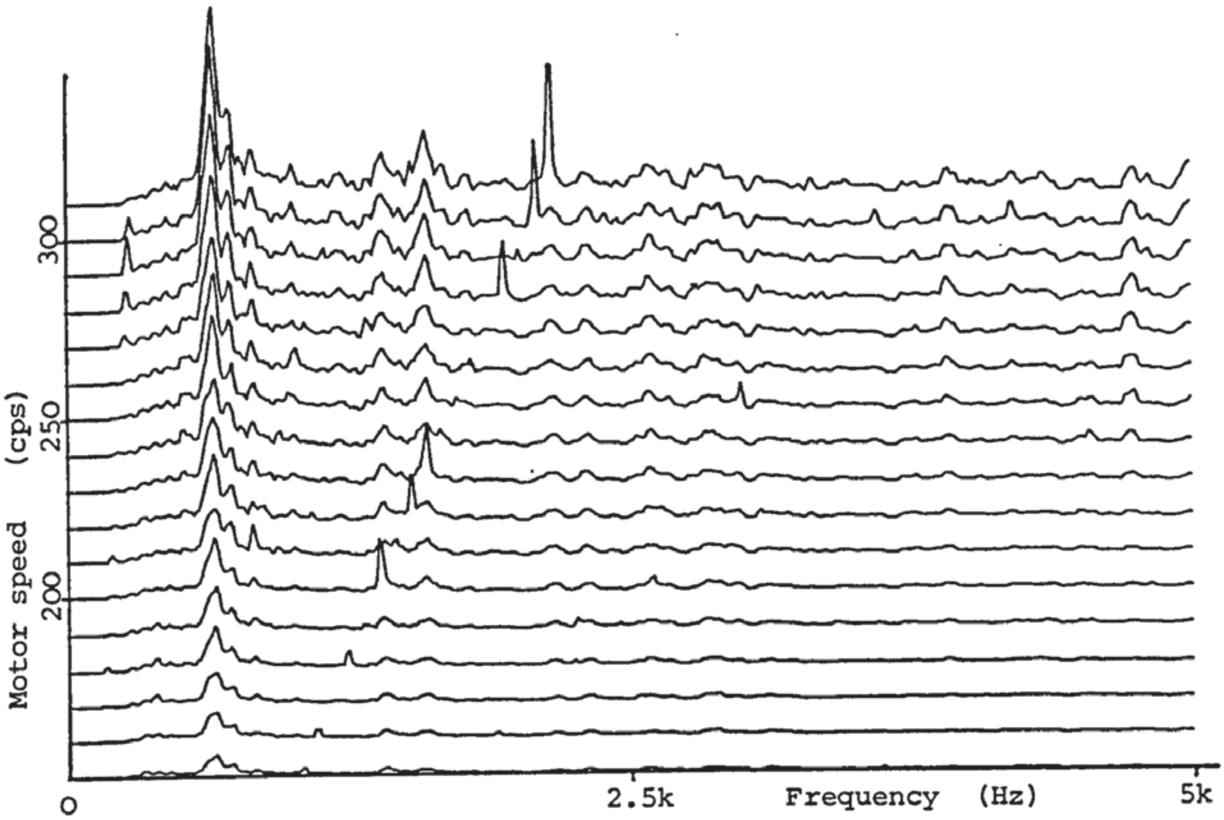


Figure 5.23 Noise spectral map. Motor assembly.

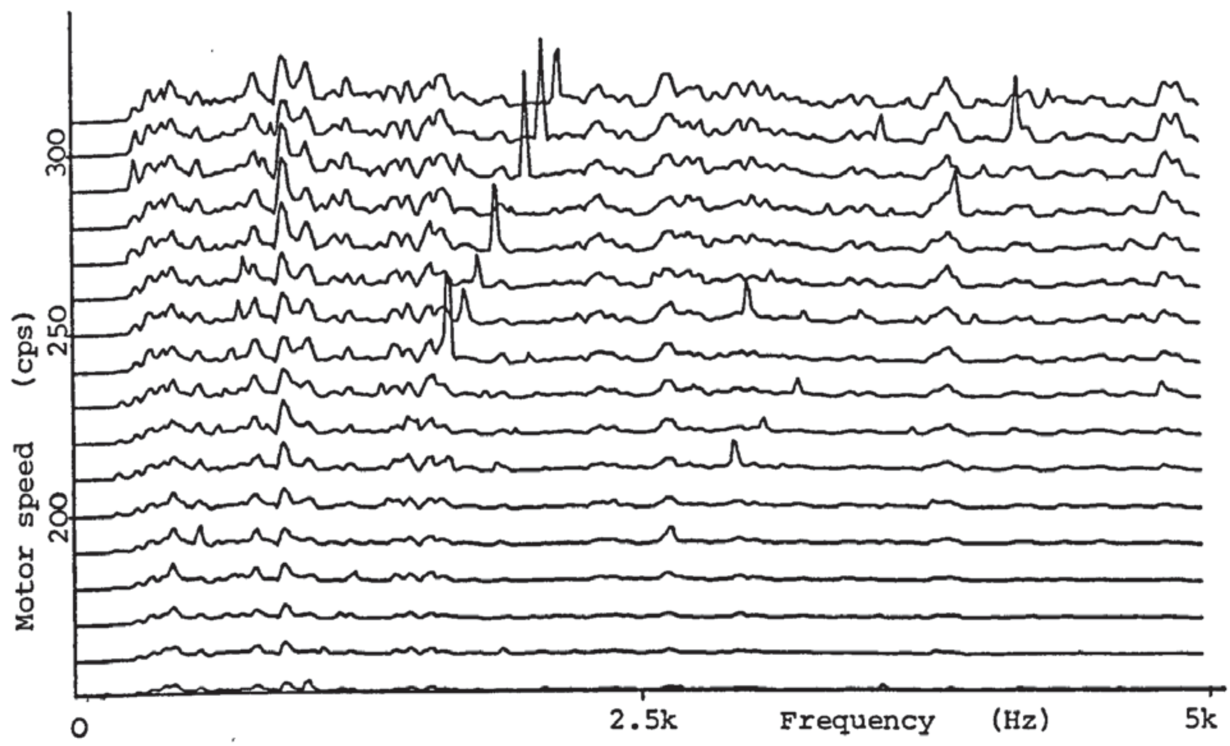


Figure 5.24 Noise spectral map. Motor installed in chassis.

5.5.5 Vibration measurements on an operating cleaner

Even a small amount of imbalance in the armature or its components can cause vibration which, when transmitted through the motor mounts excites the rest of the structure. The structure response is complex because the original forcing functions are modified by the characteristics of the structure, and some are then attenuated while others are magnified.

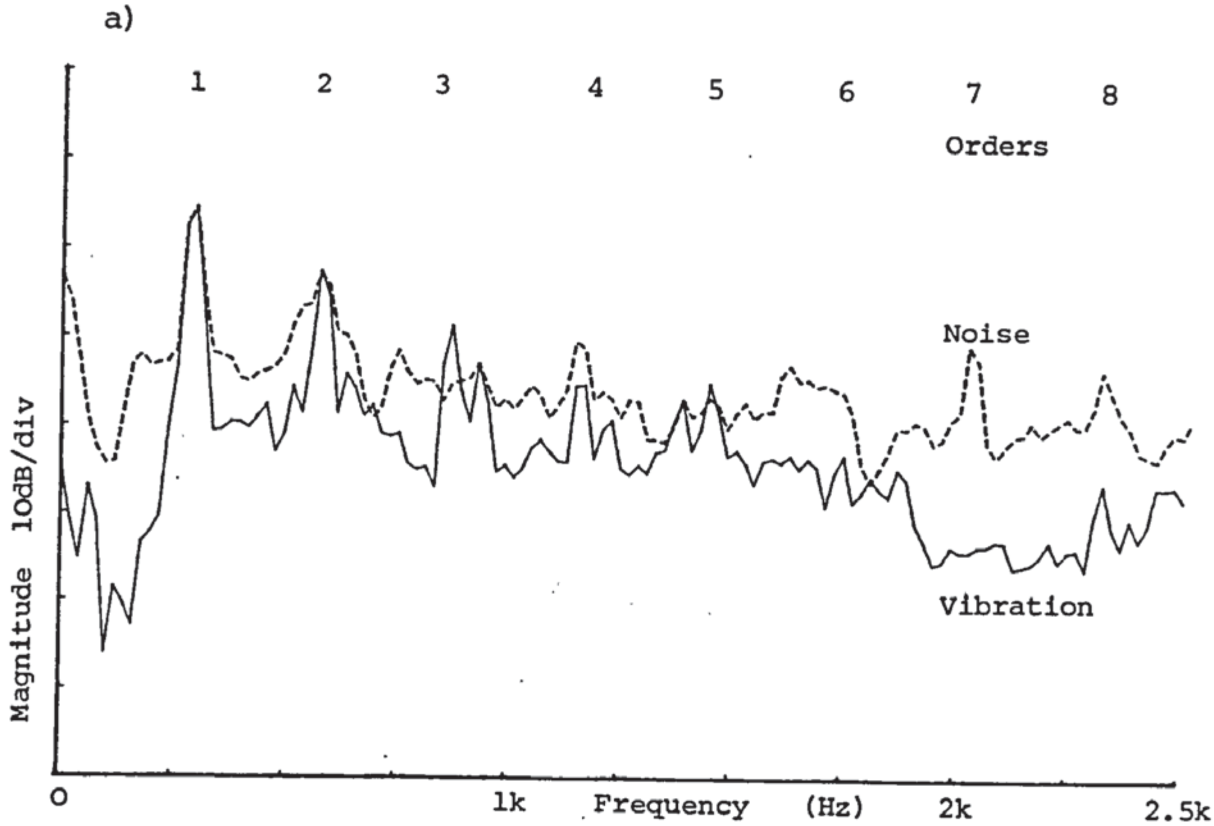
The hood

The vibration response of the hood was measured at two points: at the centre on the top (above the motor) and at the front left hand corner.

Figure 5.25a shows the noise spectrum and the vibration response of the hood at its top centre position. The relative position of each trace is not important as it depended upon the sensitivity setting on the HP 3582A analyser. Both traces are dominated by peaks at frequencies associated with the rotational speeds of the motor and its harmonics (orders), such as 1st, 2nd, 3rd, 4th, and 7th orders (clearly) and a broad peak around the 6th. On the vibration spectrum, the 1st, 2nd, 3rd and 4th orders show clearly but 6th and 7th orders do not appear. In addition, there are three new peaks at orders 3.2, 4.2 and 4.8th. Also, there is an area of generally high response around 200, 500 and 1700 Hz.

The transfer function, relating the radiated noise to the panel vibration, and the phase are given in Figure 5.25b. Unfortunately, the transfer function does not show the transmission paths very clearly. These bad results may be due to the nonlinear and high damping of plastics.

Figure 5.26 shows the noise and vibration response of the hood at its front left hand corner together with the coherence function. The vibration spectrum is very similar to that at the hood top centre. Because this measurement point was close to the agitator bearing, it was expected that vibration at the bearing-related frequencies would be prominent. A study of bearing vibration will be given later in section 5.8.3.



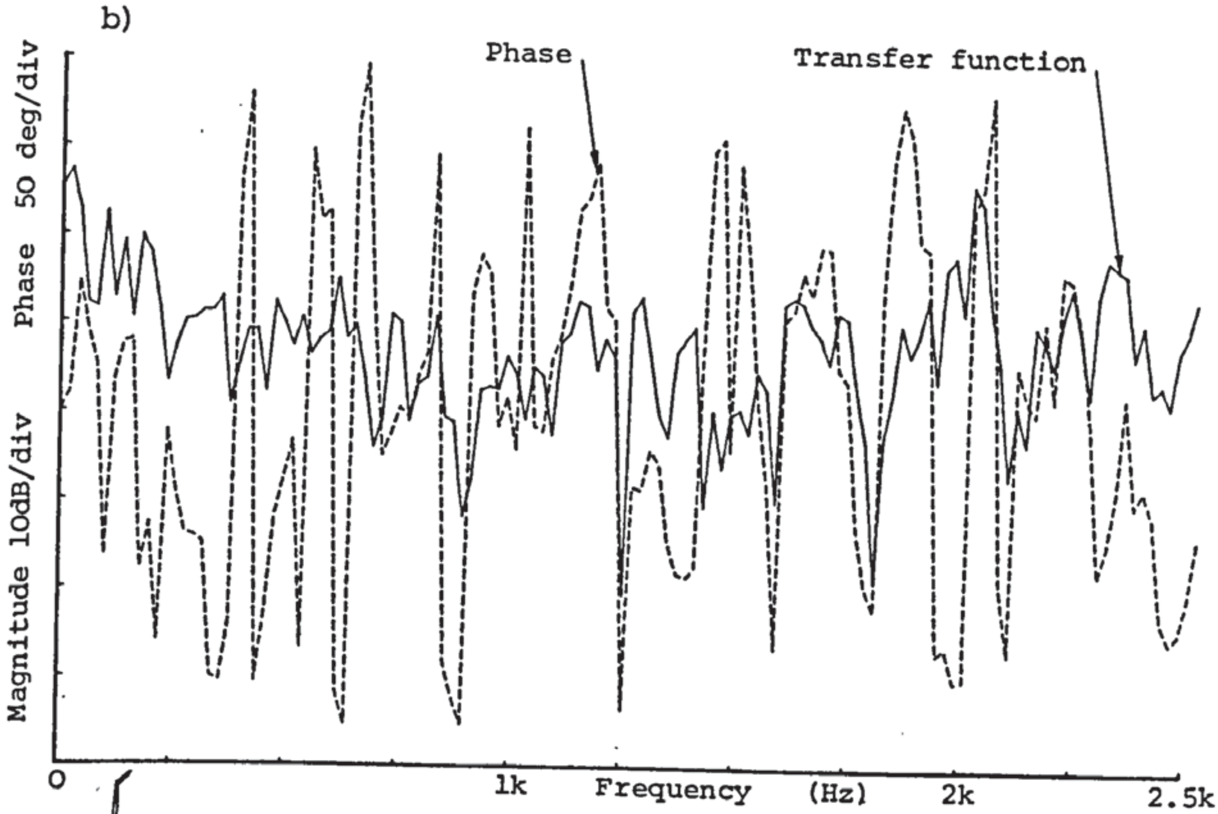


Figure 5.25 Vibration at centre of hood.
a) noise and vibration response
b) TF (noise/vibration) and phase.

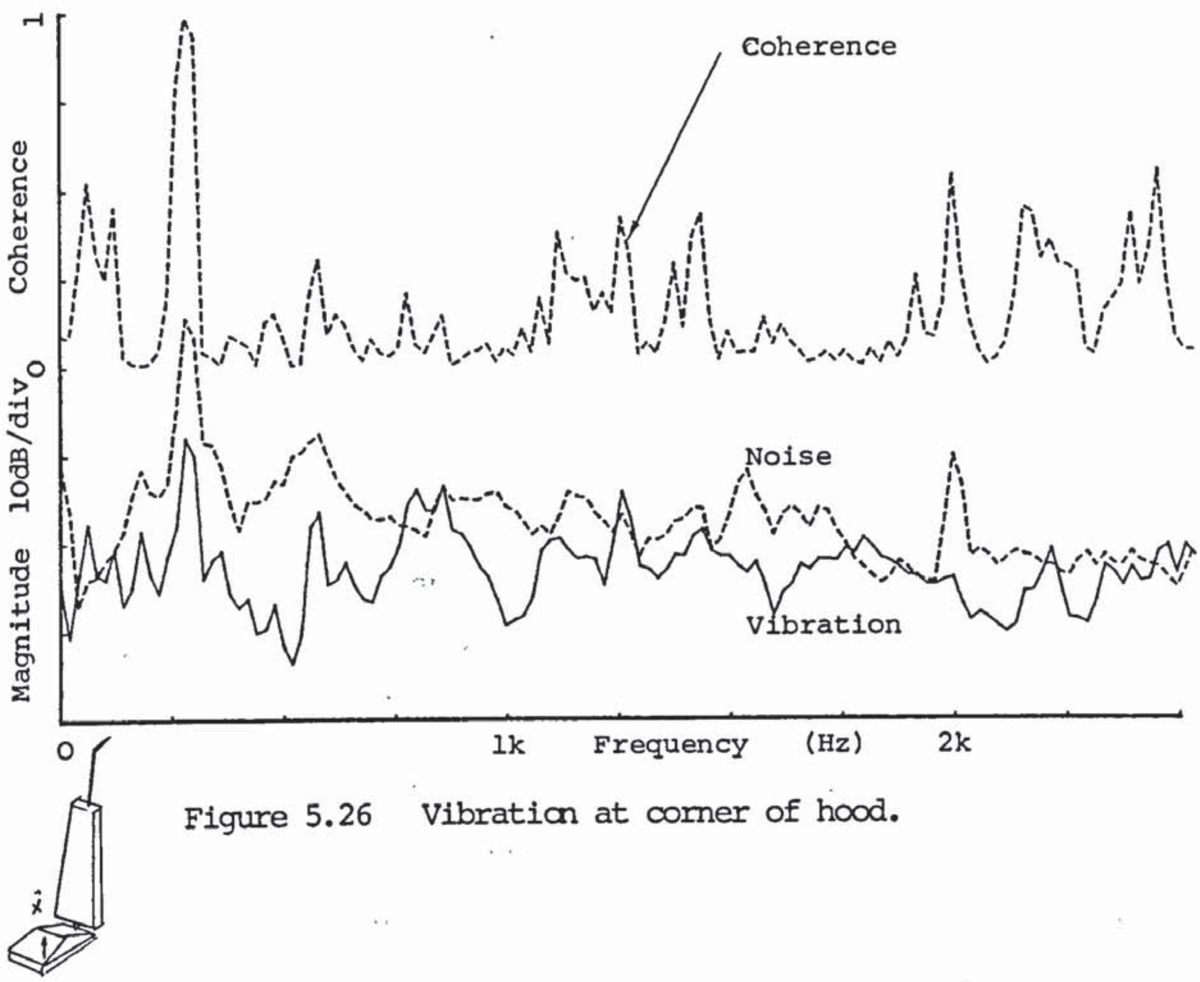


Figure 5.26 Vibration at corner of hood.

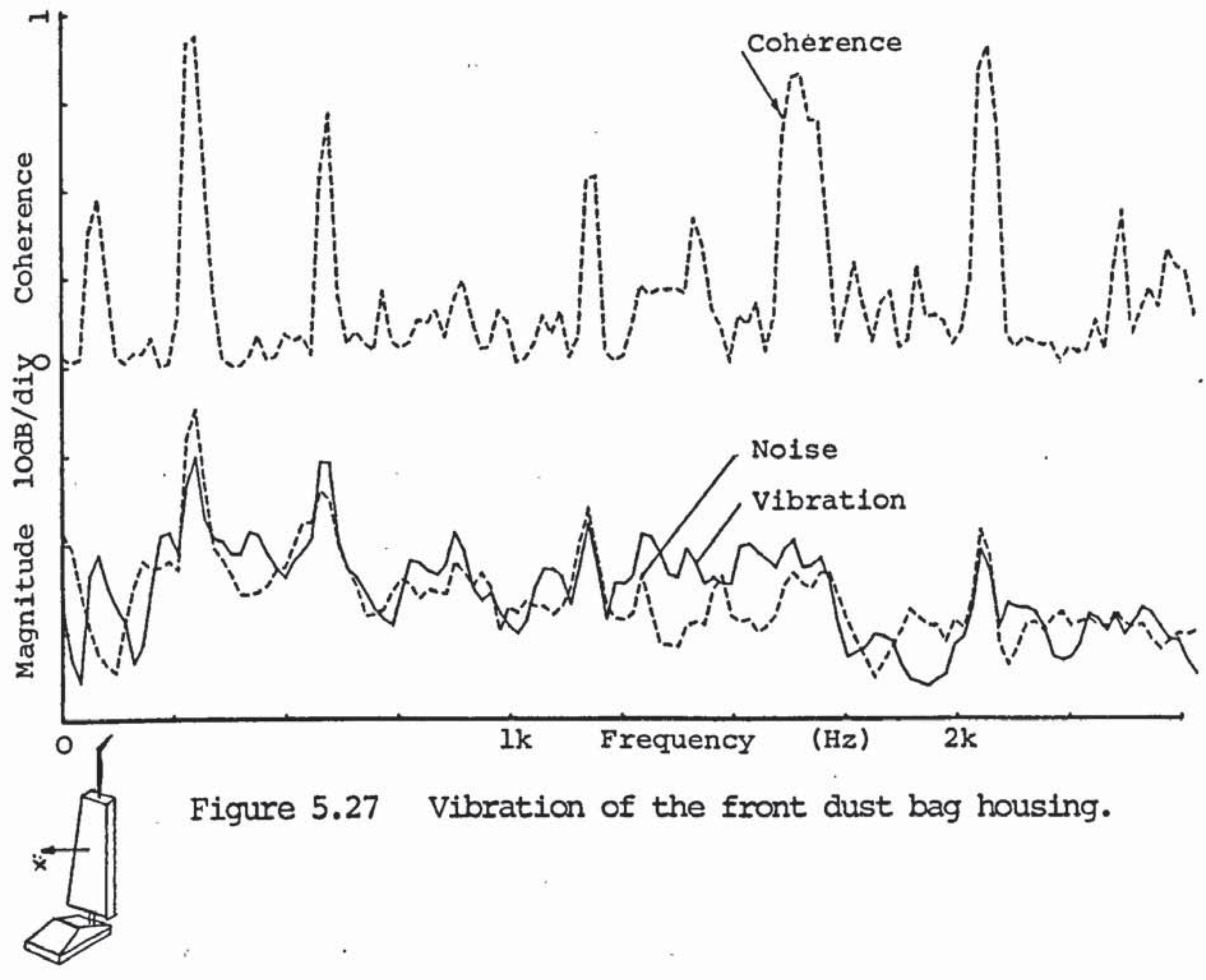


Figure 5.27 Vibration of the front dust bag housing.

Front dust bag cover

The vibration response of the front dust-bag cover at its centre is given in Figure 5.27 and shows many common peaks. The coherence function shows good correlation at the motor rotational speeds of 1st, 2nd, 4th and 7th orders. Notice the correlation around broad-band peaks of frequencies 1340 and 1640 Hz. The vibration response of the rear dust bag cover was similar to the front cover, as shown in Figure 5.28.

The air duct

The coherence function in Figure 5.29a shows that in addition to the 1st, 2nd, 6th and 7th orders there are many other peaks in the vibration response which have good correlation. This activity suggested that the emitted noise comprised mainly the airborne and air-flow noise. By comparing the results of the front and rear dust bag covers and the air duct, there is a strong suspicion that the whole upright of the cleaner pulsates as a box. The major pulsating frequencies are air-flow related. The 7 blade suction fan is a source that generates pulses in the air-flow downstream and throughout the ducting.

Interesting information was obtained when the dust-bag was removed from the cleaner. The vibration of the air-duct, measured with and without the dust-bag, indicates that the dust-bag has damping properties. This is shown by smooth, broad peaks.

Agitator bearings

Vibration measurements close to the agitator bearing support on both sides showed there was greater vibration level at high frequency, yet the coherence function showed the 1st order as the only one exhibiting good correlation, see Figures 5.30a and b. Consequently, although bearing vibration was present it does not contribute much to the emitted noise.

The cleaner motor

Figure 5.31a shows the response of the Turbopower motor. Vibration measurements were made on the casing and noise was measured in the normal

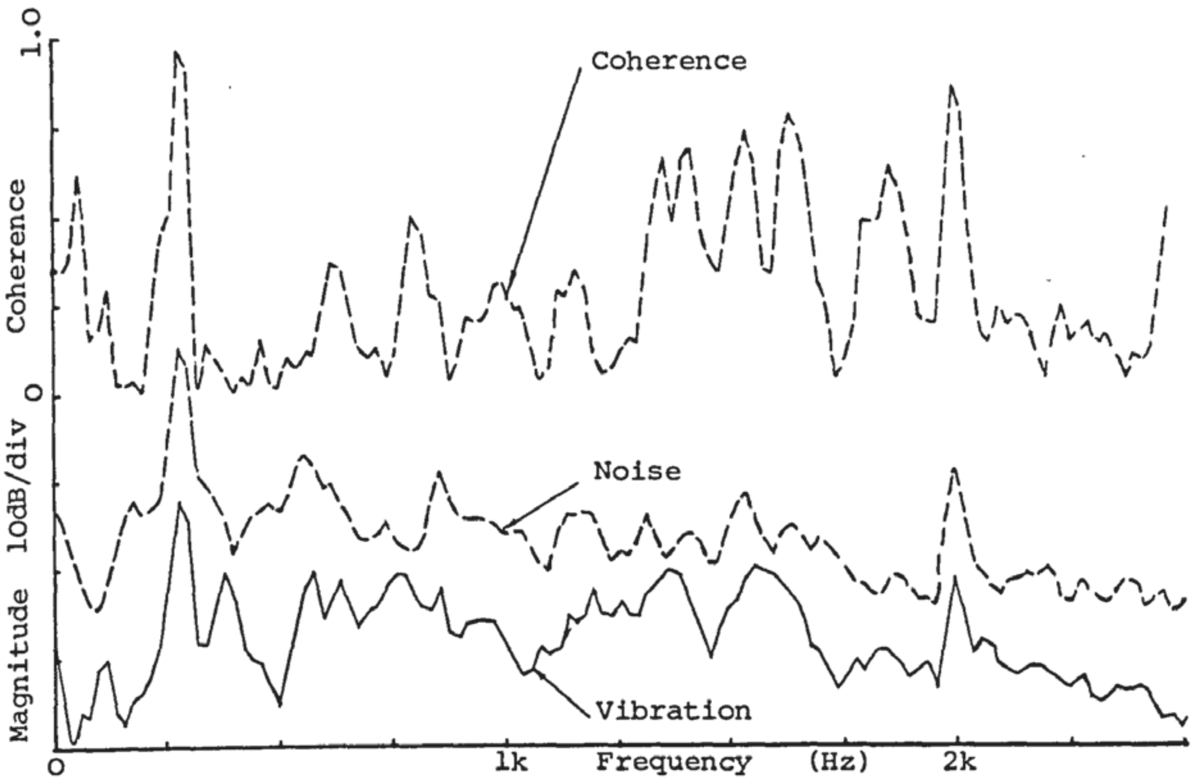
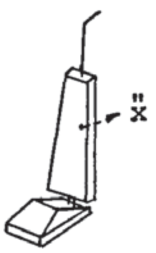
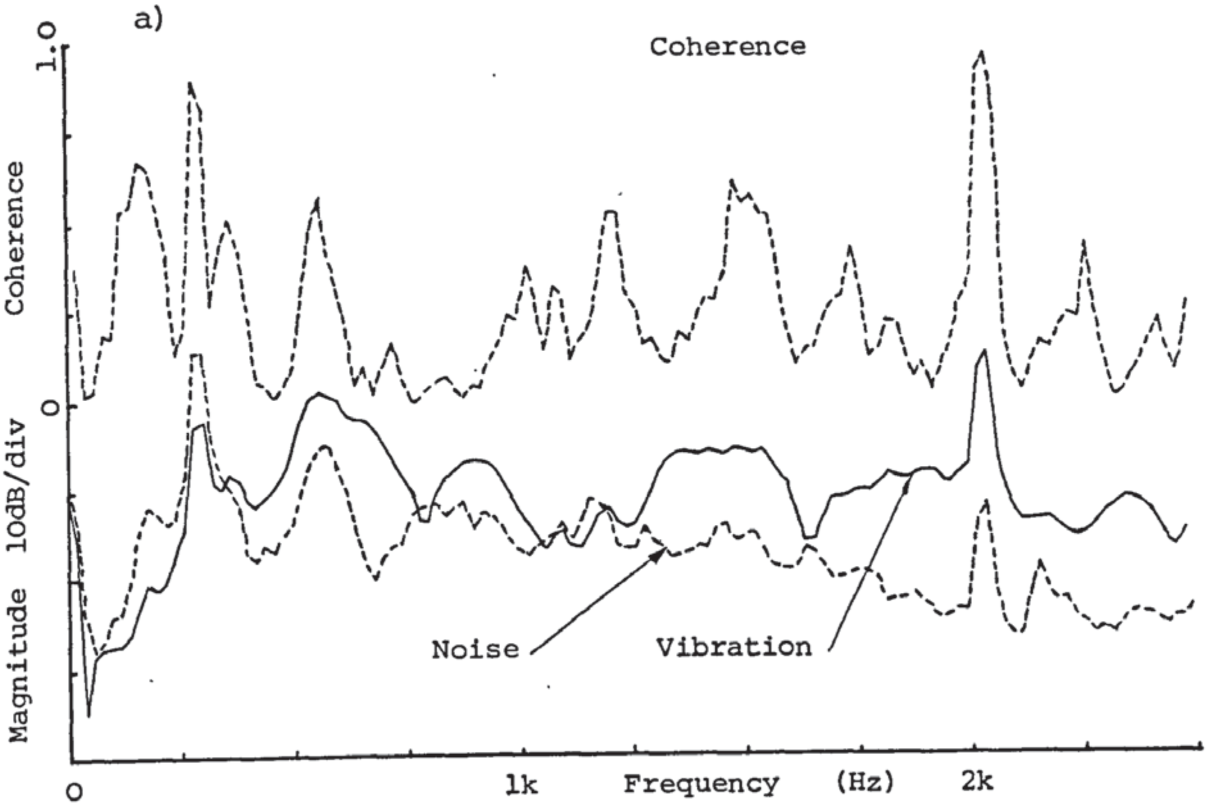


Figure 5.28 Vibration of rear dust bag housing.





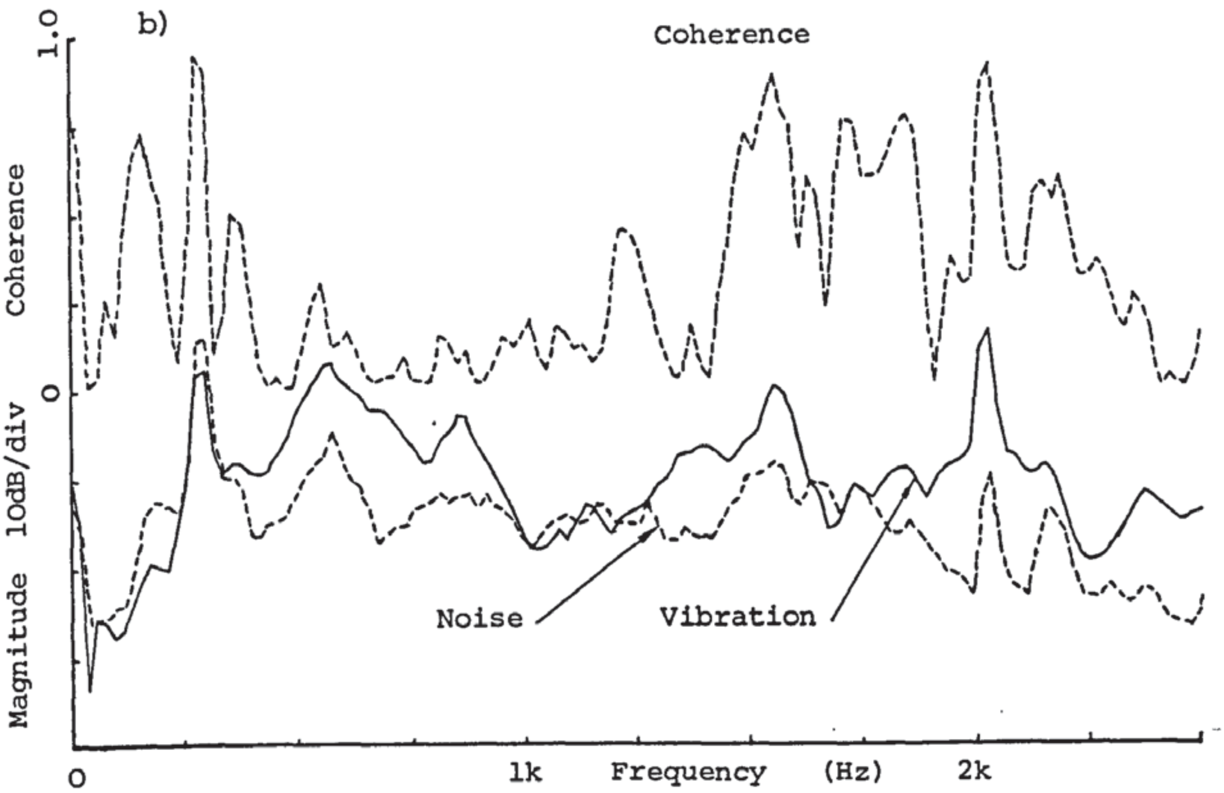


Figure 5.29 Vibration of the air duct a) with dust bag
b) without dust bag.

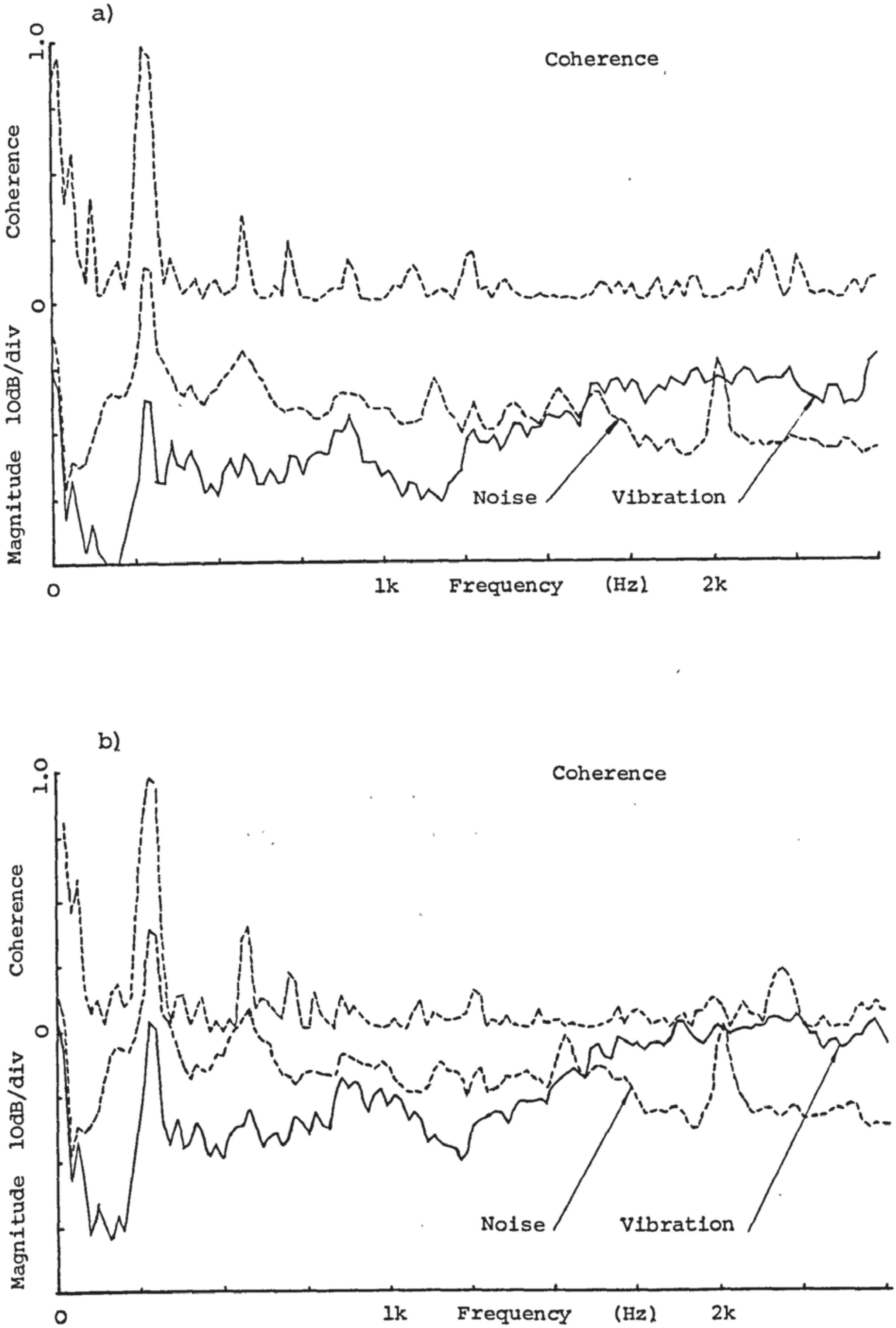


Figure 5.30 Vibration at a) nearside and b) offside agitator bearing support.

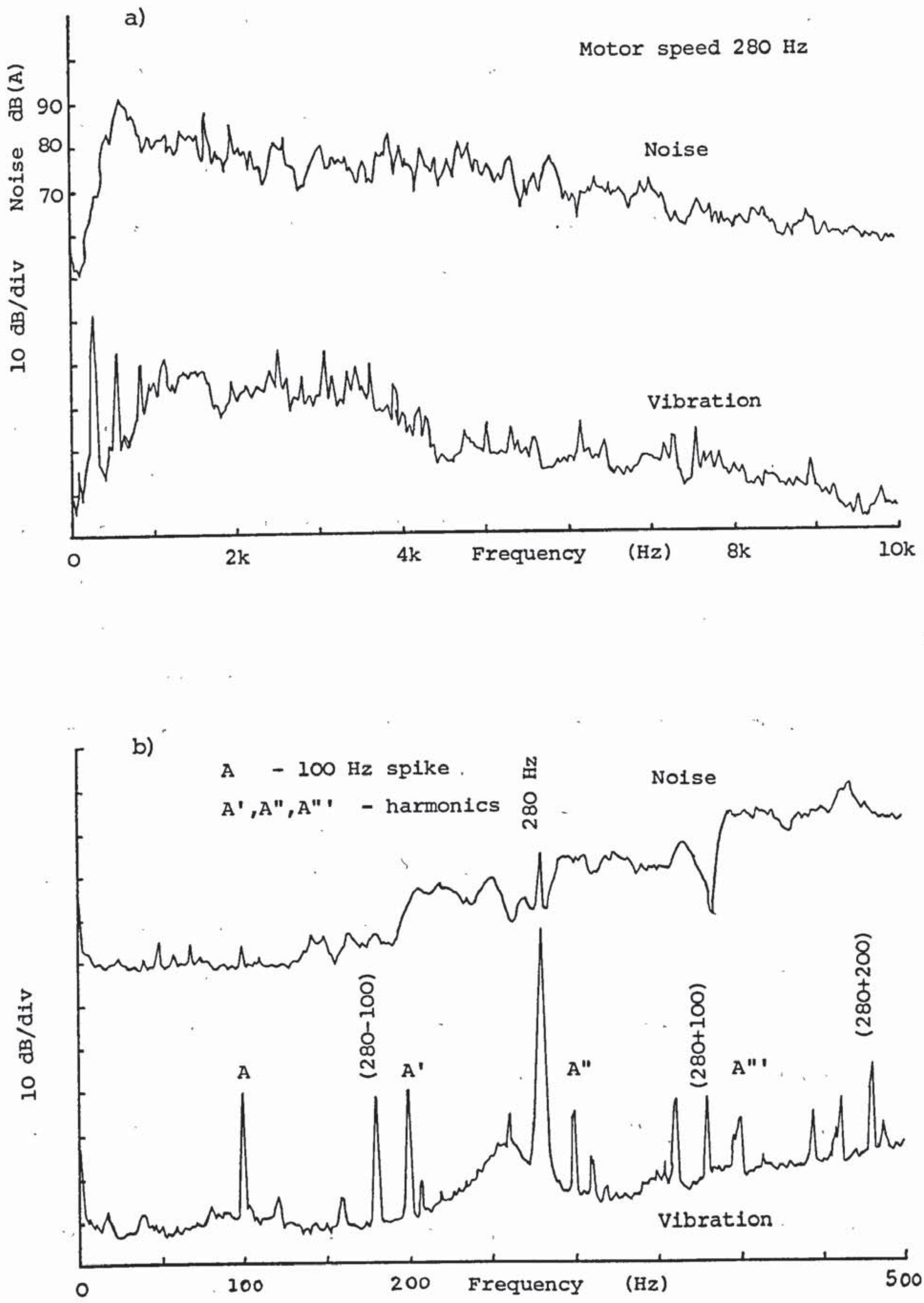


Figure 5.31 Motor noise and vibration spectra a) 0-10k
b) 0- .5k

manner at Perivale. The A-weighted noise spectrum shows a peak of maximum amplitude 615 Hz and then generally falls with increasing frequency. The vibration response is much peakier, showing the fundamental and its harmonics clearly. A zoom measurement, in the range 0-500 Hz, of the vibration response given in Figure 5.31b, illustrates the "peaky" nature. Notice a series of peaks based upon the 100 Hz spike and its harmonics. The vibration peak due to armature out-of-balance has been modulated by the 100 Hz spike which produces additional peaks as indicated.

5.5.6 Vibration testing of components

This section details the results of vibration and impact testing of cleaner components to determine their resonant frequencies.

The air duct

The air duct, with an accelerometer affixed at its mid-position, was hung by a light inextensible string and excited using an instrumented hammer. The response in Figure 5.32 clearly shows many resonances. As some of the peaks are round and broad this implies high damping. The results of impulse testing agreed well with random noise tests using a vibrator. The coherence function indicated that the measurement was totally acceptable.

Front and rear dust-bag covers

The front cover was suspended by a string and the response, shown in Figure 5.33a, was measured at its centre whilst the excitation by the hammer was at the mounting point at the lower end. The rear cover complete with the handle and the flex, was also tested and showed many peaks.

The hood

The hood was suspended by a string. The accelerometer was fixed on the upper side and the excitation was at one of the mounting points. A number of peaks are shown on the response given in Figure 5.34 whereas the extent of damping is again illustrated by smooth and rounded peaks in the zoom measurement in the frequency range 350-600 Hz.

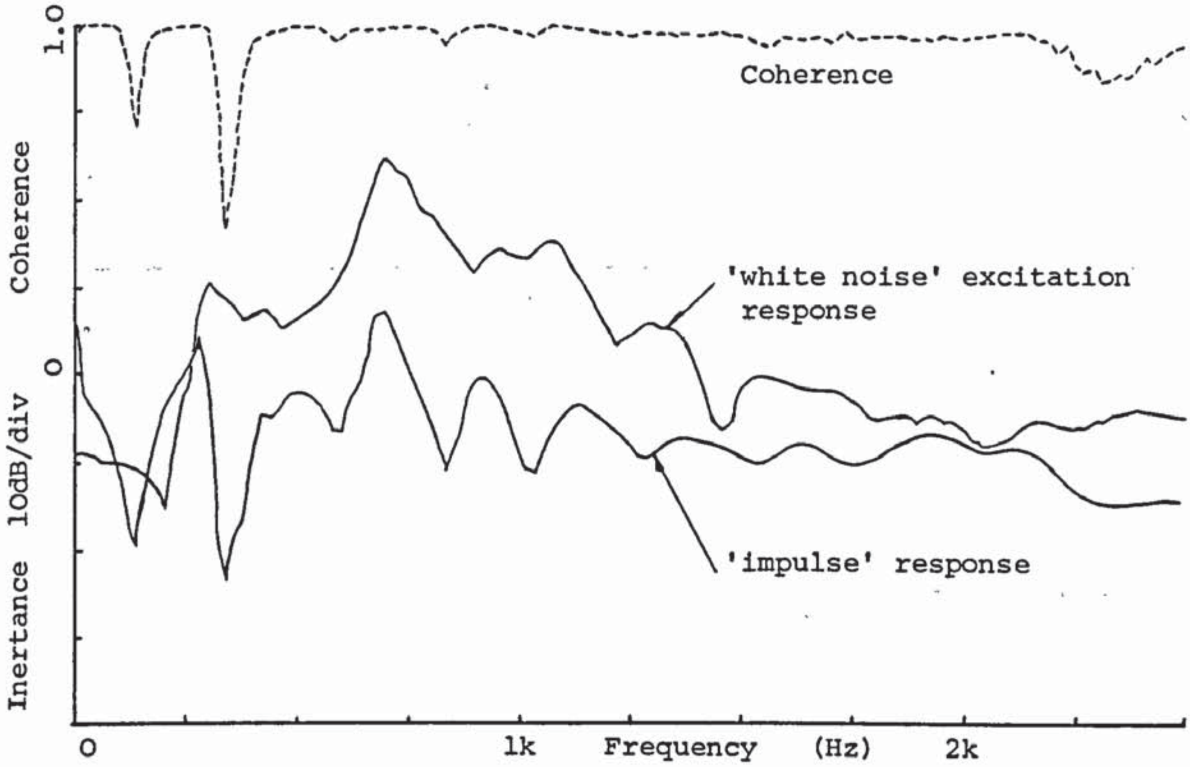


Figure 5.32 Inertance response of the air duct.

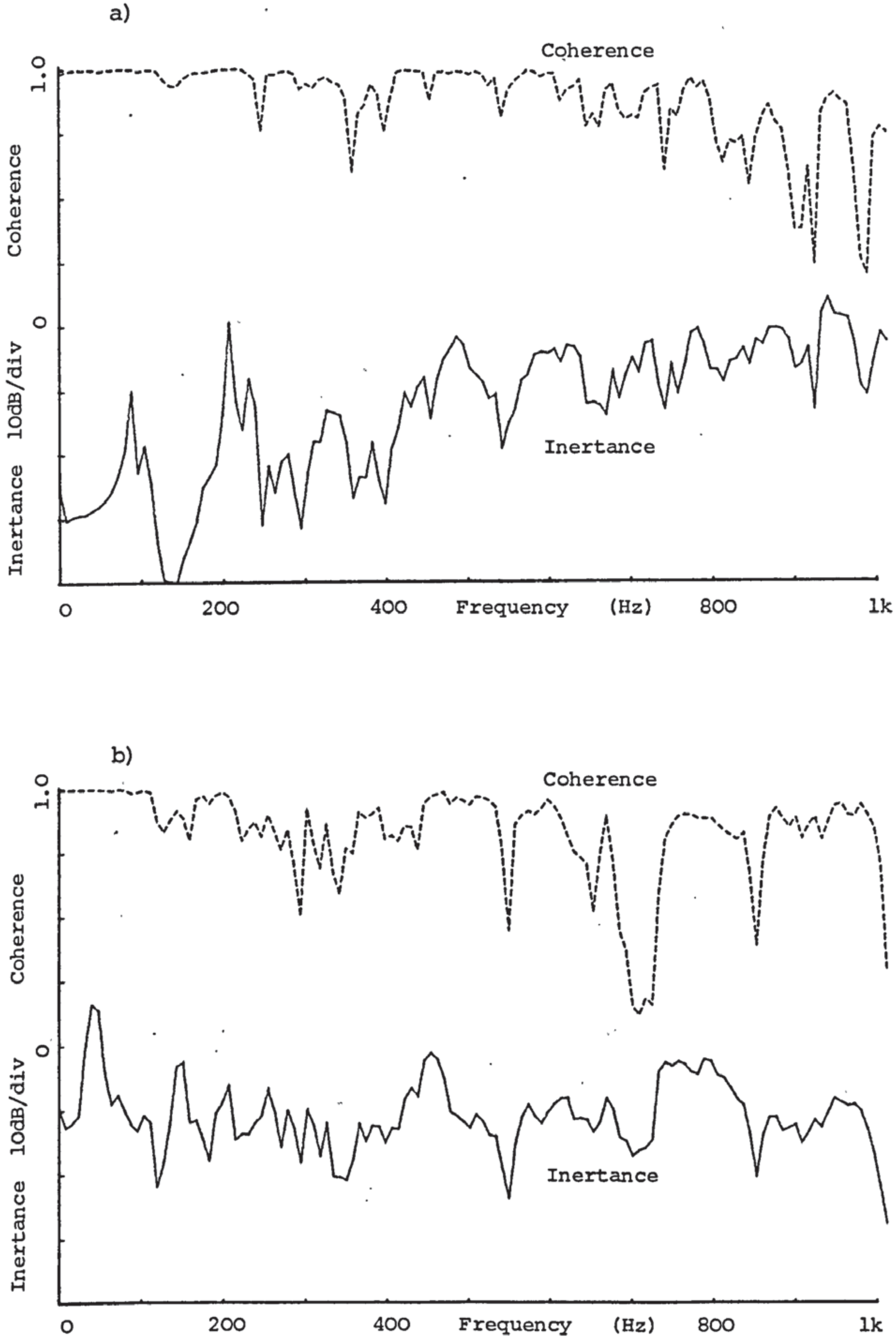


Figure 5.33 Inertance response of a) front and b) rear dust bag housings.

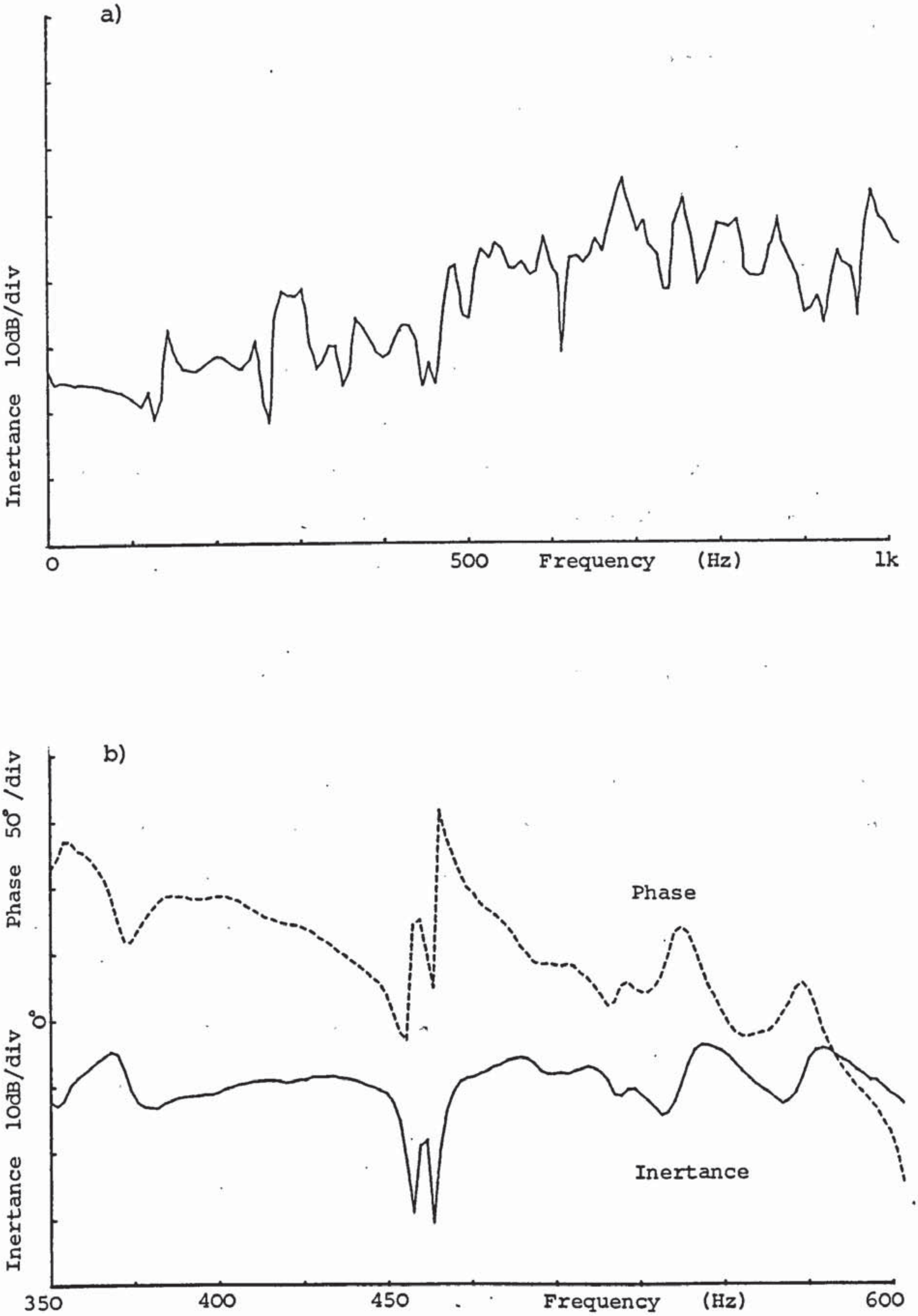


Figure 5.34 Vibration response of the hood a) 0-1k Hz
b) 350-600 Hz

To evaluate the repeatability of measurements, the above test was repeated some days later. The result, shown in Figure 5.35, is similar to Figure 5.34a at low frequency but there is a lot of discrepancy at high frequency. This discrepancy was quite alarming. Not surprisingly, these results were similar to findings of a Round-Robin survey reported by Ewins and Griffin (1981). Subsequently, further tests were conducted to check limitations of the vibration testing technique as applied to non-linear plastic components. These tests were made on the cleaner chassis.

The chassis on its own was hung by a string. The excitation and response points were selected so that they would remain unobstructed when all components were assembled onto the chassis. The inertance response is given in Figures 5.36a and b. Response of the chassis alone shows numerous peaks which are clearly defined. This suggests little damping in the chassis structure. The coherence function, which is not included, indicated the measurement was totally acceptable. When all of the components, such as the motor, wheels and the agitator were assembled onto the chassis the result underwent a radical change. The peaks do not coincide and high damping was present. Consequently, vibration testing of individual components was considered inconclusive and inappropriate.

The author had found (Sagoo 1979), that a metal system, when tested within linear limits, showed a good correlation for responses measured at two separate points. Plastic components were in complete contrast, as revealed by comparing Figures 5.37 and 5.34a which show responses for the hood measured at two separate points. Therefore, this technique was deemed unsuitable for application to lightweight plastic components.

5.5.7 Coherence measurements

The coherence function of some of the measurements has already been presented. It was used in the first instance to validate a particular set of data before recording the results and then, to identify the source by

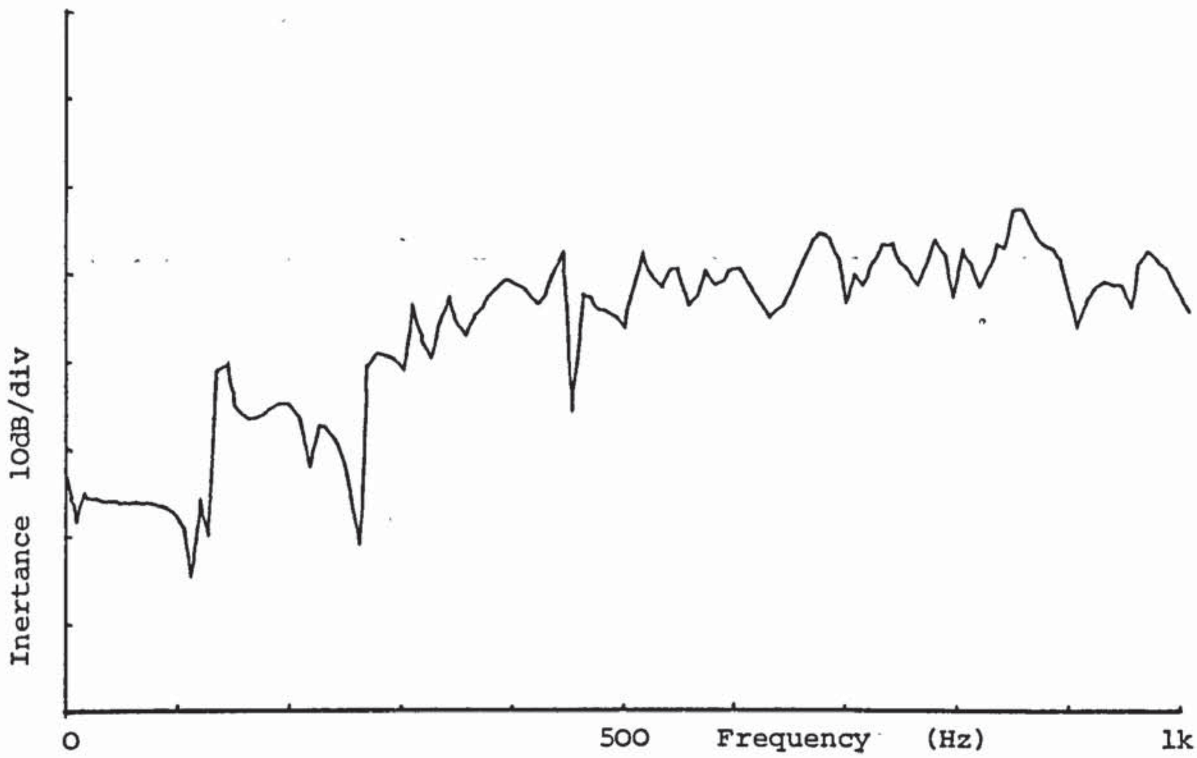
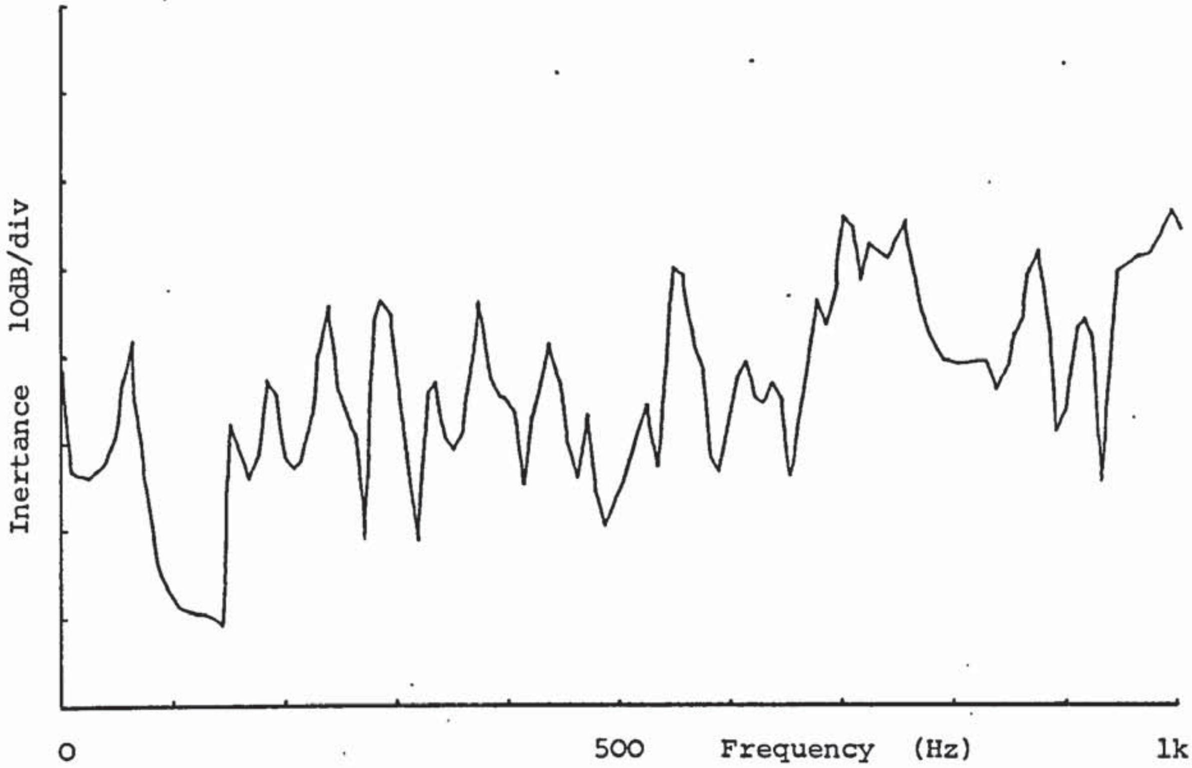


Figure 5.35 Vibration response of hood.

a) Chassis on its own



b) Chassis assembly

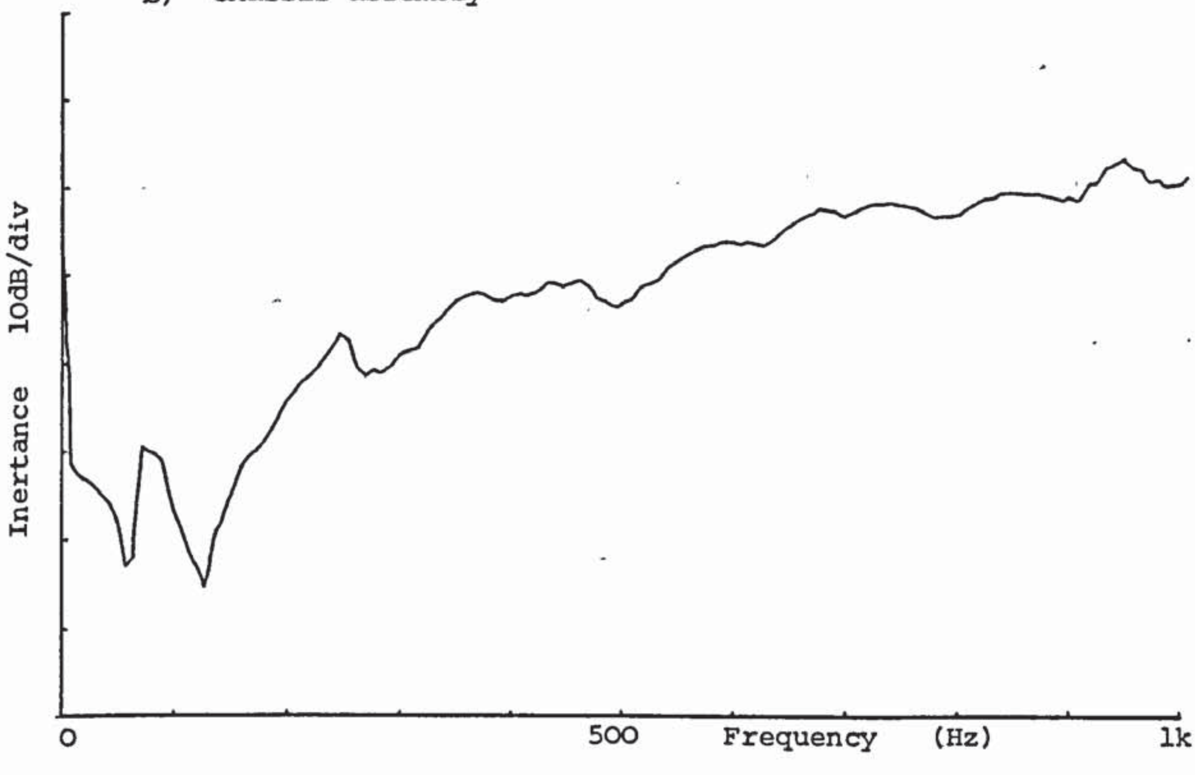


Figure 5.36 Vibration response of the chassis showing influence of changes in loading conditions.

Figure 5.37 Vibration response of the hood at an alternative to that for Fig 5.35.

considering the correlation between the two measured signals. Conclusive information was obtained which primarily helped to determine the pulsation of the dust-bag housings as a source of radiated noise.

5.5.8 List of major noise sources

Motor: out-of-balance and harmonics

100 Hz spike and its modulation of higher orders

bearing vibration frequencies

630 Hz fan chamber resonance.

Air-flow noise: due to the seven blade centrifugal fan

due to the 19 blade cooling air fan

due to a sharp right-angled bend

Panel vibration. That is, the hood

bag housing covers, front and rear

wheel trims

Radiation of noise via orifices in the hood

through the bellows

via the gap between floor and nozzle at front of the cleaner and the chassis at rear.

5.6 Ranking of major noise sources

All of the above mentioned sources contributed to the overall noise of the cleaner. That contribution was either to generate a pure tone in the frequency spectrum or broad-band noise.

A lower overall noise level is achieved by noise reduction of the worst source in a cyclic process. Reduction in an intermediate noise source would not appreciably alter the overall noise even though there may be significant drop in the noise output of that source. Consequently, to commence a noise reduction programme these sources of noise and vibration were ranked in order of priority. Ranking of noise sources is not an easy task! and it was decided to determine the contribution of a particular source by isolating one source at a time and noting the overall noise.

The armature imbalance

The influence of the armature imbalance on the cleaner noise was clearly illustrated by measurement of two identical Turbopower cleaners, the standard hardbag model U2194 and a model U2198, which comprises the autoflex facility. Results are summarised thus;

	Sound power levels dB(A)	
	U2194	U2198
Standard cleaner	87.0	92.2
Motor only on V-block	100.8	101.0
Cleaners with motors exchanged	92.8	86.8

Apparently, the motors on their own have similar noise levels and it follows from the above results that motor belonging to cleaner model U2198 contributes to the excessive noise of the model U2198. Therefore, the motors must differ in their vibration properties.

Component related

The contribution due to the hood, wheel trims, bag housings and air passage over sharp edges was determined by removal of the component and noting the cleaner noise.

Air-flow related

The influence of the air-flow noise was determined in two ways. Firstly, the air-flow into the cleaner was restricted by blocking the air inlet opening below the agitator. Secondly, the suction fan was removed thereby eliminating the air-flow altogether.

The results of the above exercises are illustrated in Figure 5.38.

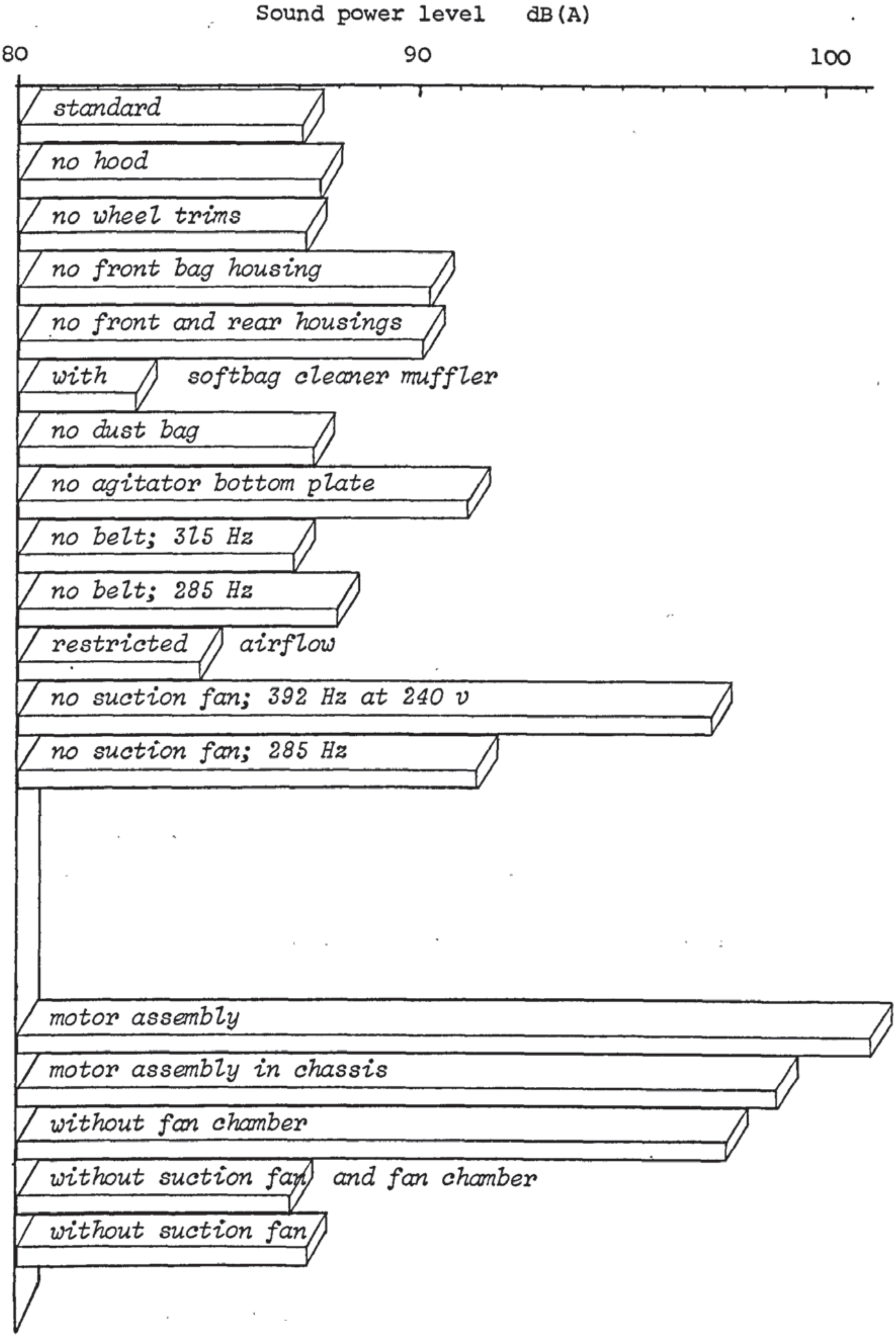


Figure 5.38 Cleaner and motor sound levels for component removal.

Ranking produced the following order:

"Source"	SWL dB(A)	Difference with baseline SWL 87.5 dB(A)
Standard cleaner	87.5	
Motor alone	99.5-102	-11.5 to -14
Air-flow; no suction fan	79.8	7.7
Motor imbalance	varies	1.5 to 5.0
Hood	varies	1.0 to 5.8
Wheel trims	varies	0.5 to 1.5
Air-flow; nozzle blocked	89.2	-1.7
Dust bag	89.5	-2.0
Front Bag housing cover	91.5	-4.0
Upright complete	92.0	-4.5

The following order was obtained by near-field sound measurements:

"Source"	Near-field SPL dB(A)
Bellows	105.6
Hood: above orifices	100.5
Wheel centre	98.8
Gap at rear between floor and cleaner chassis	98.5
Agitator bearing end	98.1
Hood: away from orifices	94.8
Rear dust bag cover: bottom	92.8
" : top	87.3

5.7 Noise reduction hypotheses

The following hypotheses were postulated:

1) That by improving the air-flow efficiency, the motor speed can be lowered to maintain the reference performance values. Lower motor speed would result in reduced noise and motor vibration.

2) That any improvement in armature imbalance would result in reduced motor vibration and hence a refined cleaner.

3) That an asymmetric seven blade suction fan would reduce the strength of the 7th order pure tone.

4) That through reduction of motor noise, by any means, cleaner noise would be lowered.

5) That by covering-up the "noise transparent" areas such as the bellows, orifices and gaps the cleaner noise would be lowered.

6) That by reducing the transmission of vibration from the motor to the chassis structure, some reduction can be obtained in the noise radiated from panels.

5.8 Noise reduction methodology

The noise reduction methodology adopted for this study was, in effect "cyclic action research". A concept was proposed, evaluated, possibly modified and tried again. This section presents a summary of the noise reduction work.

5.8.1 Turbopower motor out-of-balance

Hoover plc engineers have considered improvement of the armature balance for a long time. Gill (1979) identified the severe influence of rotor imbalance upon the cleaner noise. This was illustrated by the case of two Turbopower cleaners in section 5.6.1.

5.8.1.1 A detailed comparison of the two cleaners

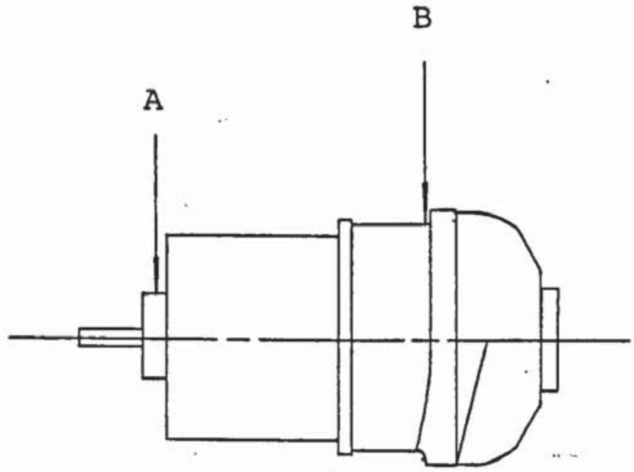
The vibration level of each motor was measured using two types of equipment, namely the vibration meter and armature balancing machine.

Vibration meter results

Vibration levels were measured on the motor body at the bearing housing on the belt-end and a point on the suction-fan-side motor casing immediately above the bearing, a method frequently used in the Floorcare Lab prior to this project. Vibration levels at three motor speeds are tabulated in Table 5.3 for both motors. The suction-fan-end was smoother than the belt-end for both motors. This was unexpected since the residual imbalance of the suction fan was always thought to be greater because it was the bulkier of the two fans. At higher speeds, the U2198 motor shows almost twice the vibration levels of the U2194 motor in both planes. In conclusion, the U2198 motor was indeed worse than the U2194 and the belt-end had a higher vibration level, leading to the situation exposed by the exchange of motors discussed in section 5.6.

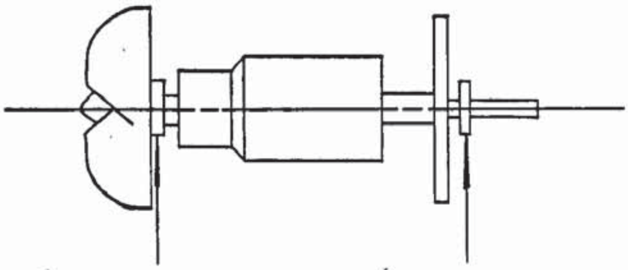
Results using the balancing machine

An accurate measure of the armature imbalance was obtained on the Jackson and Bradwell balancing machine. The calibrated imbalance values for the armature with and without the suction fan are tabulated in Table 5.4. The cooling air fan and the bearing on that end were assembled on the motor shaft at this stage. These results showed that without the suction fan (and the washers) the U2198 motor armature was much better balanced than the U2194 armature (0.128 gm-cm compared to 0.68 gm-cm), while the imbalance at the cooling-air-fan-end was one third of the value for the U2194 armature.



	VIBRATION MAGNITUDE ms^{-2}					
Motor speed (rpm)	4000		5000		8000	
Bearing plane	A	B	A	B	A	B
Motor from cleaner U2194	3.3	1.6	4.0	1.7	8.5	4.3
Motor from cleaner U2198	2.3	1.6	3.3	2.2	13.0	7.0

Table 5.3 Motor casing vibration levels at different motor speeds.



	OUT-OF-BALANCE (gmcm) Suction fan bearing plane	Vent fan bearing plane
Balanced armature	0.012	0.012
Armature from U2194	0.68	0.188
	0.68	0.24
with suction fan dismantle and refit	0.84	0.16
	1.24	0.144
"	1.28	0.184
"	1.12	0.16
average value	1.12	0.16
Armature from U2198	0.128	0.048
	0.128	0.028
with suction fan dismantle and refit	0.94	0.276
	0.76	0.356
"	0.70	0.48
"	0.88	0.64
average value	0.82	0.48

Table 5.4 Out of balance measurements on a balancing machine.

However, when the suction fan was installed, there was an increase in the out-of-balance for both armatures. The U2194 armature was still worse at the suction-fan-end: 1.12 gm-cm compared with 0.82 gm-cm for the U2198. At the belt-end, however, the imbalance values were 0.16 gm-cm and 0.48 gm-cm for the U2194 and the U2198 armatures respectively. The belt end values are well within the design tolerances (presented below) and also quite insignificant compared to the values at suction-fan-end. The belt-end imbalance for the U2198 armature assembly is three times worse than the U2194. This must be crucial, because the U2198 motor resulted in a noisier cleaner. In conclusion, out-of-balance on the belt-end bearing plane is more critical than imbalance at suction-fan-end.

Comparison of the measurement techniques shows that vibration measurements at the motor casing are inaccurate and misleading. This is because the vibration path from bearing to the point of measurement is different, non-linear in both planes and therefore fallible.

5.8.1.2 Design limits for imbalance

During manufacture, the armature is balanced whilst supported in V-blocks at the bearing journals without the bearings. At this stage the armature comprises the commutator, rotor laminations and windings but not the suction and cooling air fans. The armature balance limits are achieved on a Hoffmann automatic balancing machine by a polar milling operation. The production tolerances are,

	Tolerance in gm-cm.
Armature alone, at each plane (without the suction and cooling air fans)	0.34
Suction fan	0.432
Cooling air fan	0.432

After the armature has been balanced, a fine cut is made on the commutator. It was noted that this fine cut was as deep as the initial cut and therefore likely to impair the balance values obtained earlier.

5.8.1.3 Armature balance verses motor and cleaner noise

This test was a natural progression from the results of the above exercise. The objective was to validate or otherwise the finding that imbalance at the belt-end bearing plane was more critical than the suction -fan-end plane in its influence on the overall noise of the Turbopower cleaner.

Although each of the rotating elements have specific imbalance limits as given above, the worse possible case arises when each imbalance force vector lies in the same orientation. Therefore, the total unbalance at each bearing plane will be; 0.34 + 0.432 = 0.772 gm-cm. Four armatures were prepared as follows;

1. a well balanced armature
2. maximum out-of balance on the suction-fan-plane and belt-end-plane was balanced
3. the suction-fan-plane was balanced with maximum out-of-balance on the belt-end-plane
4. maximum out-of-balance on both planes.

Problems were encountered on the use of the Jackson-Bradwell balancing machine. Repeatability of balance values due to suction fan remount was the main difficulty and this is discussed in the next section. The best balance achieved on the machine, with the usual method of assembly, was 0.038 gm-cm. Results of the motor and cleaner sound levels are summarised on the next page.

Armature balance condition	Overall SWL dB(A) Motor only	Cleaner
standard cleaner	99.0	87.5
balanced armature	99.8	86.8
suction-M cooling-0	98.8	88.5
suction-0 cooling-M	100.3	92.0
suction-M cooling-M	98.8	90.4

where; M - maximum imbalance within design tolerance

0 - best balance

Motor noise levels

The ncise level of the motor was measured by hanging it on a light string and measuring the noise at a distance of 1.5 m. The results, given in Figure 5.39a, show the motor noise level as virtually independent of dynamic balance of the armature. The sound levels for the four different armatures were within 1.5 dB. The increase in vibration, associated with the armature imbalance, did not appear as an increase in noise level, perhaps because the noise of the motor was dominated by the air turbulence noise (the air intake and exhaust were not muffled). There was greater increase in the noise level due to imbalance on the cooling-fan-end than suction-fan-end.

Cleaner noise levels

The cleaner overall noise levels, shown in Figure 5.39b, show that when the motor was installed in a cleaner there was a reduction of 10-12 dB(A) in the overall noise level. There was 10 dB(A) attenuation at frequencies up to 2 kHz and about 20 dB(A) above 2 kHz. This reduction in noise is partly due to the sound transmission loss due the hood and partly because the air inlet and exhaust were now muffled.

Using the cleaner noise for a balanced armature as reference,

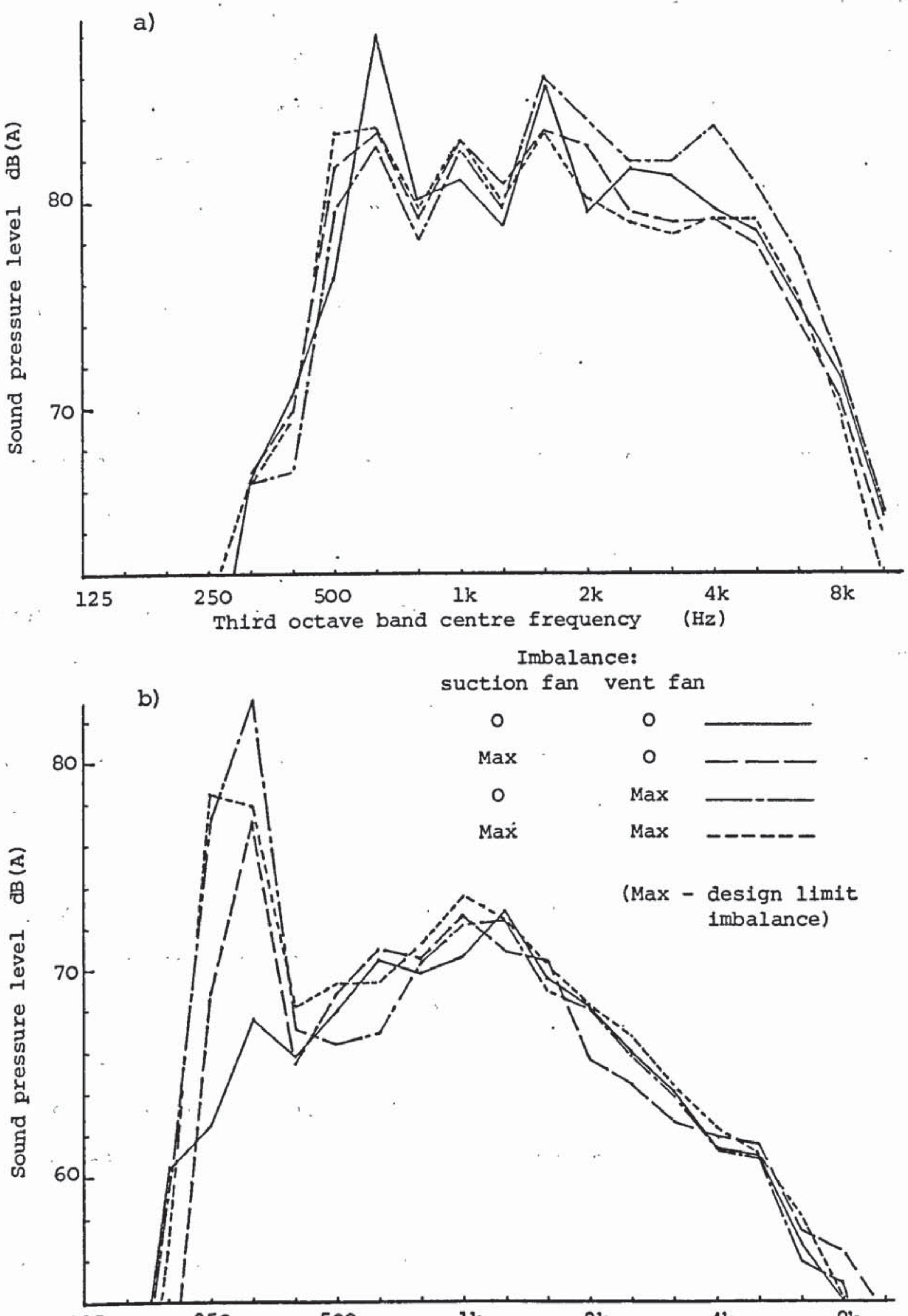


Figure 5.39 a) Motor noise and b) cleaner noise vs. armature imbalance.

notice an increase of 5.2 dB(A) due to the imbalance at the cooling-fan plane compared with 1.7 dB(A) for the imbalance only at the suction-fan plane. These results confirm quite clearly the conclusion drawn above: that the cooling-fan bearing plane is more critical than the suction-fan plane. Under current production requirements, the cleaners leaving the assembly line are likely to show a 5-6 dB(A) spread in their noise levels. Indeed, this is the case at present.

The above tests were conducted on a standard cleaner; its sound power level was 87.5 dB(A). The sound level recorded when using the best balanced armature was 86.8 dB(A). This suggests that an improvement in the armature out-of-balance by a factor 20:1 results in a benefit of less than 1 dB(A). Therefore, the most effective improvement does not lie in better absolute values of the armature imbalance but in better quality control to ensure that individual components on the armature meet the drawing specfication, and that the motor assembly is correctly balanced. The results of the above exercise highlighted the whole topic of the motor imbalance at Hoover on both sides of the Atlantic.

5.8.1.4 Factors inluencing armature imbalance

Influence of coarse tolerance in suction fan

As mentioned earlier, there were problems associated with the lack of repeatability in balance, especially for the suction fan remount. Therefore, an improved mounting for the suction fan is required. The fan is moulded with a threaded blind hole and screwed onto the end of the armature shaft with two washers in between. Observations show that the suction fan may wobble due to the excess play in the screw threads. Therefore, the design in Figure 5.40 was proposed. However, it was not possible to manufacture a sample of the modified fan/armature assembly for evaluation because of the limitations of manpower and other commitments in the model shop.

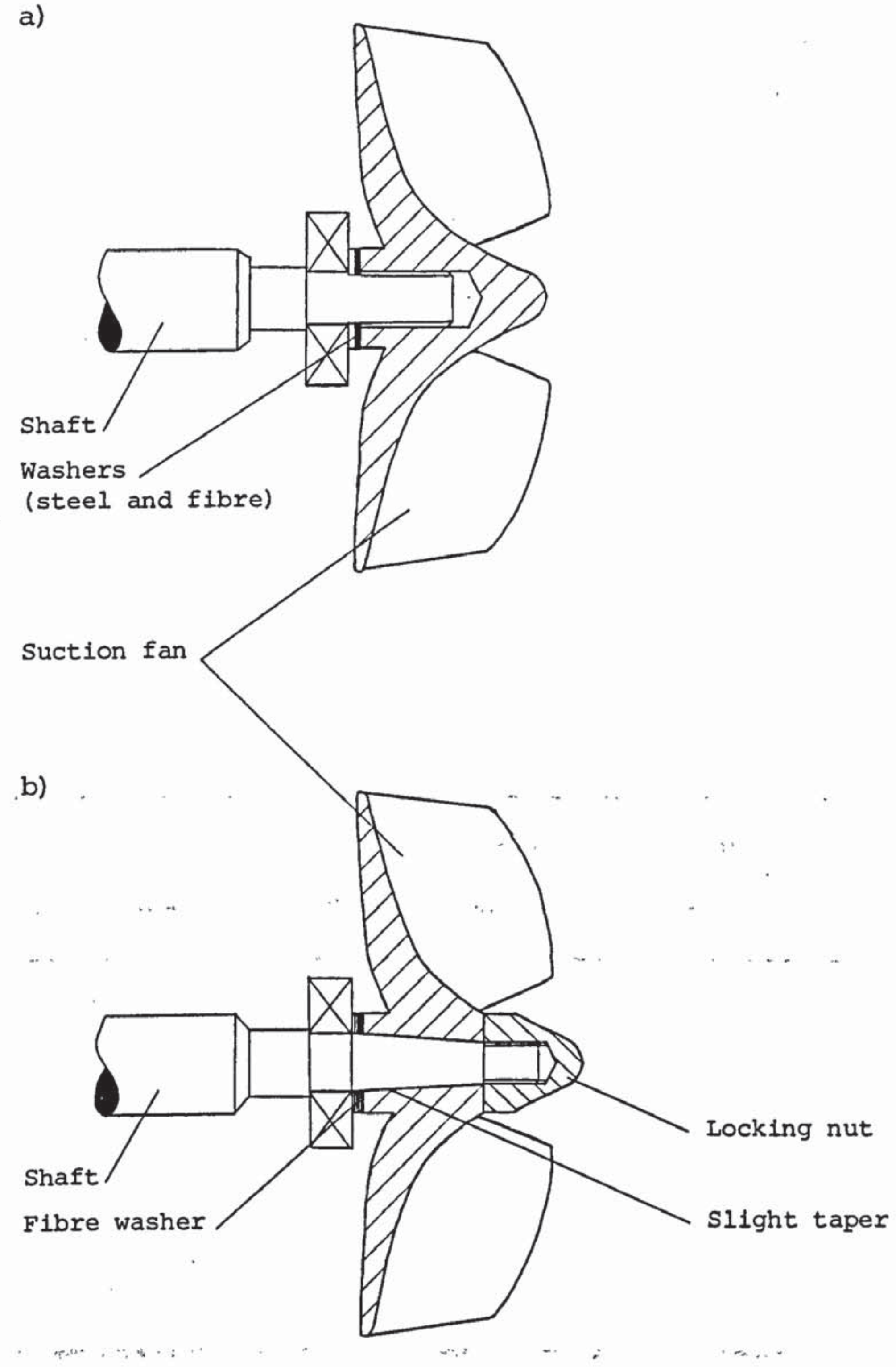


Figure 5.40 Suction fan mounting a) existing
b) proposed.

Variation in the imbalance of suction and cooling air fans

These two fans contrast completely; the cooling fan was thin, light and symmetrical, while the suction fan was much larger and asymmetrical. It was therefore surprising that both fans had the same limits on balance. Production line balance values were obtained for both fans. These values are tabulated in Table 5.5. As expected, cooling fan balance was well within the limit whilst balance of the suction fan was occasionally outside the limit.

Noise tests were continued after first acquiring confidence in the consistency of fan remounting by tightening the fan further than necessary and aligning the fan with the armature.

	Out-of-balance	(gm-cm)
Fan sample	Suction fan	Vent fan
1	0.26	0.10
2	0.40	0.225
3	0.34	0.15
4	0.12	0.10
5	0.64	0.025
6	0.38	0.075
7	0.30	0.225
8	0.45	0.20
9	0.38	0.20
10	0.51	0.175

Table 5.5 Out-of-balance values for a sample of suction and vent fans.

5.8.2 Motor vibration related harmonics

Many of the peaks in the narrow-band noise spectra are associated with vibration properties of the rotor. These can be diagnosed as follows:

1. the first order, mainly out-of-balance
2. the second order, due to;
   - bent rotor shaft. The force of the belt bends the armature shaft
   - eccentric or misaligned shaft at bearings. The play in the bearing races and the housings was measured and found to be 0.4 mm. Misalignment often leads to bearing failure, and to a rise in vibration harmonics.

   The noise contribution of sources which give rise to the second order and higher harmonics, was difficult to isolate. As the magnitude of these orders is greatly accentuated by the imbalance of the rotor, the greatest improvement was achieved by eliminating this imbalance.
3. harmonics resulting from bearing looseness and inaccuracies
4. electrically induced vibrations;
   - 100 Hz spike, and its modulation of the major vibration peaks.

Investigation of the 100 Hz spike (Figures 5.41 and 5.42)

The 100 Hz "spike" was first noticed on the Compact Super suction cleaner. An RPM spectral map of vibration of the cleaner body, in the frequency range of 0-2 kHz, is shown in Figure 5.41 and shows clearly the excitation at 100 Hz. It is independent of the motor speed and of a magnitude proportional to the supply voltage. A second harmonic is also present, albeit very low in magnitude.

It was quite apparent that this 100 Hz spike was an electrical excitation because it disappeared upon disconnection of the mains supply. Moreover, this was checked by an alternative electrical supply, at a

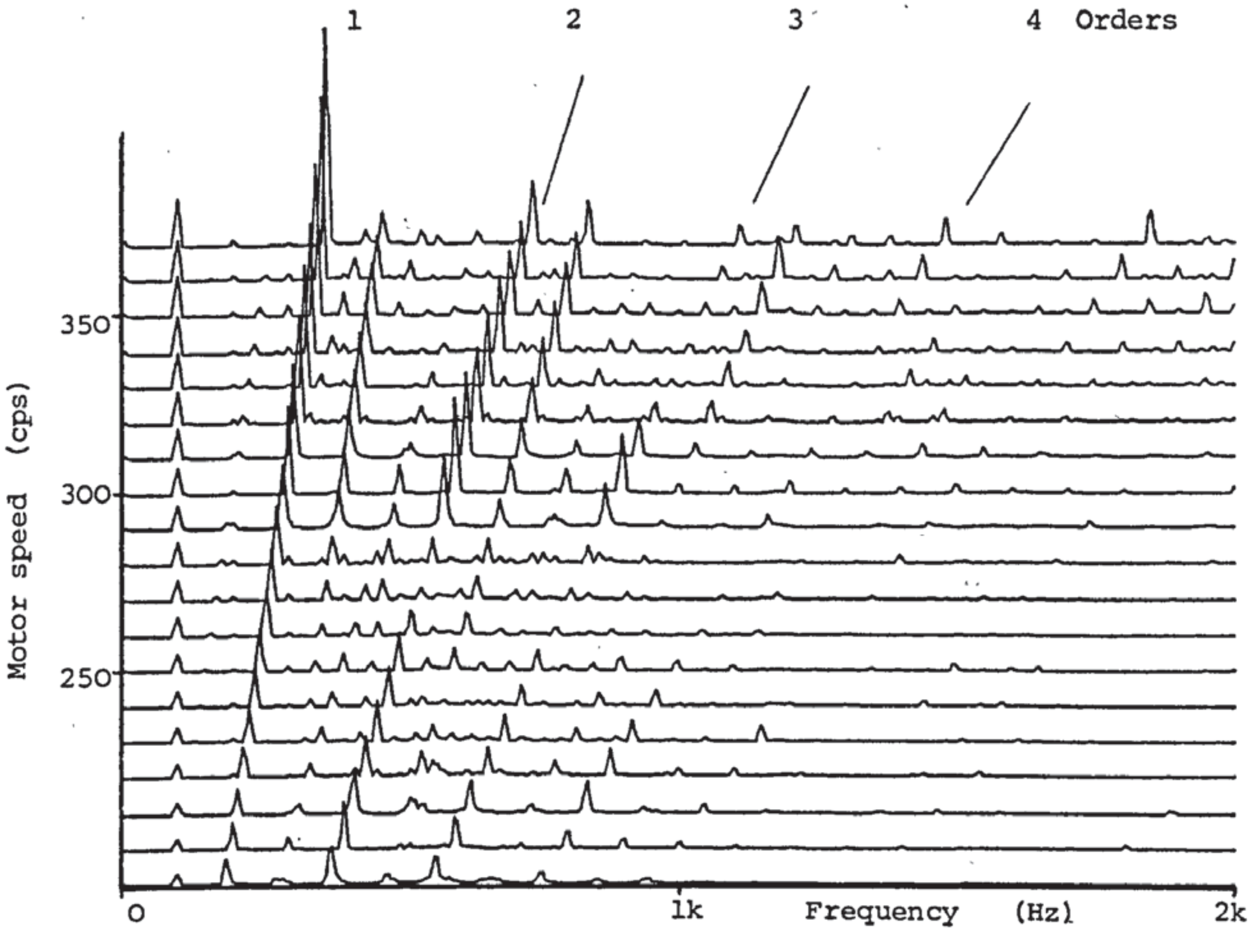


Figure 5.41 Vibration spectral map for Compact Super cleaner showing the 100 Hz spike and its modulation of other peaks.

frequency of 67 Hz from a generator. As a result, the "spike" shifted to 135 Hz. The results are tabulated in Table 5.6. This electrical phenomenon affected both the noise and vibration responses. For the Compact Super cleaner, see Figure 5.42a, and an Electrolux cleaner the spike was about the same magnitude as the out-of-balance peak. For the Turbopower, the imbalance peak was worse as shown earlier in Figure 5.31b. This meant that for a motor with improved rotor balance, (the motor of the Compact cleaner is post-assembly balanced), the vibration due to the 100 Hz spike is significant. Although the 100 Hz peak shows up on the noise spectrum it was attenuated by the A-weighting, and is small compared with noise from other sources.

The 100 Hz spike modulates high frequency peaks to produce

Electric supply to motor.	Vibration peaks present. Frequency (Hz)	Magnitude (dB)
Household mains 240 v 50 Hz	100 200 300 377.5 *	113.6 104 101.4 100.5
Alternative supply 240 v 67 Hz	100 135 363.7 *	v small 115.7 102.7
DC supply 240 v 0 Hz	100 136 367 *	82.7 87.6 105.2

(* - Motor speed)

Table 5.6 Motor vibration characteristics due to electrical supply.

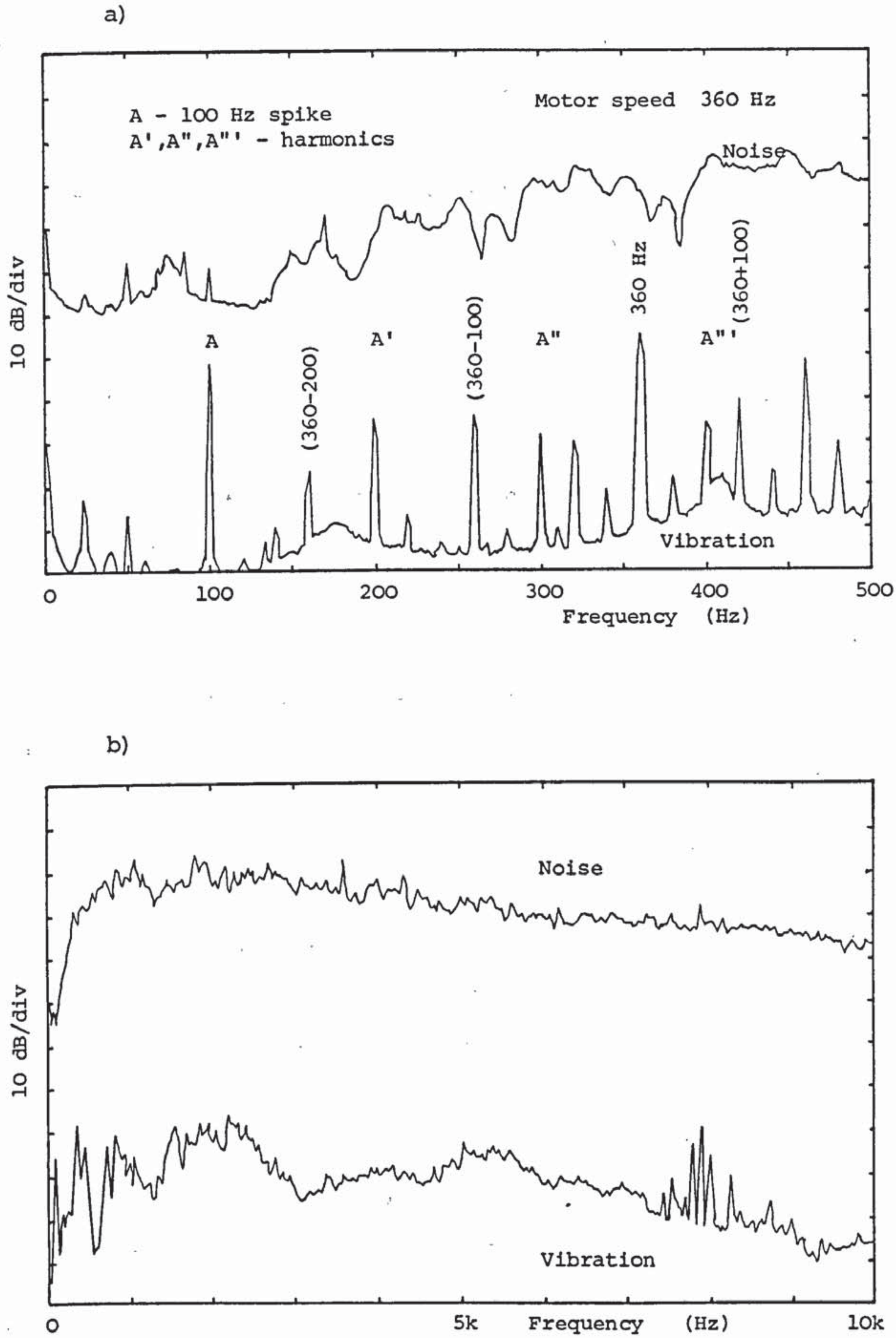


Figure 5.42 Noise and vibration spectra for Compact Super cleaner motor a) o - 500 Hz
b) o - 10k Hz.

sidebands. These sidebands are shown clearly in the results. The effect upon the noise spectrum was the appearance of a cluster of peaks instead of a single peak as shown in Figure 5.42b. In the case of the Compact cleaner, the cluster led to noticeable "beats". This source was not considered to be too serious in the case of the Turbopower cleaner. This situation may have to be reviewed when the armature balance has been improved.

5.8.3 Bearing characteristics (Figure 5.43)

Armature bearings

Ball bearings are used on the armature and the agitator. These bearings are bought from external suppliers such as SKF, NTN and Koyo. The bearings comprise a train of rolling elements, details of which are tabulated below together with the predicted vibration frequencies. To improve vibration reduction, quiet bearings and a rubber cup housing were evaluated.

	Bearing: Armature	Agitator
Elements	7	11
ball diameter (mm)	3.99	3.18
inside race dia (mm)	11.23	8.66
outside race dia (mm)	20.60	15.16
bearing I/D (mm)	7.98	6.30
bearing O/D (mm)	22.00	17.42
Main frequencies: in Hz.		
Fundamental	285	58
train freq.	100.5	21.1
rolling element	519.3	100.7
inner raceway indent	1291.0	406.1
outer raceway indent	703.7	232.0

Quiet bearings

Samples of high grade quiet bearings were obtained from the above manufacturers and tests were conducted of the motor and cleaner noise. The dimensional details of these quiet bearings were similar to the existing bearing and hence there will be no difference in the predicted frequency response, however the magnitude of the components in the noise and vibration spectra, related to "bearing frequencies", was expected to be lower. To emphasise the bearing noise, the armature was well balanced and the suction fan was removed (to eliminate the dominant air noise).

The only viable method of checking vibration of an assembled motor is by using a vibration meter. Although this method was unsuitable for comparing vibration on one bearing plane to the other, it was still good for comparison of motors. The vibration levels for different types of quiet bearings are given in Table 5.7. The Koyo 608 13ZZ gave the lowest

Bearing	Motor casing Vibration level (g)		Cleaner sound power level (dB(A))
	Suction fan plane	Vent fan plane	
Standard NTN 608 (new)	0.24	0.112	84.1
NTN 608 ZZ/5K	0.27	0.14	86.0
NTN 608 ZZ/5C	0.21	0.11	85.4
NMB 608 ZZ	0.31	0.081	85.2
SKF 608 ZZ/LHT23	0.21	0.055	84.9
KOYO 608 - 11ZL	0.125	0.073	85.5
KOYO 608 - 13ZZ	0.12	0.057	82.5

Table 5.7 Motor casing vibration and sound levels for "quiet" bearings.

vibration level and the lowest motor noise level. The results show that the reduction in noise became insufficient when the suction fan was installed: the motor SWL went up to its normal value of 101.3 dB(A).

Rubber cup as an isolator

A thin rubber cup was made to house the suction-fan-end bearing. The bore in the motor casing was enlarged to accept the rubber cup. The motor noise in third octaves are presented in Figure 5.43. There was less than 1 dB(A) reduction in the noise level of the motor.

5.8.3.2 Agitator bearings

Two ball bearings are used on the agitator ends. The races are made in-house but the rolling elements are purchased in bulk from a supplier. The following observations were made:

- the quality and consistency of size of these balls was poor
- surface finish of each of the races was also poor and rough
- the agitator seemed very harsh and rough when rotated
- the thrust on the ball bearings depends on the length, between shoulders, of the central bar of the agitator assembly and the tightness of the end caps. The thrust was inconsistent and this resulted in some agitators being very tight and rough and others being slack.

The rotational speed of the agitator is around 3400 rpm. The vibration generated by the bearings is, therefore, of low frequency and considered to contribute little to the cleaner noise level, because low frequency is highly attenuated by the A-weighting. Nevertheless, a pure tone can result from a defective bearing. In contrast to the Turbopower agitator, the Electrolux Z610 agitator assembly was noticeably smoother and quieter. It comprised phosphor bronze bearings and a vastly superior method of thrust control. Consequently, except for a complete re-design and improvement in quality, nothing can be done.

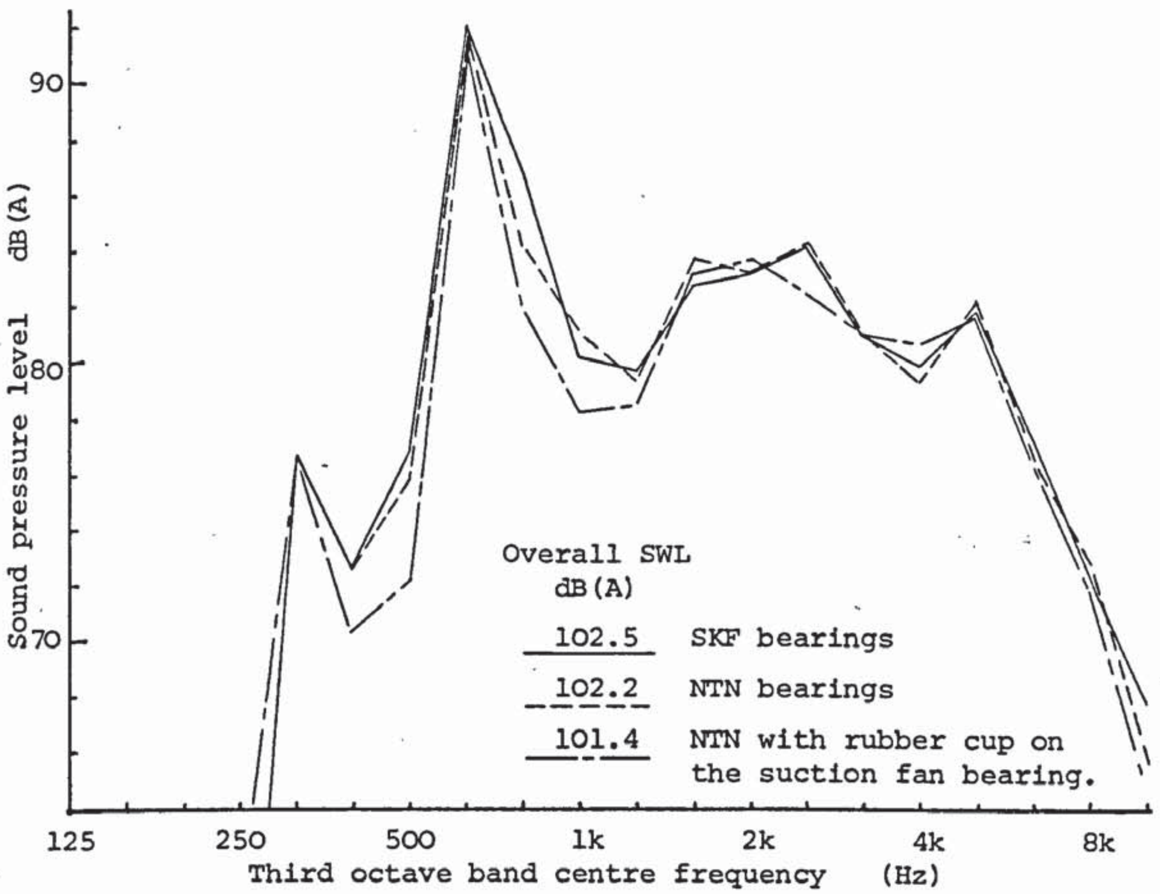


Figure 5.43 Turbopower motor noise vs. bearing type and bearing isolation.

5.8.4 Armature rotor slots (Figures 5.44 and 5.45)

There are 12 rotor slots. This results in a pure tone at 12 x 285 (motor speed) = 3420 Hz. Similarly, the 24 elements on the commutator ring gave rise to a peak at 24 x 285 = 6840 Hz. These peaks can be eliminated or reduced by increasing the rotor-stator gap, skewing the rotor slots or by a smooth armature.

Rotor-stator gap

The design specification for the radial gap between the rotor and the stator is 0.4mm. Naturally, this gap is achieved only when the armature is positioned centrally. If there is any misalignment or eccentricity of the armature, then the gap between one field winding is going to be less than the other.

Discussions with the motor laboratory staff revealed that the radial gap had been optimised according to the following considerations: an increase in radial gap resulted in a reduction in the power output of the motor and an increase in the temperature due to the resistance to the magnetic flux. A close radial gap increases the pure tone due to the rotor/stator interaction and also increases the likelihood of rotor/stator interference in the light of manufacturing tolerances.

Noise tests were conducted on the motor with the field windings offset radially. This method of adjusting the gap was chosen because the alternative was to machine either an armature or a field for each gap setting. This was not feasible in the model shop. A minimum gap of 0.1mm and a maximum of 0.5mm were evaluated. The results, in Figure 5.44, showed that the magnitude of the 12th order was dependent on the radial gap yet the overall noise level of the motor was not.

Rotor slot skew

It was not possible to evaluate this modification because of the manufacturing difficulties of preparing one or two samples. However,

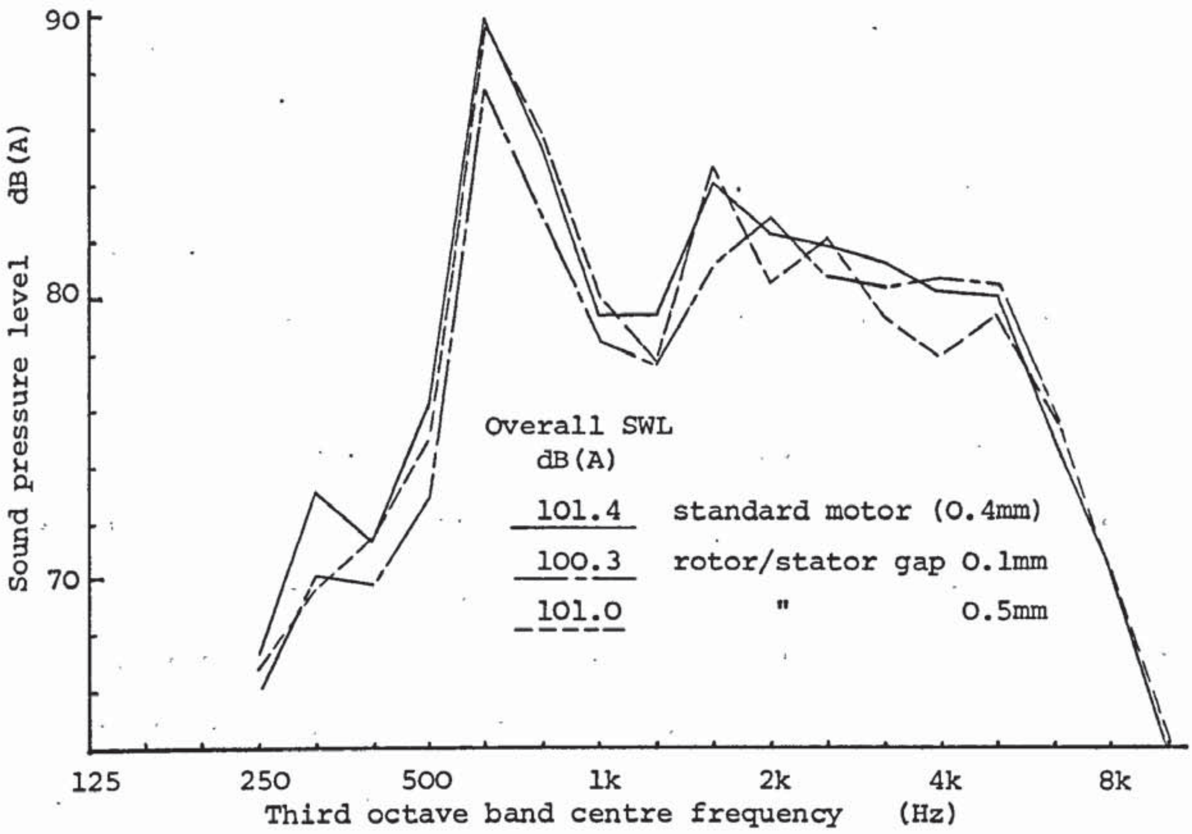


Figure 5.44 Influence of rotor/stator radial gap on noise spectra of a Turbopower motor.

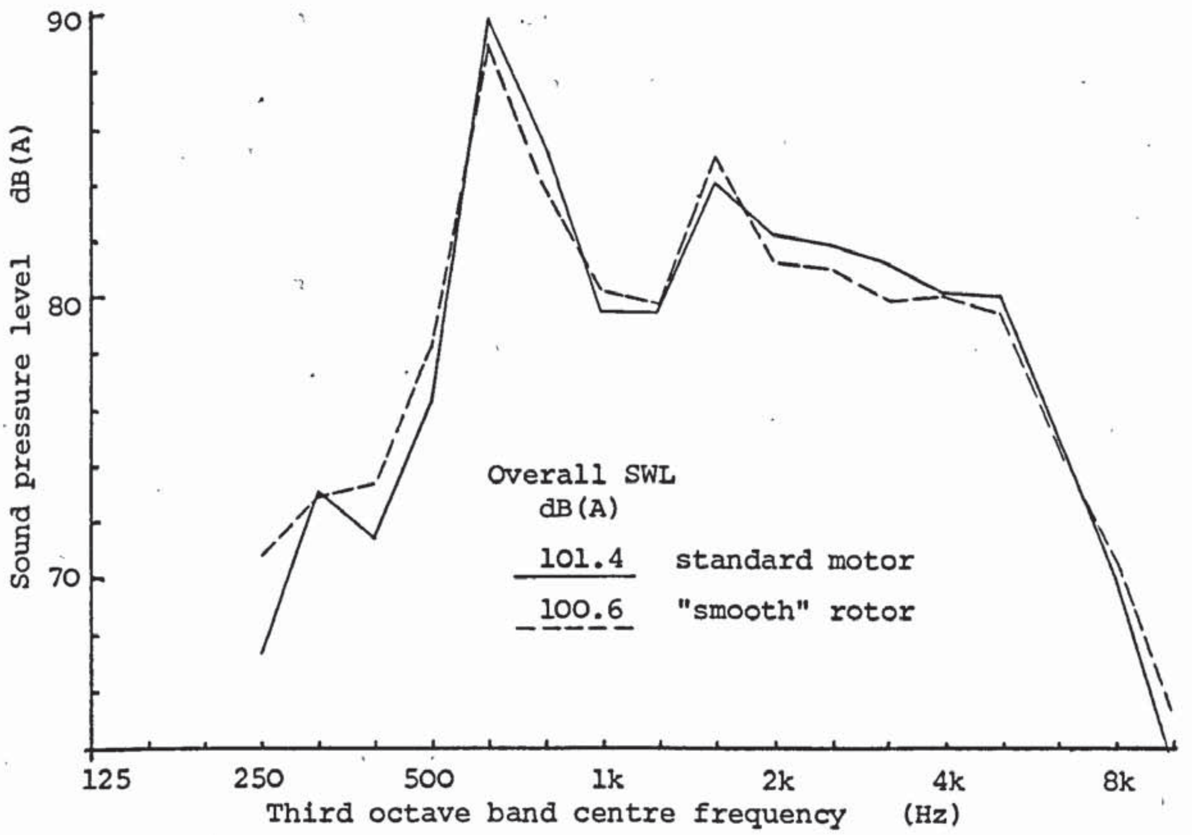


Figure 5.45 Influence of a smooth armature on noise spectra of a Turbopower motor.

discussions with staff of the motor laboratory highlighted the following;

- slct skew enhances flux waveform which means smoother running,
- it reduces noise,
- there is no loss in the electrical power output of the motor,
- the cost of manufacture is similar to conventional motors,

but, - there would be problems in manufacturing, eg epoxy coatings.

Smooth rotor

The rotor slots were filled-in using an epoxy resin. The armature was then machined smooth and dynamically balanced. As a consequence of filling the rotor slots, the flow of the cooling air was greatly reduced and thus the temperature of the armature stack increased. The temperature rise was appreciable, and after some time, grease was expelled from the bearing, which then caused a pronounced "whistle".

A smooth rotor was obtained in another way. A heat-shrinkable tape was wound exactly once around the armature stack and the motor speed was increased gradually to allow the tape to tighten around the rotor. The results, in Figure 5.45, showed no reduction in the motor noise level.

A similar exercise was carried out on a washing machine motor. The result in this case was a reduction of 12 dB(A) in the corresponding pure tone and a almost 5 dB(A) in the overall motor noise. This significant noise reduction was probably due to the larger size of the washing machine motor and that it was made of a zinc alloy compared with plastic for the Turbopower motor. Furthermore, in the case of a Turbopower motor, the motor noise is dominated by the air turbulence noise, which explains why there was no reduction in the overall noise of the motor.

5.8.5 Tests on the fan chamber (Figures 5.46-5.49)

Near-field sound measurements

The motor was suspended by two thin wires from the ceiling of the acoustic room. A stand was erected to restrict the gyratory motion of the motor during operation. The effect of a windshield on the microphone to eliminate the dominant air turbulence noise, is shown in Figure 5.46. Notice the enormity of air turbulence noise and the appearance of the underlying features after the turbulence noise was "filtered". The noise spectra on four sides of the motor are given in Figure 5.47. There is a peak at 630 Hz for measurement on the fan chamber side. Further evidence was shown by Figure 5.20 which presented the noise spectra for the motor assembly with and without the fan chamber. Consequently, the 630 Hz peak belongs to the fan chamber.

Vibration test

The results of the impact test on the fan chamber using the instrumented hammer is given by Figure 5.48. There are resonances at 440 Hz, 938 Hz and 2195 Hz. However, as mentioned earlier, the response of a lightweight component would alter radically when installed in a system. Consequently, this data cannot be used at face value.

Addition of damping materials.

The traditional way of treating panel vibration is by using a heavy damping material. The noise measurements for the fan chamber modified in this way are presented in Figure 5.49. The amount of damping material that could be stuck on was limited to areas having 3-4mm clearance in situ. This limited the material application to about 50 sqcm. A fan chamber made of a different material was proposed as a second modification, but it was not possible to evaluate this because of problems of manufacture.

5.8.6 Vibration response of the armature shaft (Figure 5.50)

The presence of laminations, the commutator and the vent fan makes prediction of resonant frequencies difficult. Subsequently, vibration response of the armature was studied. Figure 5.50a shows that the response of the armature changed when the suction fan was installed. The armature was then tested in the motor assembly and Figure 5.50b shows the influence

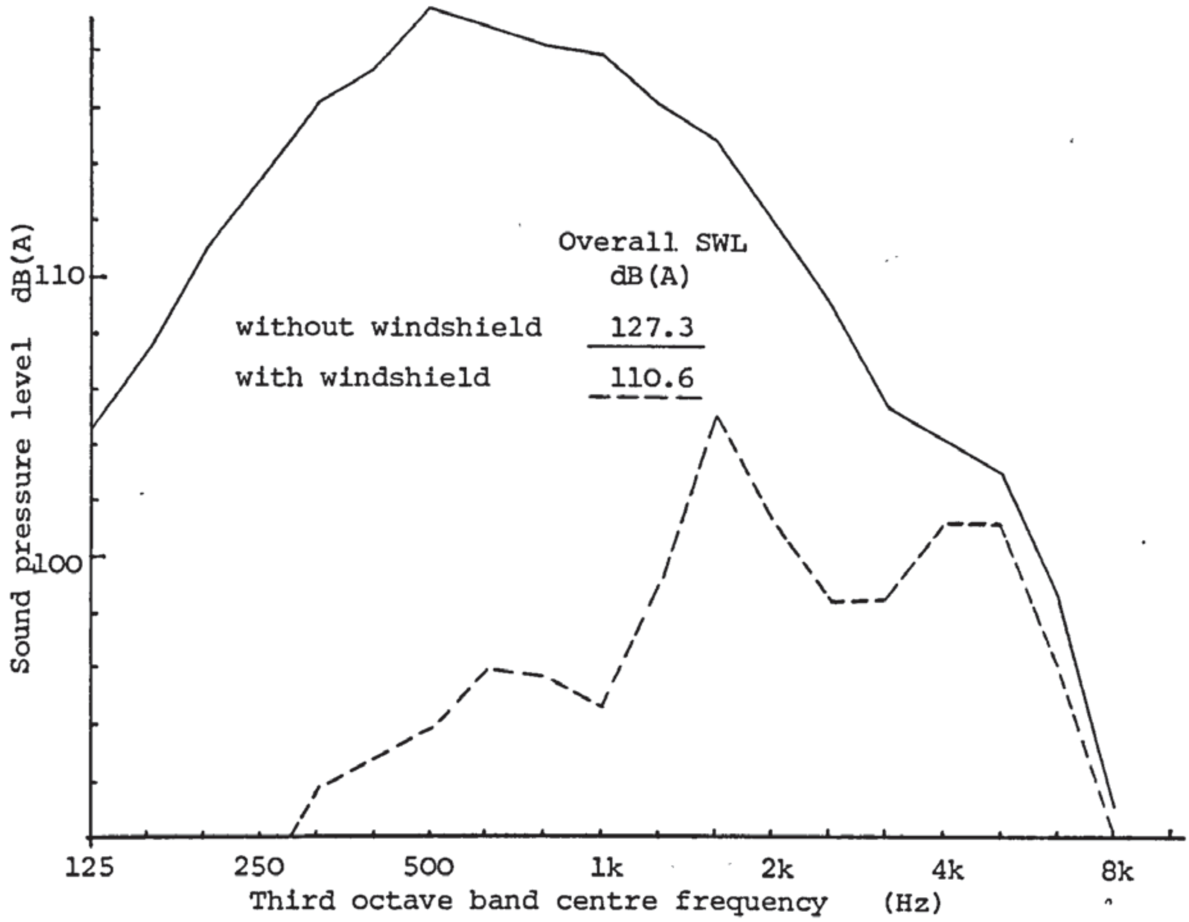


Figure 5.46 Near-field sound measurement of motor with and without a windshield.

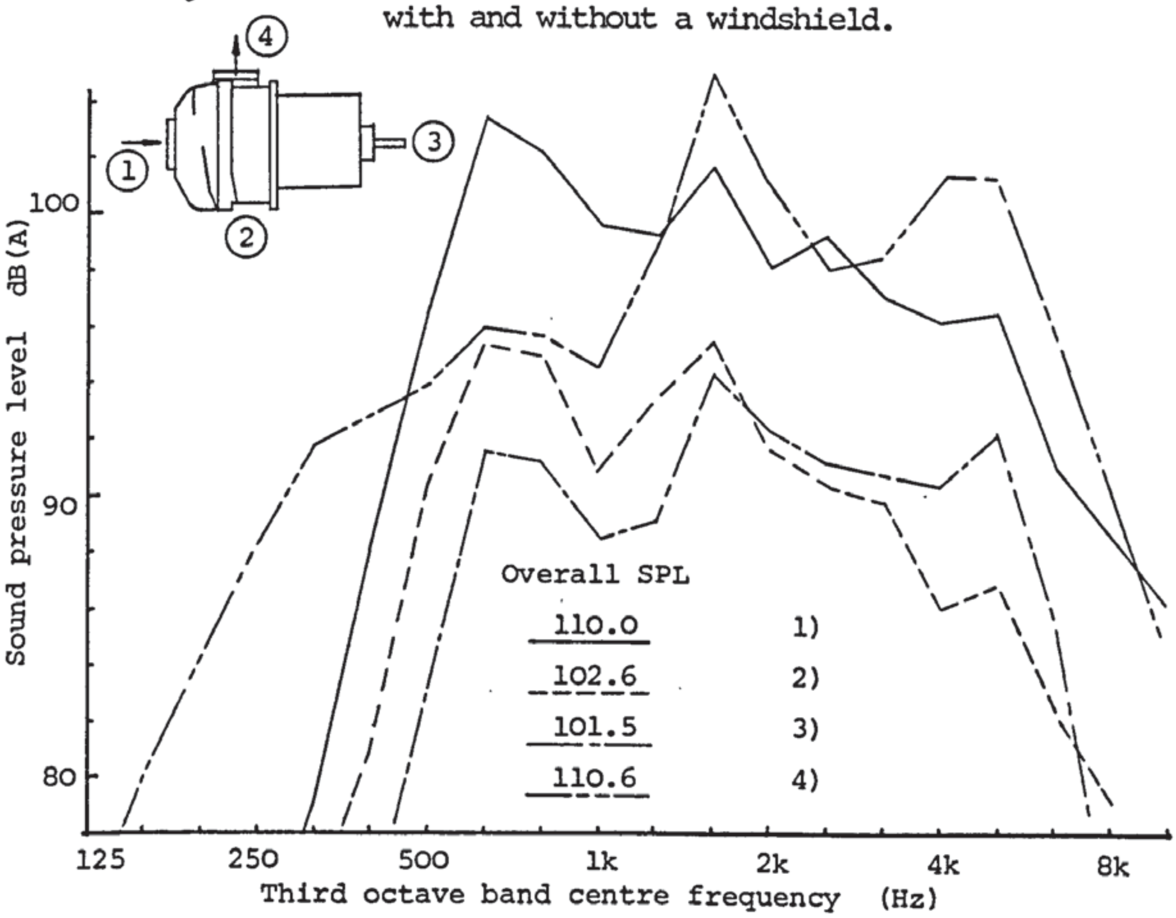


Figure 5.47 Near-field sound measurements around motor.

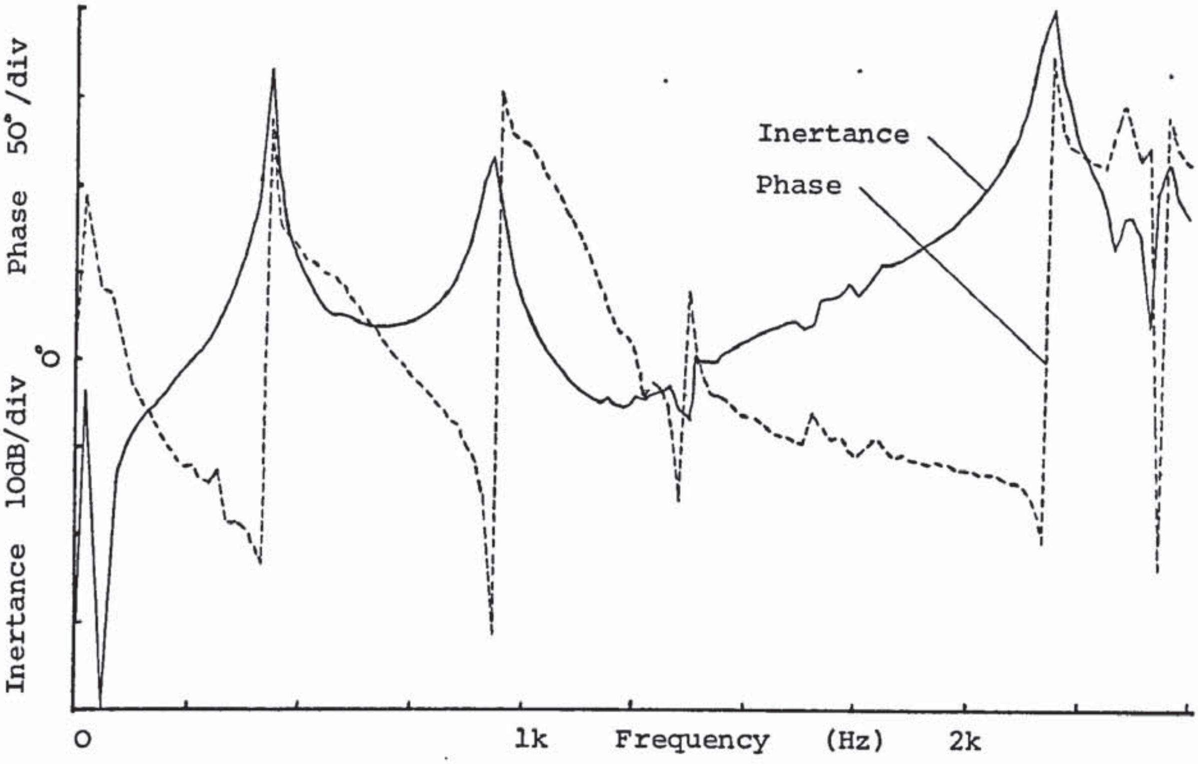


Figure 5.48 Hammer test on a fan chamber.

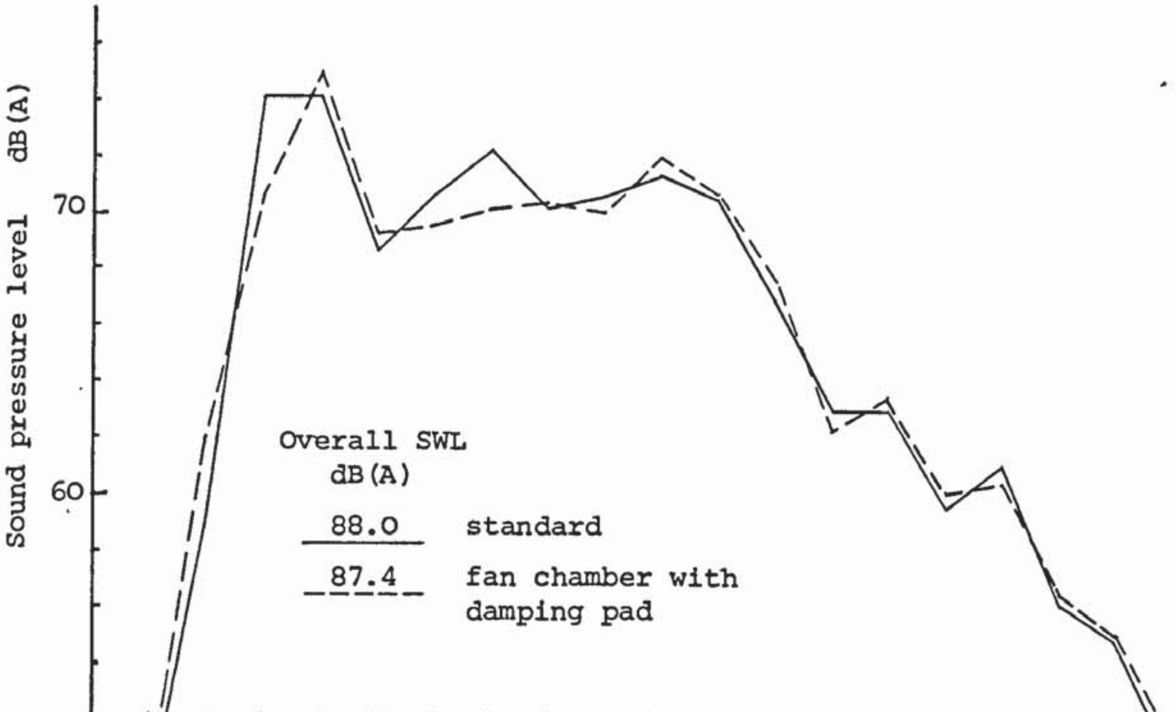
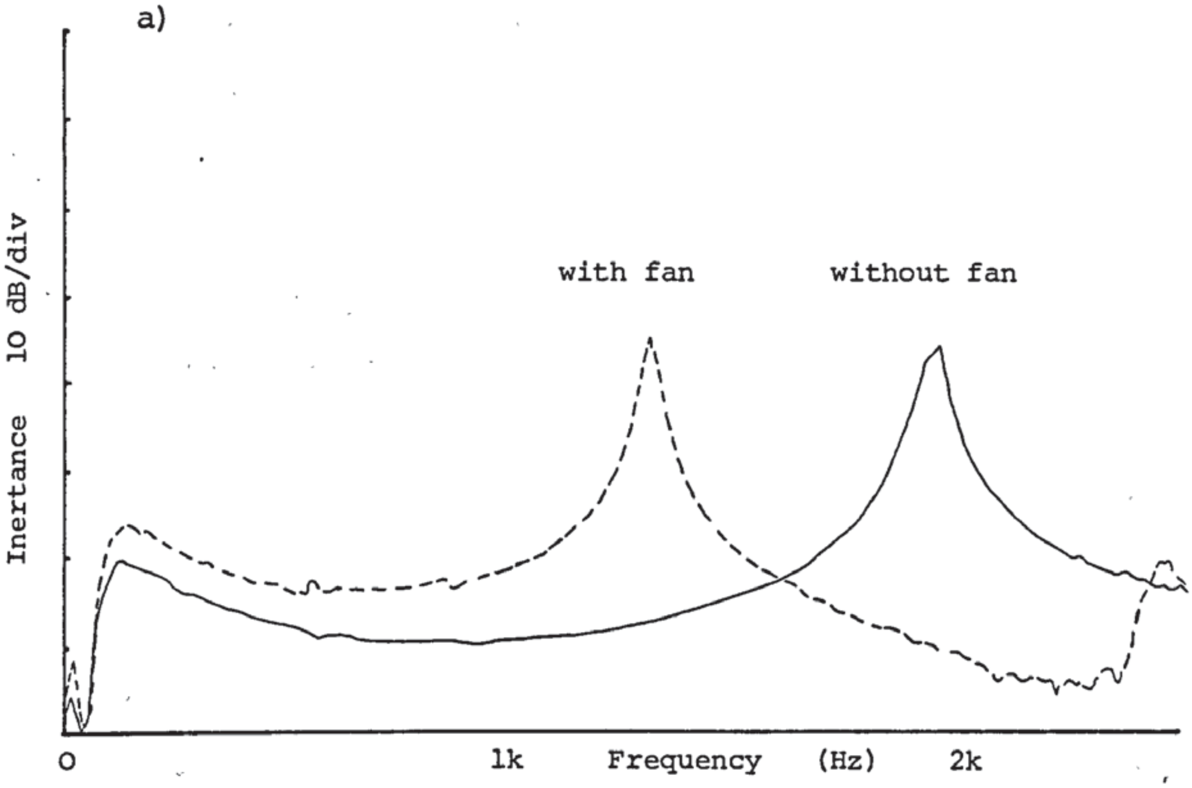


Figure 5.49 Influence on cleaner noise by using damping material on the fan chamber.



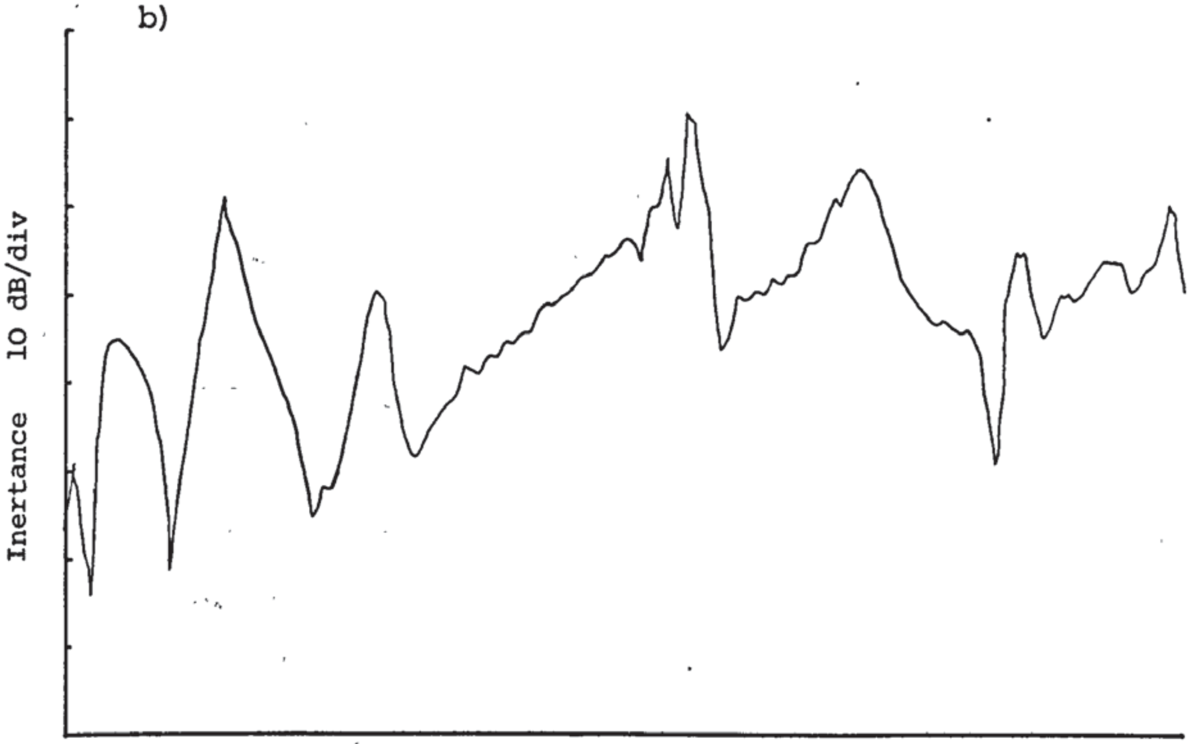


Figure 5.50 Impulse response of the armature shaft
a) with and without suction fan
b) installed in a motor.

of the other components such as bearings and the motor body. A further change would result when the belt was installed. Therefore, it was difficult to determine the contribution of a particular component to the overall motor noise.

5.8.7 Air-flow efficiency (Figures 5.51-5.55)

The aim was to enhance the air-flow efficiency of the cleaner. It was hypothesised that for any increase in efficiency the running speed of the motor could be reduced to maintain the reference performance of the cleaner. This meant measurement of,

- baseline performance of cleaner,
- air-flow efficiency vs motor speed,
- air-flow efficiency vs suction fan blade to fan chamber clearance,
- influence of the sharp 90 degree bend,
- efficiency loss due to air leakage in the inlet ducts,
- alternative blade profiles.

5.8.7.1 Performance level

Figure 5.4 presented the performance of a standard cleaner, and these values were used as reference. The maximum volumetric air-flow rate for unrestricted flow of air into the cleaner when the suction level is at its minimum is 37.5E-3 m^3/sec. In the extreme case of a blocked nozzle the air-flow is zero (or a minimum) and the suction is at its greatest. In practice, the carpet hinders the air-flow and therefore there is a compromise. A new term called the suction power is used, defined as:

$$\text{Suction power} = \text{suction} \times \text{air-flow}$$

For consistent units of watts;

$$\text{Suction power} = \text{suction} \times \text{air-flow} \times 0.00981$$

where; suction - mm H_2O

air-flow - dm^3/sec = $10^{-3} m^3$/sec.

and, Efficiency = suction power/electrical power input

5.8.7.2 Air-flow efficiency vs motor speed

The maximum air-flow efficiency of a standard cleaner, relative to the electrical power input, is approximately 14% and that of the motor on its own is around 23.0%. The efficiency for motor speeds of 265 Hz, 275 Hz, 285 Hz and 295 Hz are plotted against the motor speed and given in Figure 5.51. Notice a drop in the efficiency to 22.8% (0.4% reduction from baseline value) for a reduction in the motor speed from 285 Hz to 275 Hz (3.5%). There was a reduction of almost 2 dB(A) in the cleaner noise level by reducing the motor speed, but there was no reduction in the air-flow. The results are summarised below:

Motor speed (Hz)	Air-flow rate $10^{-3} m^3/sec$	Motor efficiency	Cleaner SWL dB(A)
295	16.0	23.2 %	89.8
285	15.5	22.9	87.3
275	15.5	22.8	85.4
265	14.7	22.1	85.3

5.8.7.3 Air-flow efficiency vs blade tip clearance

Currently a gap of 4.0mm is specified between the fan chamber and the blade tips. Neise (1976) found that the gap was detrimental to the air-flow efficiency of the suction fan, but is necessary to allow passage for bulky dust particles and debris. The aim of this test was to determine the effect of the gap on efficiency and then to optimise the gap if necessary. The suction fan has a screw thread of 1.25 mm pitch and hence, shims of that thickness were prepared. This maintained the alignment of the fan with respect to the armature and therefore the imbalance was consistent for all tests. The results of air-flow and efficiency for motor

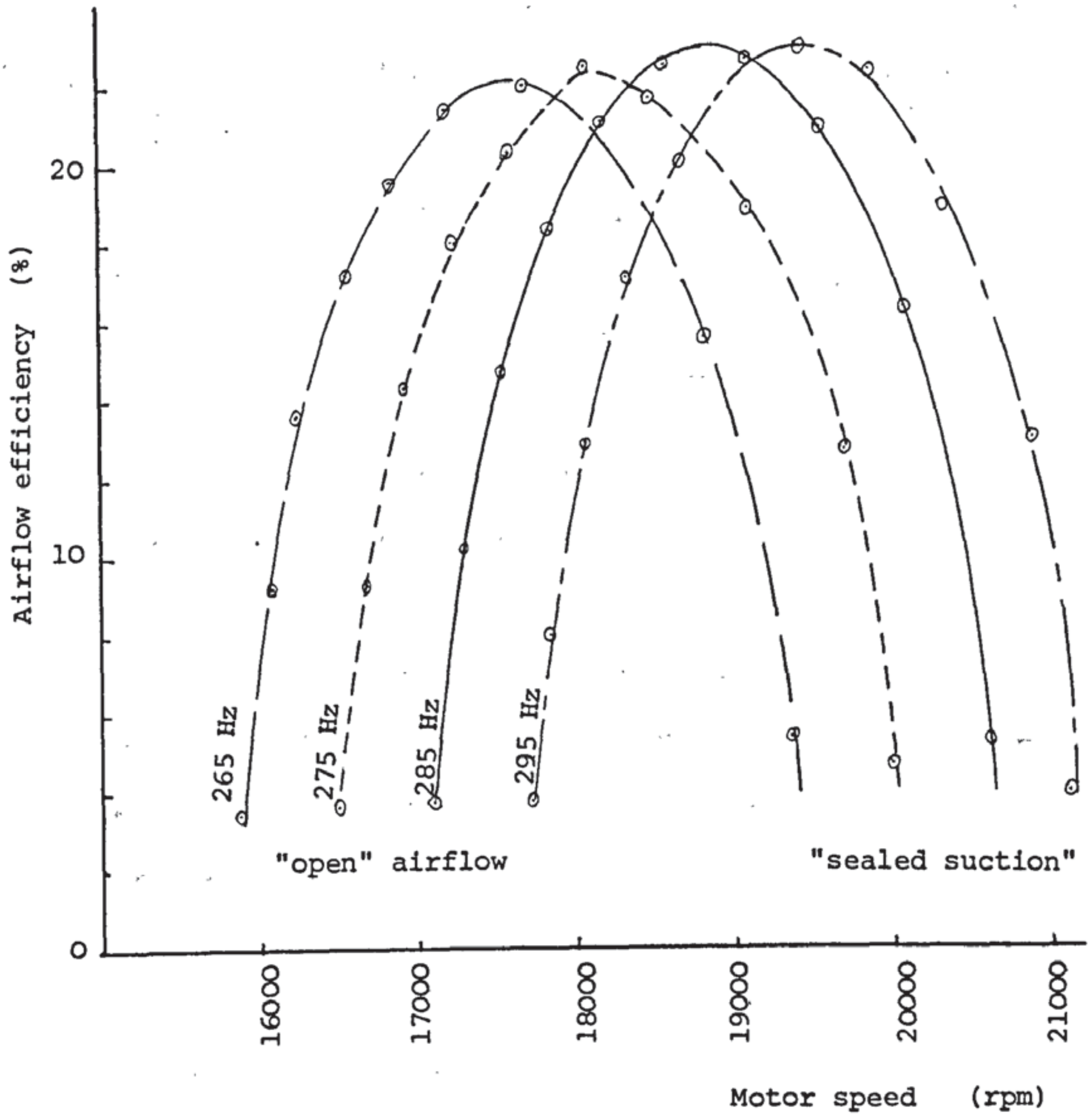


Figure 5.51 Variation in the airflow efficiency with airflow restriction.

only are tabulated below. The efficiency was found to be dependent on the gap, but the change was less than 1%. In fact, the standard gap of 4.0 mm gave the highest efficiency, and next to the lowest cleaner noise.

Blade-to-fan chamber gap	Air-flow dm^3/sec	Efficiency %	Cleaner SWL dB(A)	Motor SWL dB(A)
3.75 mm	15.5	22.9	87.9	102.0
2.50	15.0	22.8	87.8	99.7
1.25	15.5	22.0	88.5	100.4
minimal	16.0	22.3	88.0	97.0

Figure 5.52a presents the sound levels for the motor only. Some reduction in noise was obtained by closing the fan-chamber to blade-tip clearance. The overall cleaner noise, in Figure 5.52b, illustrated a definite relationship in the frequency range 400-2000 Hz but the overall sound level of the cleaner was almost constant at 88 dB(A). The variation in the cleaner sound levels with the motor speed is plotted in Figure 5.53. There was little consistency in the results and no valid conclusion could be drawn.

5.8.7.4 Influence of a sharp bend

There is a 90 degree turn at the inlet to the suction fan and this gave rise to a suspicion that the bend must result in an uneven velocity profile.

Determination of velocity profile

The view looking into the air inlet to the fan chamber is given in Figure 5.54 and shows the line on which the velocity profile was thought to be most pronounced. Subsequently, a rig was designed and made which allowed a hot-wire anemometer to traverse the line of interest. Much care was necessary to handle this sensitive probe and therefore the rig and

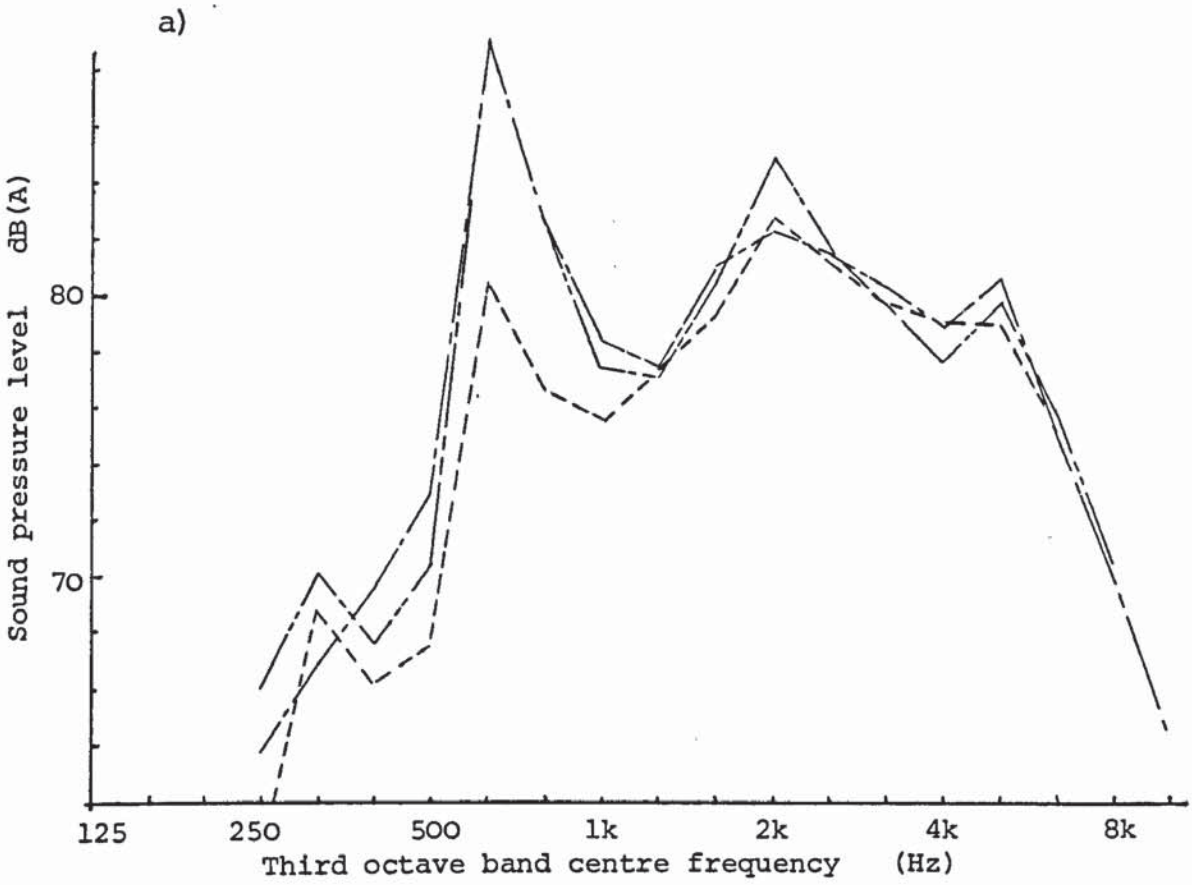


Figure 5.52 Influence of fan blade tip/chamber clearance on a) motor noise and b) cleaner noise.

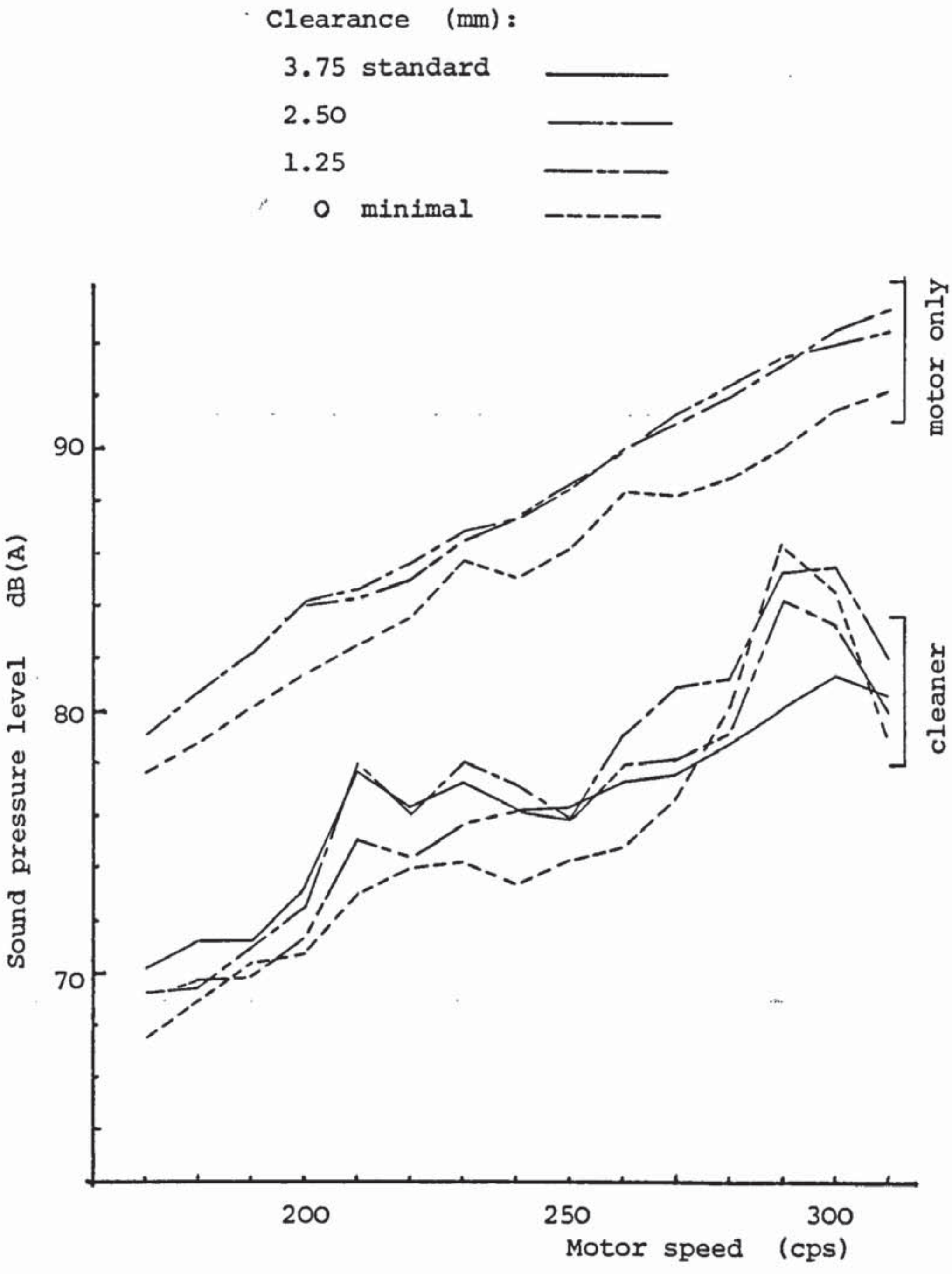
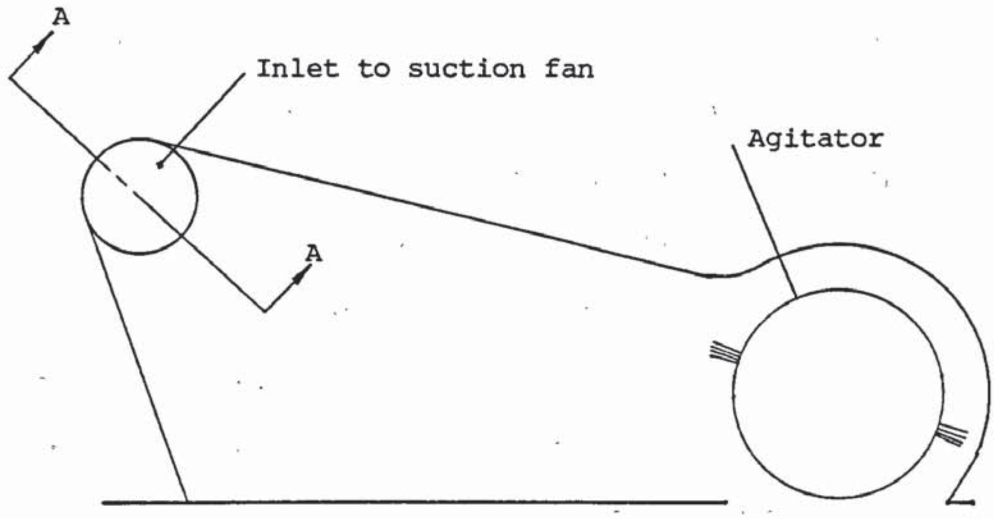


Figure 5.53 Sound level variation with motor speed and fan blade tip/chamber clearance.

a) "A-A" line of pronounced velocity profile



b) 1) standard cleaner inlet
2) straight inlet to suction fan
3) "smoothed" throat of standard cleaner

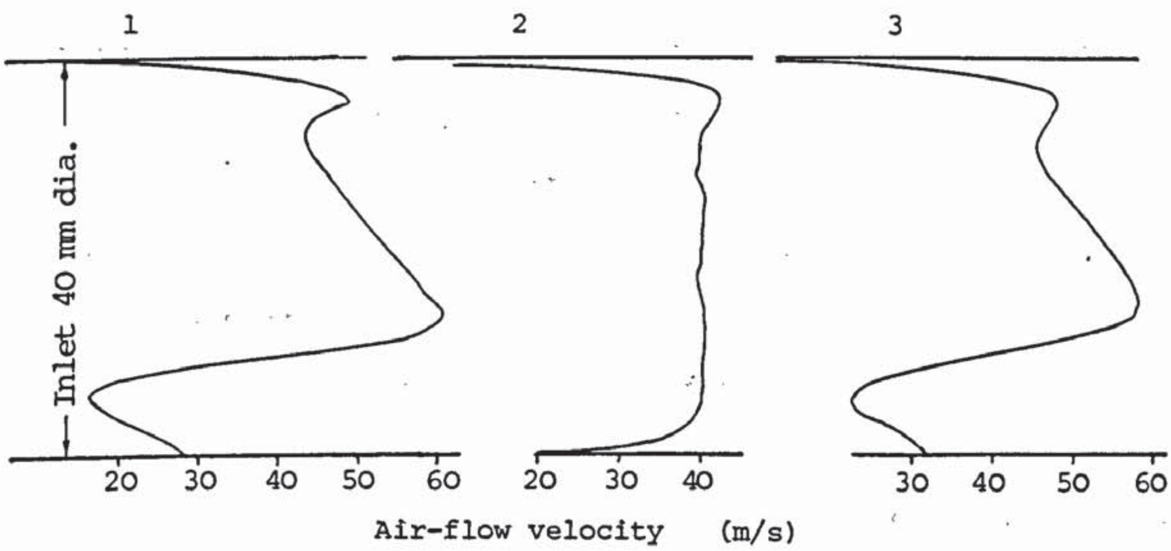


Figure 5.54 Velocity profiles at the inlet to the suction fan.

clamping of the motor and chassis assembly were sturdy. The hot-wire anemometer was calibrated before use and the readout compared well with the air-flow rate obtained at Perivale on the vortex flowmeter rig.

The velocity profile at the fan inlet is shown in the above figure. The profile is clearly uneven. For comparison, the velocity profile at the same plane for a straight inlet was obtained. This is superimposed on the above figure. It shows the normal symmetrical profile. It was thought that with this uneven profile, the suction fan would be starved on part of its revolution and therefore, the fan blades would experience fluctuations in resistance. This could be a source of a pure tone. Therefore the potential benefits of an even velocity profile were two-fold: the fan will output more air giving greater efficiency and the pure tone will be eliminated. Tiny vanes or deflectors just before the inlet bend would suffice to normalise the profile. This, however, was not possible because of the obstruction to debris. A rounded instead of a sharp-edged "throat" at the bend was evaluated. The resultant profile is presented in Figure 5.54 and showed very little improvement.

The presence of a "starved" region for the fan suggested that not only a strong 7th order would be present but also a second harmonic at 14th. Figure 5.55 presents the exhaust noise trace from a motor-chassis assembly and the 14th order is certainly present but low in magnitude. The sharp bend must be the source because without the bend the 14th order peak disappeared as shown.

The effect of the sharp bend on performance

A motor-chassis assembly was modified and certain other parts fabricated to enable an appraisal of the performance of the motor with a straight inlet. The results showed that the bend reduced the air-flow efficiency from 22.9% to 21.2% (7% from baseline) and the air-flow from 15.5 10^{-3} to 14.1 $10^{-3} m^3$/sec (9% from baseline).

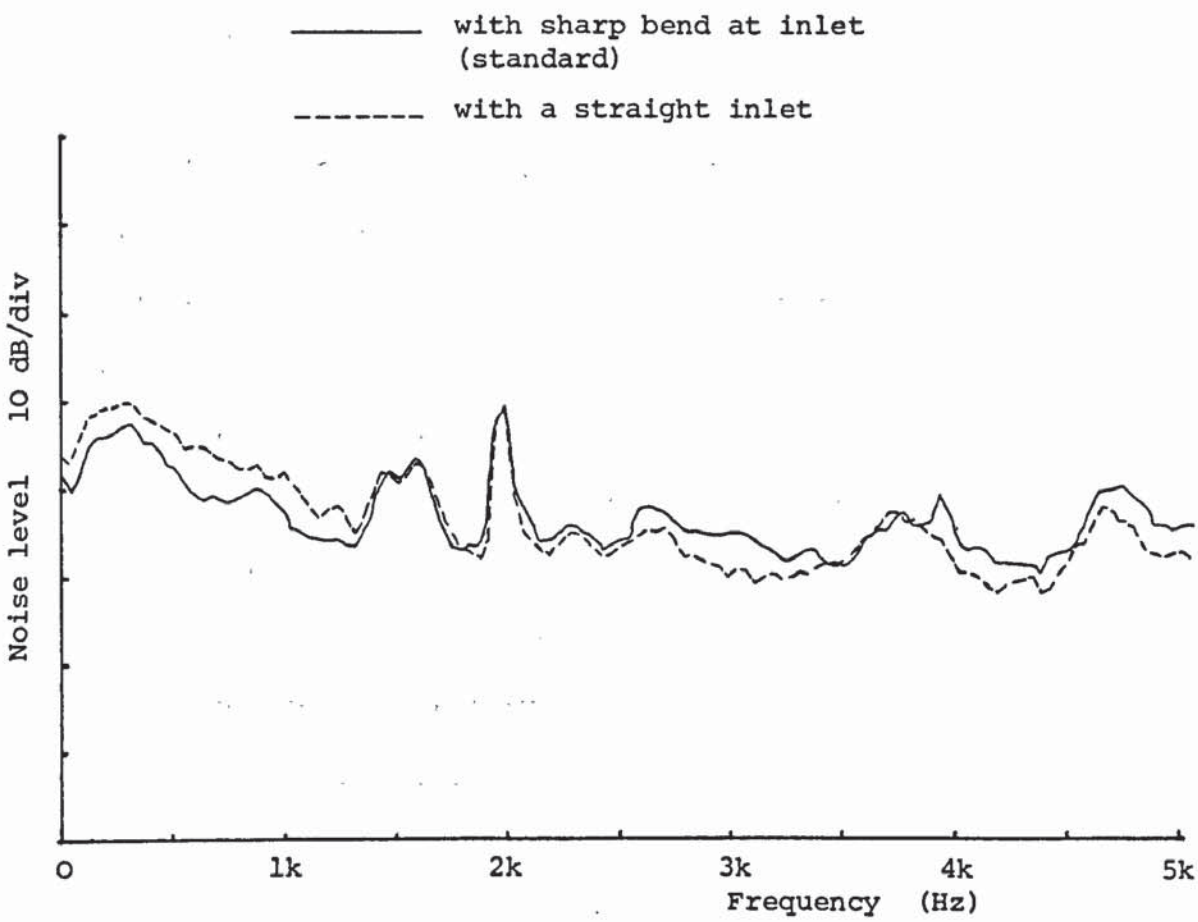


Figure 5.55 Effect of a sharp bend at inlet to the suction fan upon motor noise spectra.

5.8.7.5 The effect of leakage in the inlet duct

The following leakage points were observed in the chassis:

- badly fitted agitator thrust end caps into the chassis
- badly fitted bottom plate
- belt end throttle or "bleed" hole.

Each opening was sealed in turn and the performance was noted on the vortex rig. By sealing the badly fitted joints, the efficiency of the cleaner rose to 17.6% (+21.4% from baseline). The quality of assembly, therefore, should be improved because the associated loss in the cleaner efficiency was 3.0%. By blocking the bleed hole the cleaner efficiency rose to 20.4% (+40.7% from baseline). The bleed hole creates a passive air-flow from the belt side of the cleaner nozzle to the suction fan side and thereby prevents the collection of residue in the vicinity of the belt drive. Although the bleed hole is justified, its size should be reduced. Consequently, by improving quality, (or introducing a seal) and reducing the bleed hole size a cleaner efficiency of around 19% is possible. This is a rise of 32% on baseline performance.

State of cleaner	Air-flow rate 10^{-3} m^3/sec	Efficiency %
standard	16.5	14.6
all sealed except "bleed"	17.0	17.6
only "bleed" hole sealed	17.0	17.2
all air leaks sealed	16.9	20.4

5.8.7.6 Alternative suction fan blade profiles

A photograph of the standard suction fan showing its size and blade profile is given in Appendix A. Two alternative fans were supplied by Hoover, North Canton, for evaluation by Perivale staff. The following work was carried out by Don Brown, Development Engineer in the Floorcare

Laboratory at Perivale. Fans tested were,

- standard suction fan,
- highly swept back blades, larger O.D. sample B,
- highly swept back blades, small O.D. sample C.

There was very little difference in the air-flow efficiency for the motor; in fact the standard fan was best at 25%. Similarly, the standard fan tied with fan "B" for cleaner efficiency at 13.7%. As for the sound power level, a benefit of 1 dB(A) was obtained for fan "B" while "C" was a little noisier. In conclusion, the standard suction fan offered the best compromise.

	Fan type		
	Standard	Sample B	Sample C
		MOTOR ONLY	
Max airflow 10^{-3} m^3/sec	46.0	44.4	37.6
Peak efficiency %	25	24	22
Max sealed suction mmWater	742	723	648
Open airflow speed (rpm)	17800	18600	20200
	TURBOPOWER	SOFTBAG	CLEANER
Max airflow	37.0	36.0	31.5
Peak efficiency	13.7	13.7	12.8
Max sealed suction	610	608	530
Open airflow speed	17400	18000	19600
Sound power level dB(A)	88.0	87.0	88.5

5.8.8 Reduction of suction fan pure tone (Figures 5.56-5.57)

The narrow-band frequency spectra of the motor and cleaner noise showed a strong 7th order at approximately 2000 Hz. This was the blade passing tone caused by the seven blade suction fan. The frequencies which are most sensitive to the human ear are in the range 1-4 kHz. Therefore, a pure tone at 2 kHz would be quite noticeable and probably annoying. The blade passing frequency can be shifted up or down in frequency either by altering the motor speed or the number of blades.

To shift the pure tone to, say, 1500 Hz, the seven blade suction fan has to run at 1500/7= 214 Hz (12800 rpm). This is a big reduction in speed and, because the corresponding performance was likely to be significantly reduced, this method was not acceptable. Conversely, for a motor speed of 285 Hz the suction fan requires 5 blades for a pure tone at 1425 Hz. The subsequent effect on the air-flow and suction performance could easily be evaluated using the computer programme at Hoover, North Canton, but there was no access at Perivale. A reduction in performance was envisaged but nevertheless, this method could not be evaluated because of fabrication problems discussed below.

In addition to the two methods stated above, an asymmetric fan has been proved to attenuate and reduce the blade passing tone and replace it by less strong sidebands. So, an asymmetric fan and a cover on the tips of the blades of a standard fan were tested.

Asymmetric suction fan

Mellin and Sovran (1970) described a circumferentially-unequal blade spacing that reduced the tonal annoyance due to blade passing tone. They considered both a naturally balanced and unbalanced blade arrangement and presented blade spacings for fans having 4 to 15 blades. Spacing for seven blades was selected so that the air-flow and suction characteristics of the standard fan will be maintained. Three schemes were tested as shown on the next page.

Manufacture of the asymmetric fan proved to be problematic. The standard fan is made of polyproplene which is extremely awkward to fabricate. Hence a sample of the standard fan was made in ABS plastic by injection moulding in production tools and then, blades were cut at their lowest point, repositioned according to a particular scheme and permanently affixed. Initially glue was used, then the blades were pinned,

Blade spacing in degrees		
Mellin and Sovran		Sagoo
Unbalanced	Balanced	Unbalanced
43.5	40.7	42.0
48.7	68.3	45.5
57.0	52.5	48.5
61.6	37.0	51.5
57.0	54.0	54.5
48.7	67.4	57.5
43.5	40.1	60.5

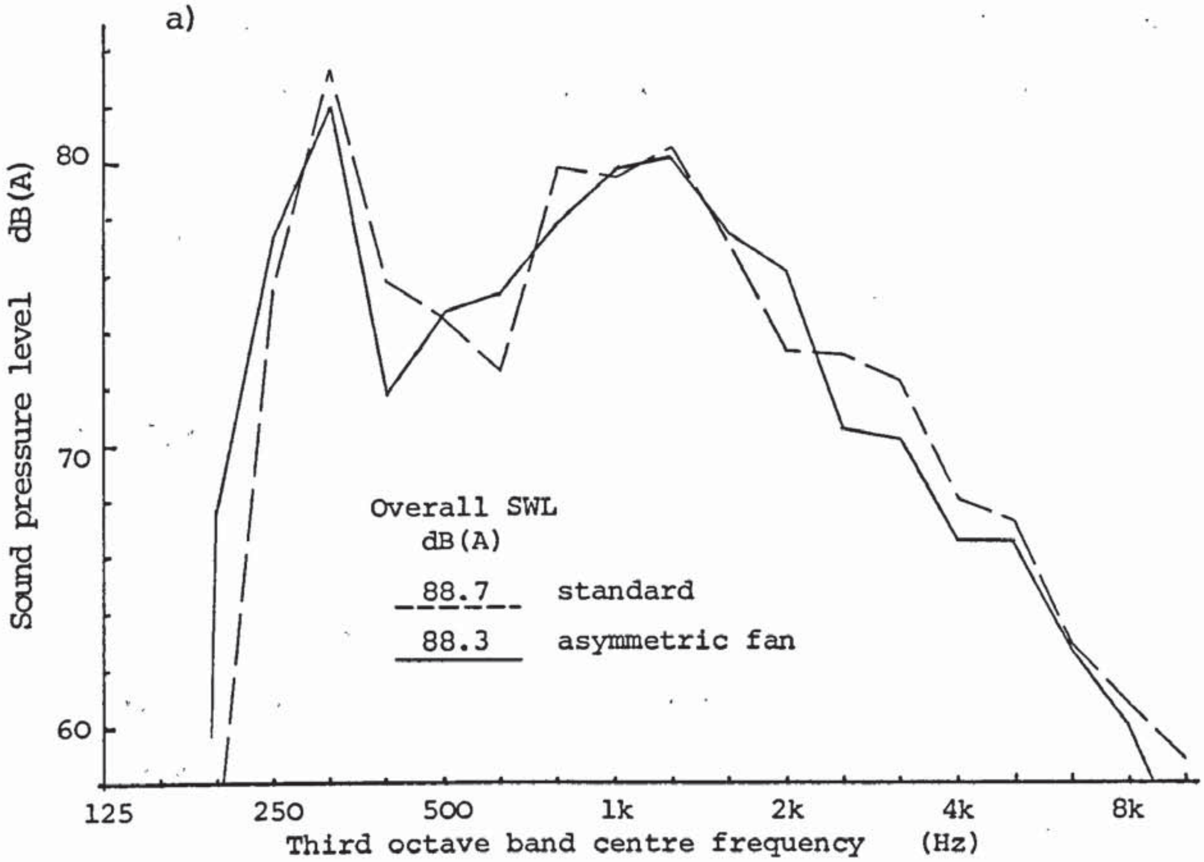
and finally the blades were glued and screwed from underneath. After preparation, each fan was dynamically balanced on the Jackson-Bradwell machine.

All possible precautions were taken so that the fabricated blades did not break prior to the test. Although the motor speed was increased very gently, only one of the fans actually survived. Blades on the other two fans came off. The results, in Figure 5.56a, show no real change. The narrow-band spectra, in Figure 5.56b, showed that sidebands were present but there was no attenuation of the 7th order.

Suction fan with ring on blade tips.

A blade produces a pressure disturbance every time it passes a fixed point. Therefore, the object of this modification was to minimise the pressure fluctuations by covering blade tips with a sheet of plastic to obtain a smooth upper surface. Naturally, the fan blade trailing edge could not be covered because of the flow of debris laden air.

Two fans were prepared. One had an aluminium ring screwed onto the blade tips and the other a thin plastic sheet (see Photograph A1.5). Both fans were dynamically balanced. The sound levels are plotted in Figure 5.57 and show a small increase in noise in the frequency range 2-5 kHz but



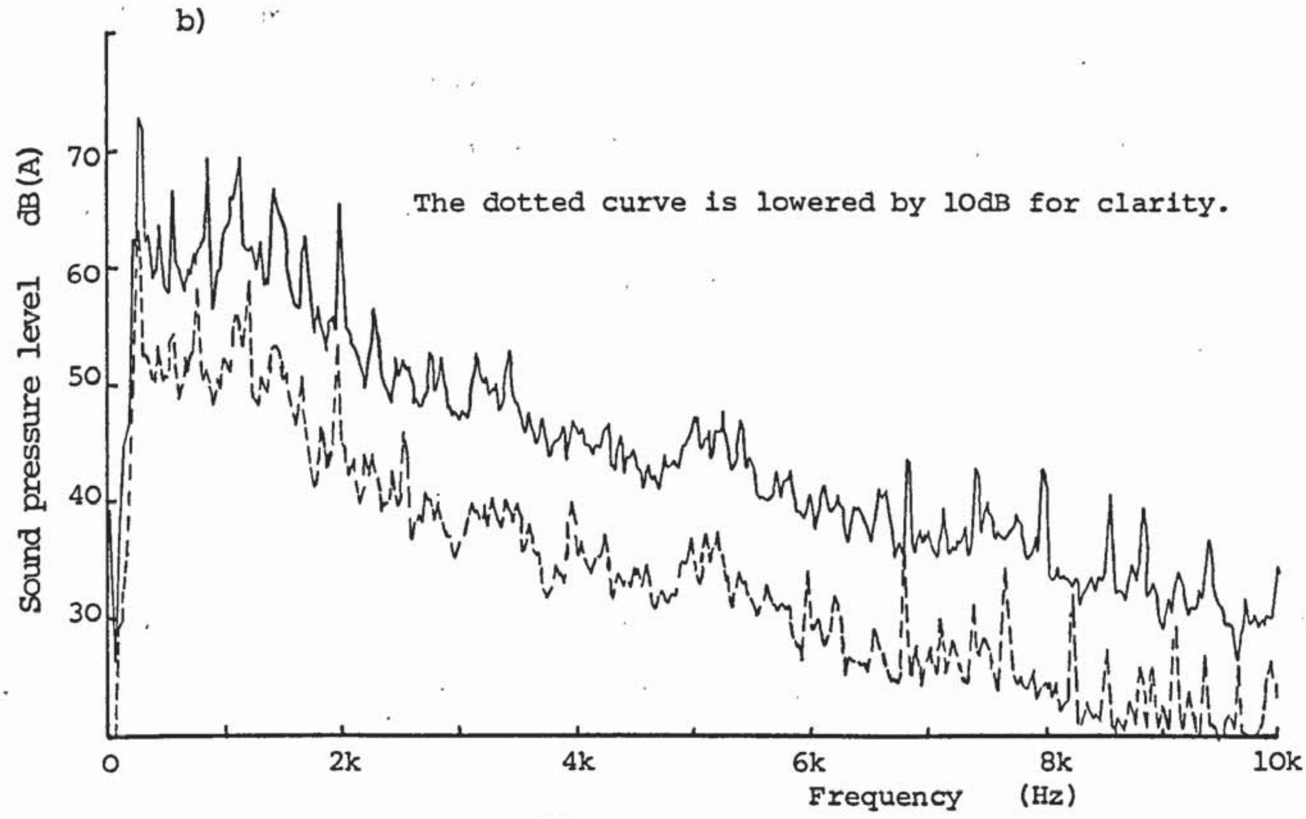


Figure 5.56 Evaluation of an asymmetric fan
a) third octaves and b) narrow-band.

this was non-contributable to the overall cleaner level. The 1.5 dB(A) reduction in the overall noise due to the plastic ring, is however, due to a change in the armature imbalance because the first order peak is abnormally low. In conclusion, there was no reduction in the overall noise level of the cleaner by a ring on the suction fan blade.

5.8.9 Air-flow over sharp edges (Figure 5.58)

There are many areas in a Turbopower cleaner where the air flows over sharp edges, such as,

1. the air entering the nozzle
2. the air flowing over joints within the air ducts, and
3. the exhaust air as it leaves the bag housings.

1. The air inlet region cannot be modified radically. Small alterations such as round-edges in the ducts were made.

2. It was thought that a great deal of noise would be generated by the air flowing through the corrugated bellows and at the joint of the bellows to the air duct tube because the leading edge of the latter faces the air flow. An internally smooth-walled flexible duct was evaluated in place of the bellows and its joint with the air duct was reversed. There was, therefore, no sharp and abrupt edge obstructing the air stream. This modification had another benefit in that it would free a possible blockage area. However, there was no reduction in the overall noise level of the cleaner.

3. Downstream of the suction fan, the cleaner is under positive pressure. Hence, its design does not require sealing of joints. After the dust has been deposited in the dust-bag the air passes through the tiny pores in the paper, and out of the bag housings via a gap along the periphery. To eliminate the flow of air over sharp edges, the air was exhausted via two new apertures, one in the thin web of the lower cover plate next to the existing opening for the bellows/top fill tube at bottom

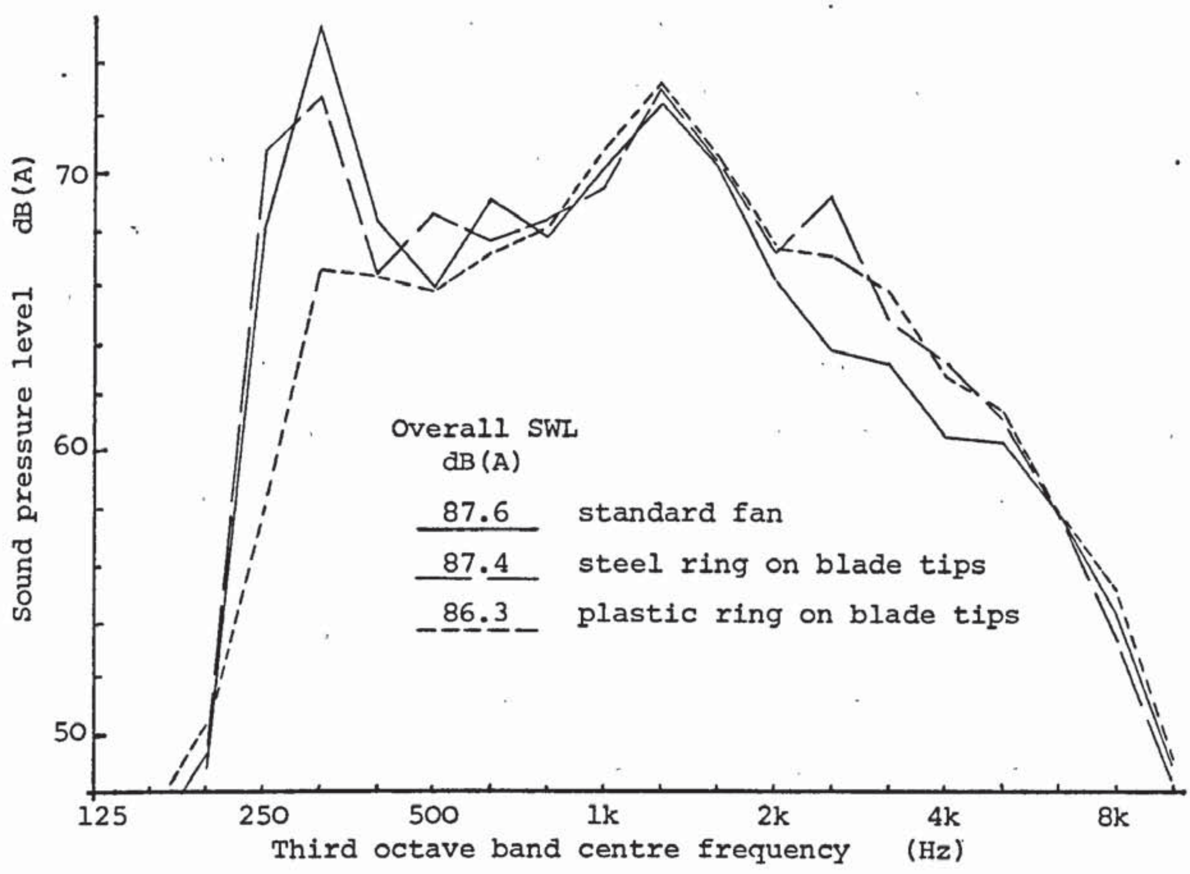


Figure 5.57 Evaluation of a ring on fan blade tips.

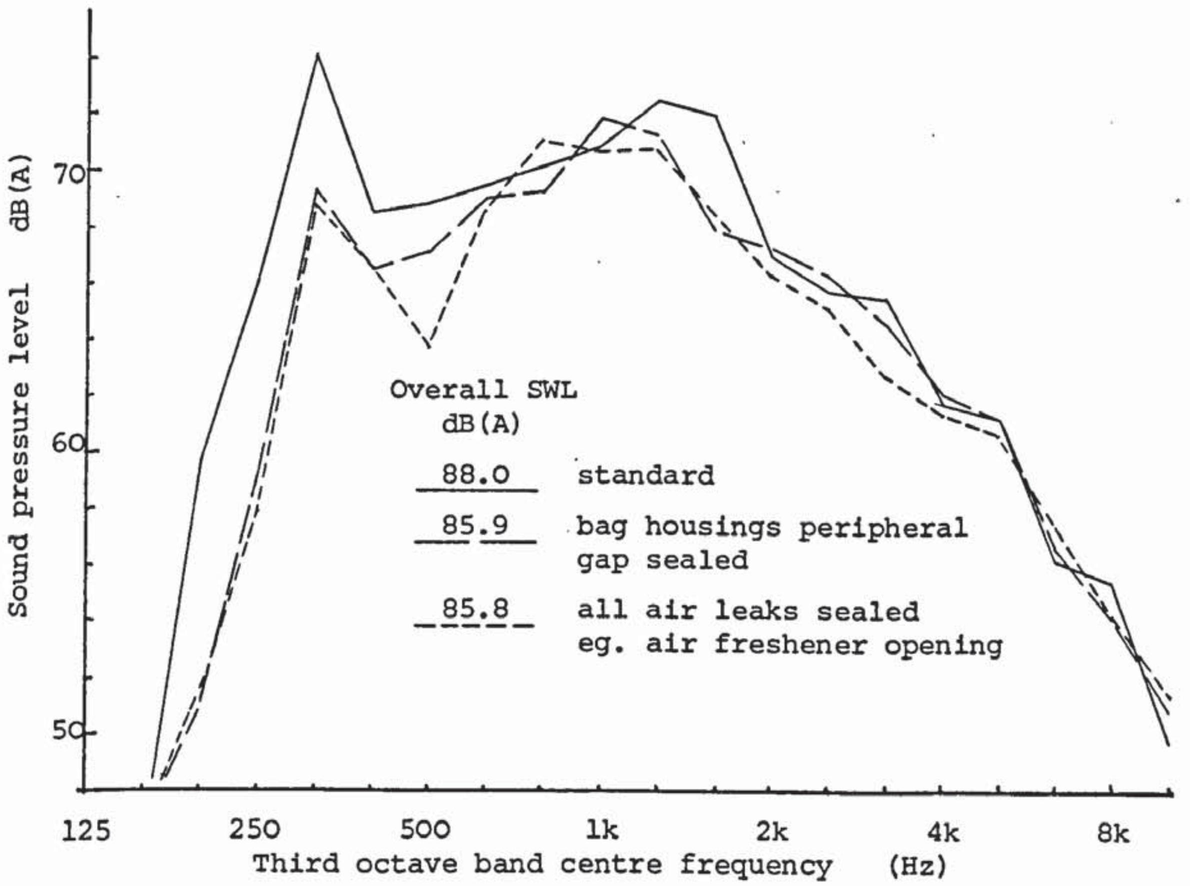


Figure 5.58 Cleaner noise level vs. control of exhaust air.

of bag compartment, of size 70mm x 55mm, and the other in the panel left of the lower cord storage hook, of size 35mm x 50mm. The aperture area was larger than the minimum sectional area in the air ducts so as not to increase the air speed inadvertently. Both apertures were internally lined with micro-filters.

The modification introduced a more torturous path for the air-flow and therefore benefitted from the extra inherent attenuation which lowered the overall noise of the cleaner by 2.1 dB(A). It can be seen in Figure 5.58 that the reduction comprised frequencies up to 2 kHz. Although some reduction of high frequency noise was envisaged, this was not clear from the results because of noise radiation from the hood and bellows areas. Low frequency noise, upto 630 Hz, was attenuated because this modification strengthened the upright box section. In conclusion, the most practical way in which to reduce the air turbulence noise was by sealing the gap between the dustbag housings.

5.8.10 Attenuation of air noise (Figures 5.59-5.64)

A Helmholtz resonator built into the fan chamber and alternative types of silencers for the top fill tube were evaluated to attenuate noise in the air stream.

5.8.10.1 Helmholtz resonator

It is a characteristic of the basic Helmholtz resonator that it attenuates sound in a narrow range of frequency given by;

$$f = \frac{C}{2\pi}\sqrt{\frac{A}{Lv}}$$

where; C - velocity of sound in air, 342 m/s

A - cross area of neck, m^2

L - length of neck, m

v - volume of cavity, m^3

The objective was to evaluate the Helmholtz resonator at a point on the fan chamber. A hole of diameter 5mm in the fan chamber formed the neck of length 3mm (thickness of fan chamber) and a tube of internal diameter 27.5mm was used with an moveable plunger to vary the volume of the cavity. The frequency equation becomes,

$$f = 54.43\sqrt{11.02/L}$$

It was not possible to install the hood with this modification, consequently, the sound measurements for the cleaner were made without the hood. The result is given in Figure 5.59. Despite the small attenuation at the predicted frequency, there was very little difference in the overall noise level. In practical terms, there was no room in the vicinity of the fan chamber for a large cavity except if the cavity were made shallow. As the preliminary results were not encouraging, no further work was done.

5.8.10.2 Alternative silencers for top fill tube

The aim was to use the basic principles of exhaust silencer design and reduce the noise emission from the duct and thereby reduce the overall noise of the cleaner. The following were postulated and tried for the hardbag cleaner;

i. a standard softbag muffler with and without metal baffles (modified inlet and outlet)

ii. the hardbag air duct as a straight through silencer

iii. the hardbag air duct comprising baffles

iv. air duct enlarged at its middle

v. air duct vibration-isolated from ends

vi. centre of air duct replaced by dense foam tube

vii. centre of air duct replaced by plastic tube with baffles covered with foam.

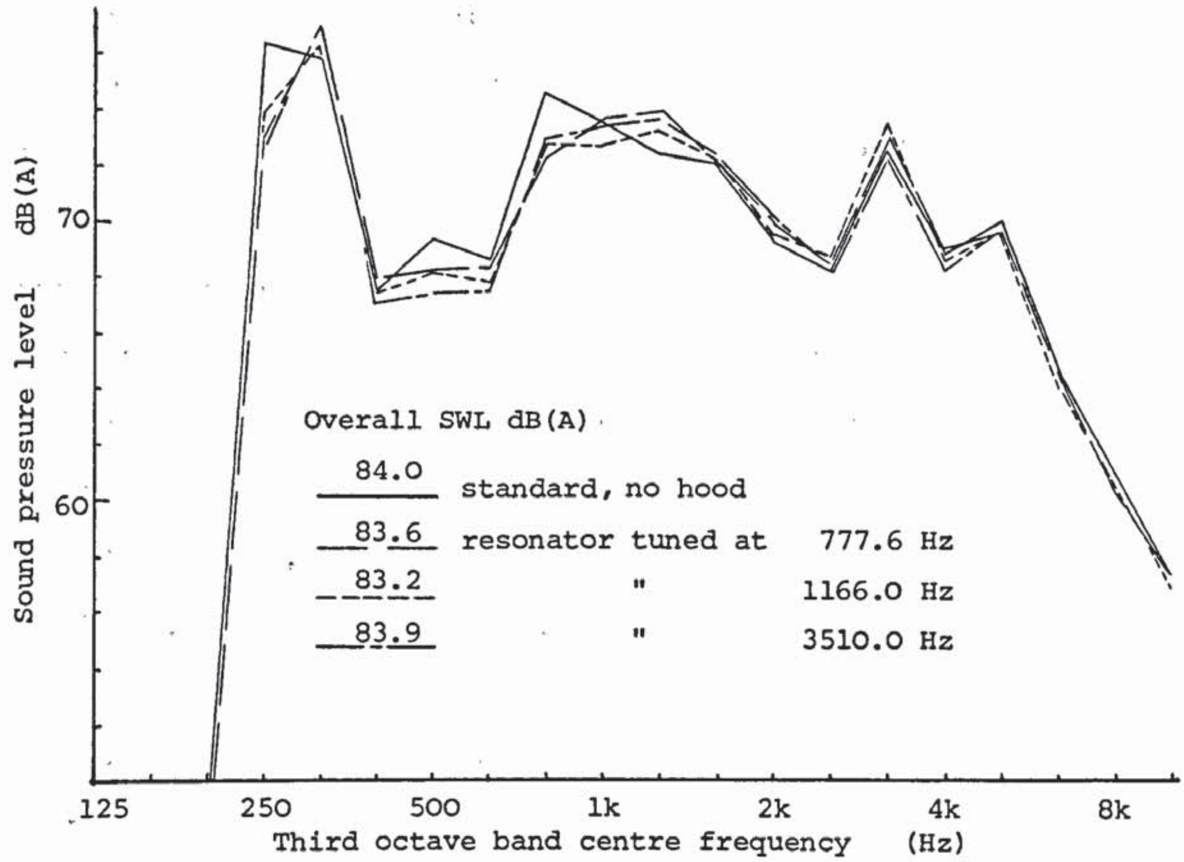


Figure 5.59 Cleaner noise levels with a Helmholtz resonator positioned on fan chamber.

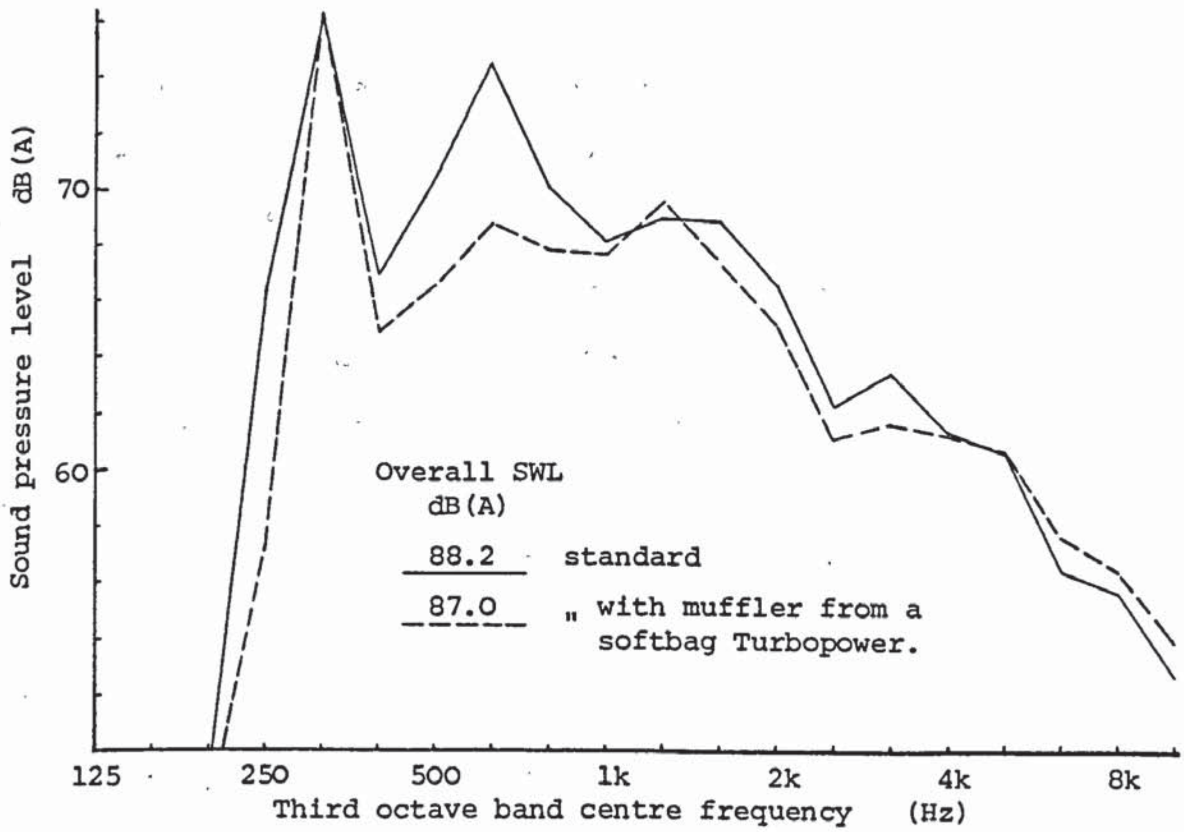


Figure 5.60 Cleaner noise levels with muffler from a softbag cleaner.

Tests using a muffler from a Turbopower softbag

Some work had been done at Hoover plc to lower the noise level by means a muffler. That work was aimed at reducing noise level of the then newly launched softbag Turbopower. The softbag muffler comprised a rectangular section and contained a metal baffle. It reduced the noise of the softbag cleaner by 3-4 dB(A). In the standard form, the softbag muffler does not fit the hardbag. Therefore, the inlet and the outlet of a normal hardbag air duct were fabricated onto the muffler. The results, given in Figure 5.60, showed a definite noise reduction in the frequency range 1-2 kHz, but there was no reduction in the overall cleaner noise level.

The hardbag air-duct as a straight through silencer

It was extremely important to exclude obstructions and "zero-airflow" regions in the flow of air which might result in a blockage. Therefore, a straight-through silencer seemed best suited to this cleaner. A resonator-silencer comprises a tube with a series of circumferential orifices which are covered by an expansion chamber, shown in Figure 3.13. According to the 'Handbook of Noise and Vibration' (TTP 1979, p558), the sound transmission loss for this type of silencer depends upon the volume of the expansion chamber and the diameter and number of orifices. The object was to use the dust bag housing as the expansion chamber. This is drawn in Figure 5.61. It was not possible to vary the volume, so the attention was concentrated at the diameter and the number of orifices.

The rear face of the air duct virtually touches the rear housing cover in-situ and so cannot be utilised because no absorbing material could be placed. Consequently, the front and side faces of the air duct were available for sound treatment. The number and diameter of the holes required to expose a percentage of the total area are tabulated below. For a preliminary investigation, only the tests which are underlined were chosen. It was anticipated that a sensitivity analysis of the results

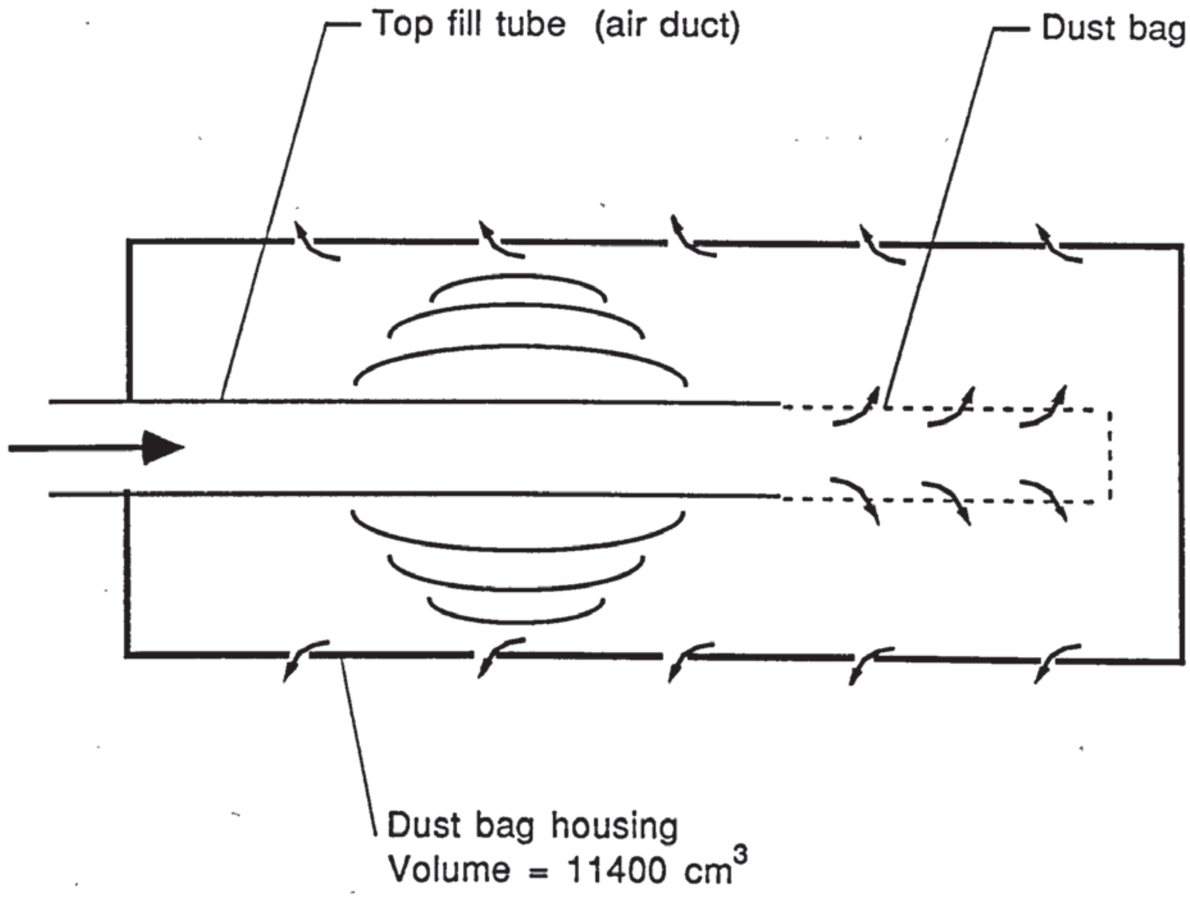


Figure 5.61 The Turbopower hardbag considered as a silencer.

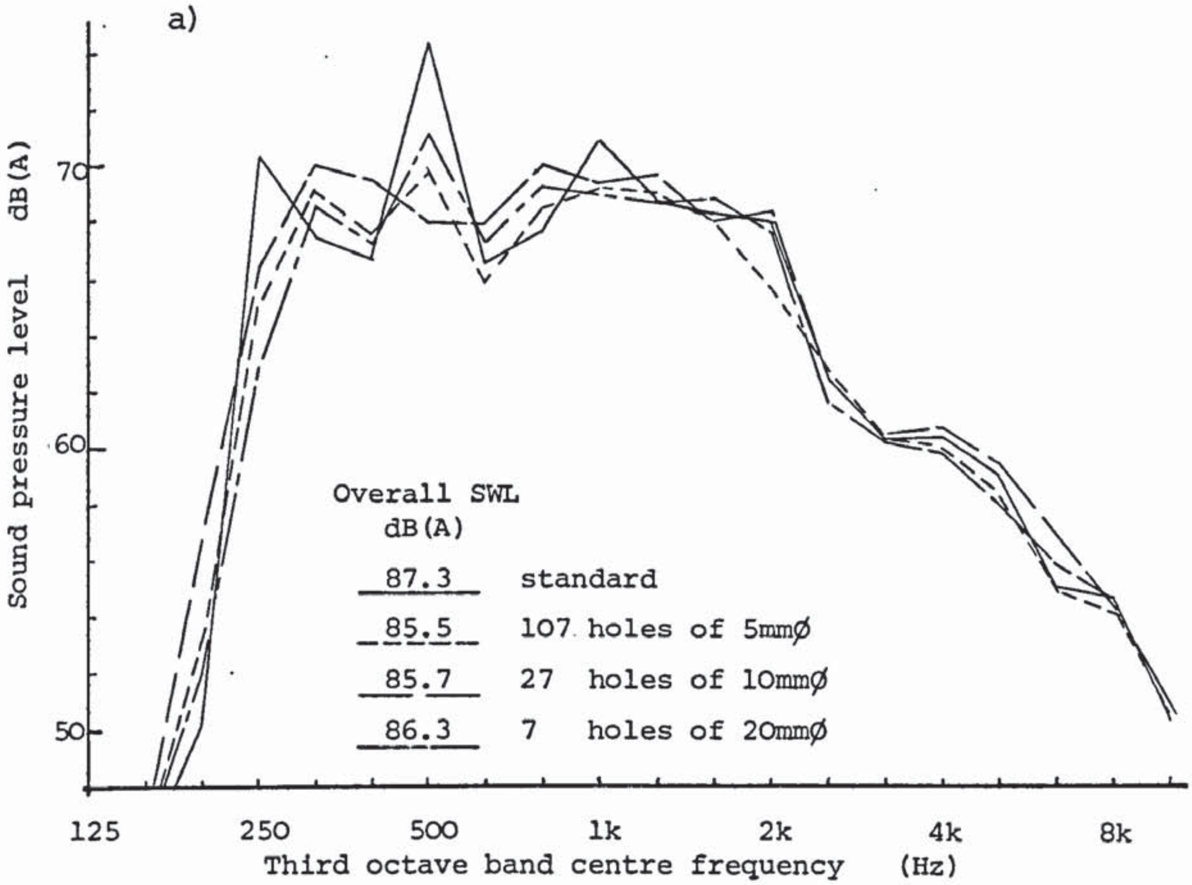
would produce the most critical factor, if any. The evaluation was extended by three conditions, as follows,

- bare holes,
- holes covered by sellotape,
- holes covered by sellotape and 7mm of polyurathane foam.

Hole Dia. mm	Area of hole mm^2	Number of holes required to expose % total area 5%	15%	25%
5	0.196	107	321	540
7.2	0.40	52	157	262
10	0.785	27	80	135
15	1.767	12	36	60
20	3.142	7	20	34

The sound levels, plotted in Figures 5.62a and b, showed that noise reduction was not particularly encouraging: about 1-1.5 dB(A) at best. Even this was lost when a re-check of the standard cleaner itself resulted in 1.5 dB(A) less noise. The actual number of holes, whether or not covered with tape or foam, had no influence on the noise level, as shown in Figures 5.62c and d. The greatest improvement was centred on the 315 Hz third octave which contains the out-of-balance component. From experience, this was known to be extremely susceptible to removal and refit of the motor.

According to Davies (1973), the "noise reduction potential of a single expansion resonator with correctly chosen dimensions, is on average 10dB attenuation over a limited frequency range, and further enlargement of the expansion chamber is not beneficial". This implies that for a silencer tuned to a specific frequency, say 1200 Hz, there will be a potential drop of 10 dB around 1200 Hz. This may not result in a large



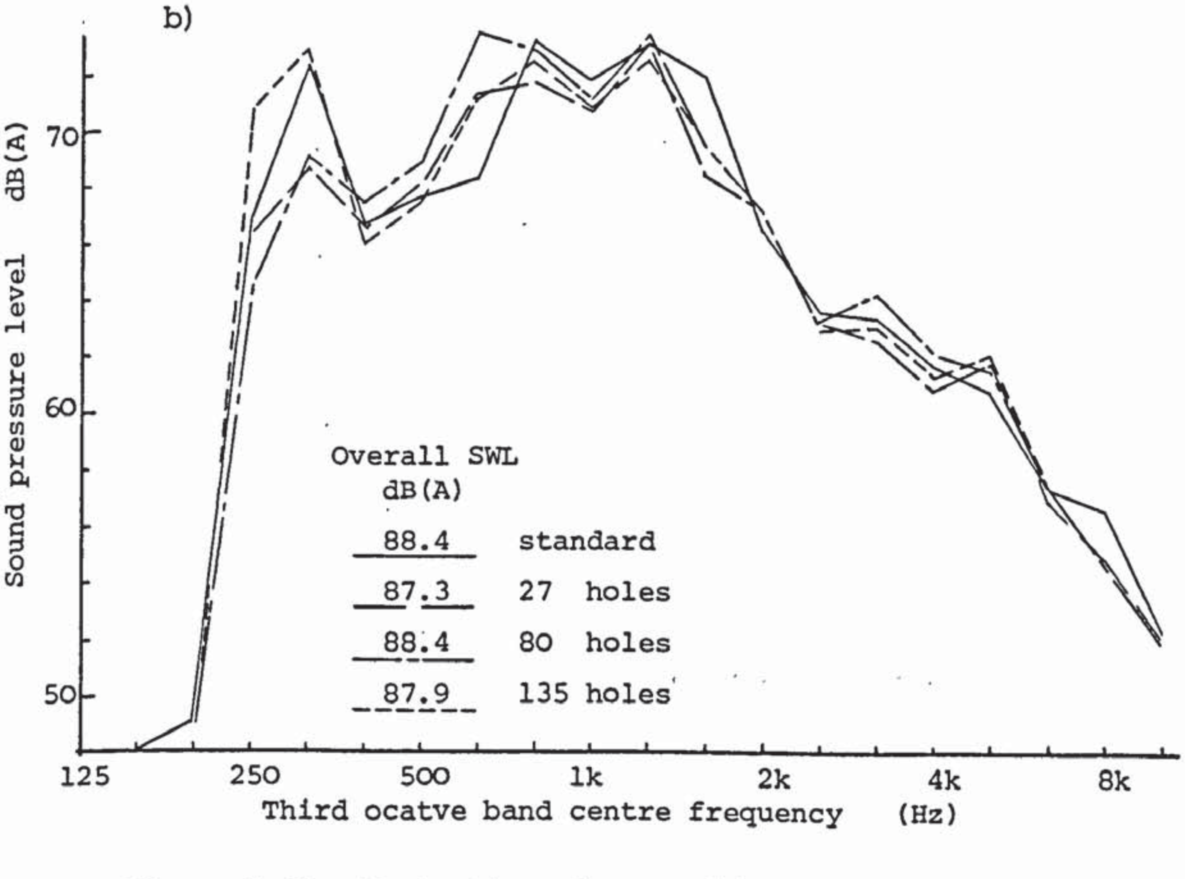
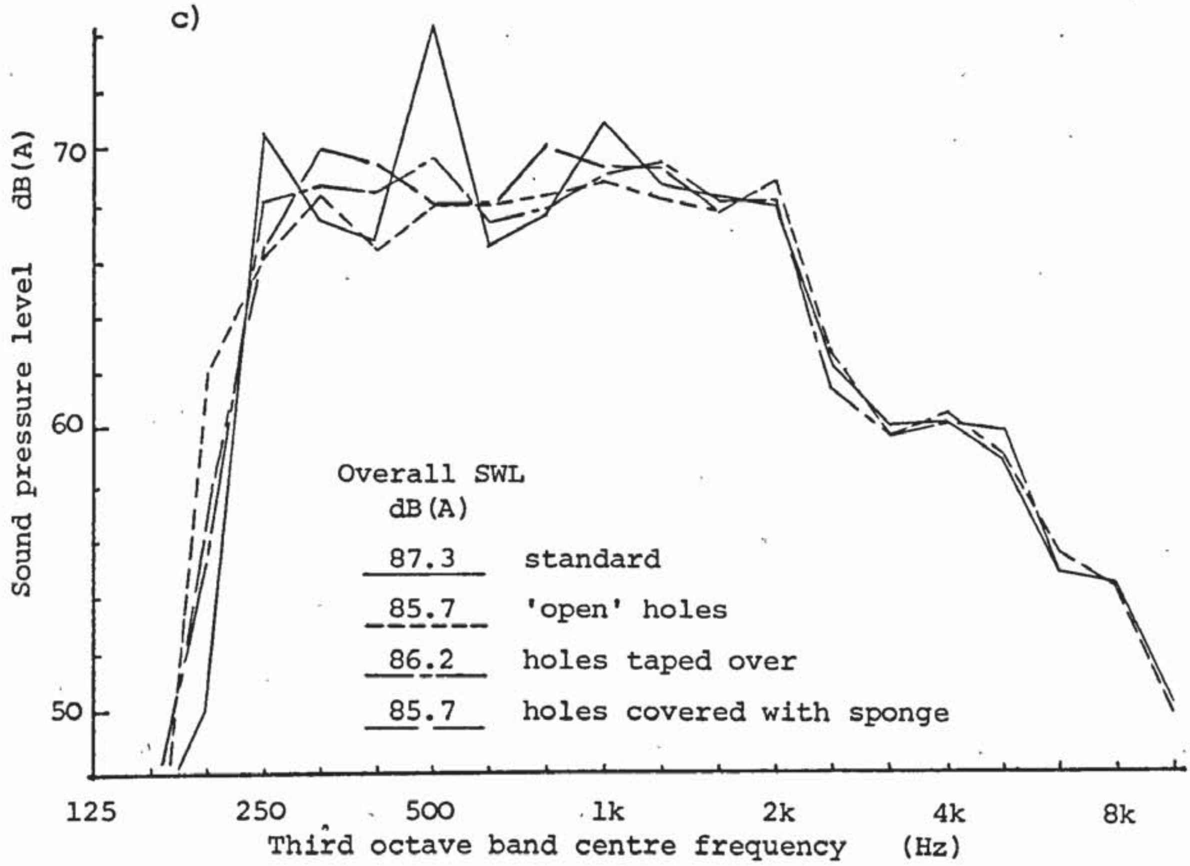


Figure 5.62 Evaluation of a straight through silencer.
a) total exposed area of 5%
b) 10mm dia. holes.



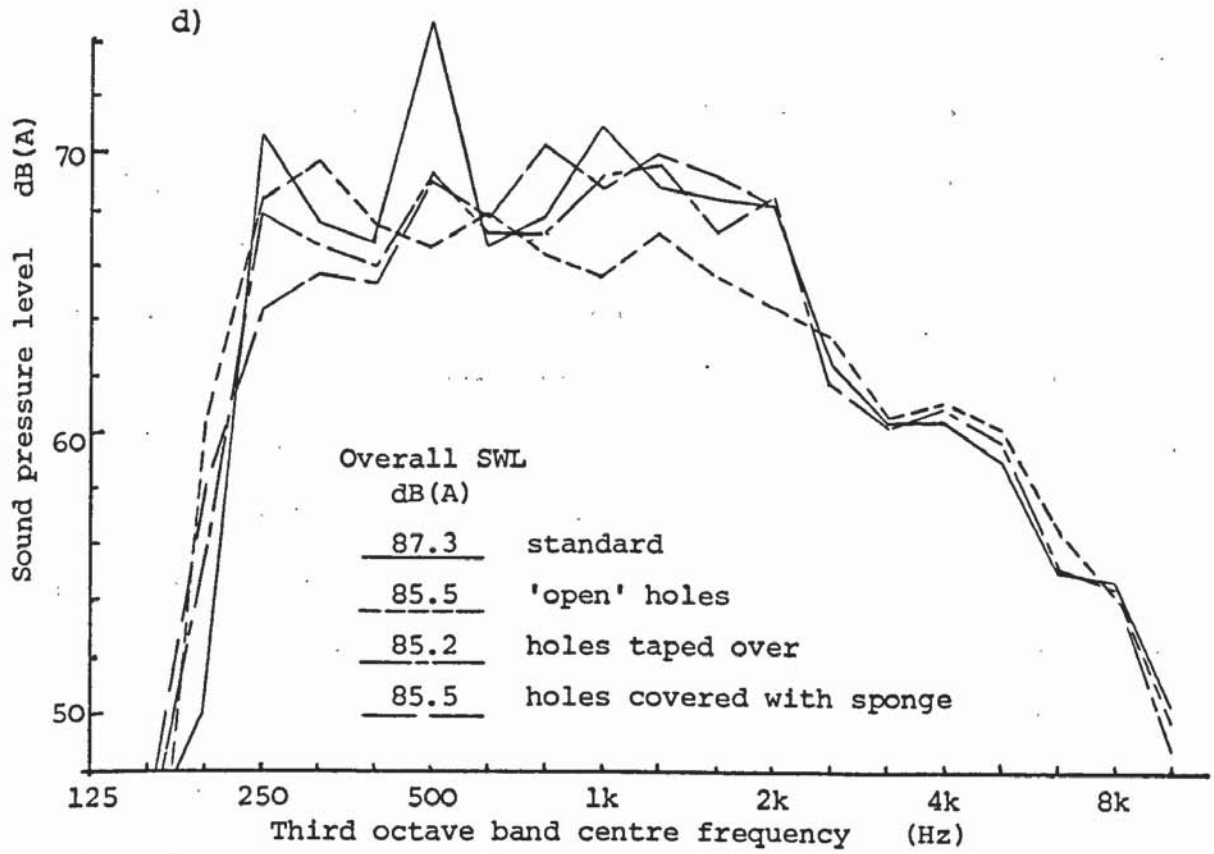


Figure 5.62 Evaluation of a straight through silencer.
c) 27 holes of 10 mm dia.
d) 80 holes of 10 mm dia.

reduction in the overall cleaner noise level because there are many low frequency peaks. It would be interesting to determine the potential drop in the overall cleaner noise level due to an "ideal" silencer. This is demonstrated by the following test.

The B & K frequency analyser, type 2107, has a "frequency rejection" facility. This offered a sharp attenuation at the chosen frequency. The result of frequency-rejection centred at 1200 Hz is shown in Figure 5.63. This showed that the 1250 Hz band was reduced by 11.1 dB(A) but the overall noise was reduced by only 1.8 dB(A). Frequency rejection centred at 630 Hz and 315 Hz gave similar results. Therefore, it appeared that the attenuation covering a limited range of frequencies will have a potential of around 1.8 dB(A) reduction in the noise level of the cleaner. Further improvement can only be obtained with noise reduction covering a wider frequency range.

Air duct comprising baffles

14 baffles of size 13mm by 50mm were cut in the front face of the air duct and covered by polyurethane foam. The results are plotted in Figure 5.64a. Again, there was little difference in both the overall noise level and third octave values.

Middle portion of the air-duct enlarged

The shape of the air duct alters from circular at its inlet, to a square, then rectangular with deep sides, then gradually transforms into shallow rectangle and finally to a circular profile at outlet. The cross-sectional area remains almost constant. The centre part was enlarged enough to allow foam lining. The results are included in the previous figure and show very little difference to the overall noise level.

Middle portion of the air-duct was vibration isolated

Greater vibration levels would be expected at the lower end of rear dust bag housing because it is hinged to the cleaner chassis. As a result, the lower part of the air-duct would experience greater vibration. A test

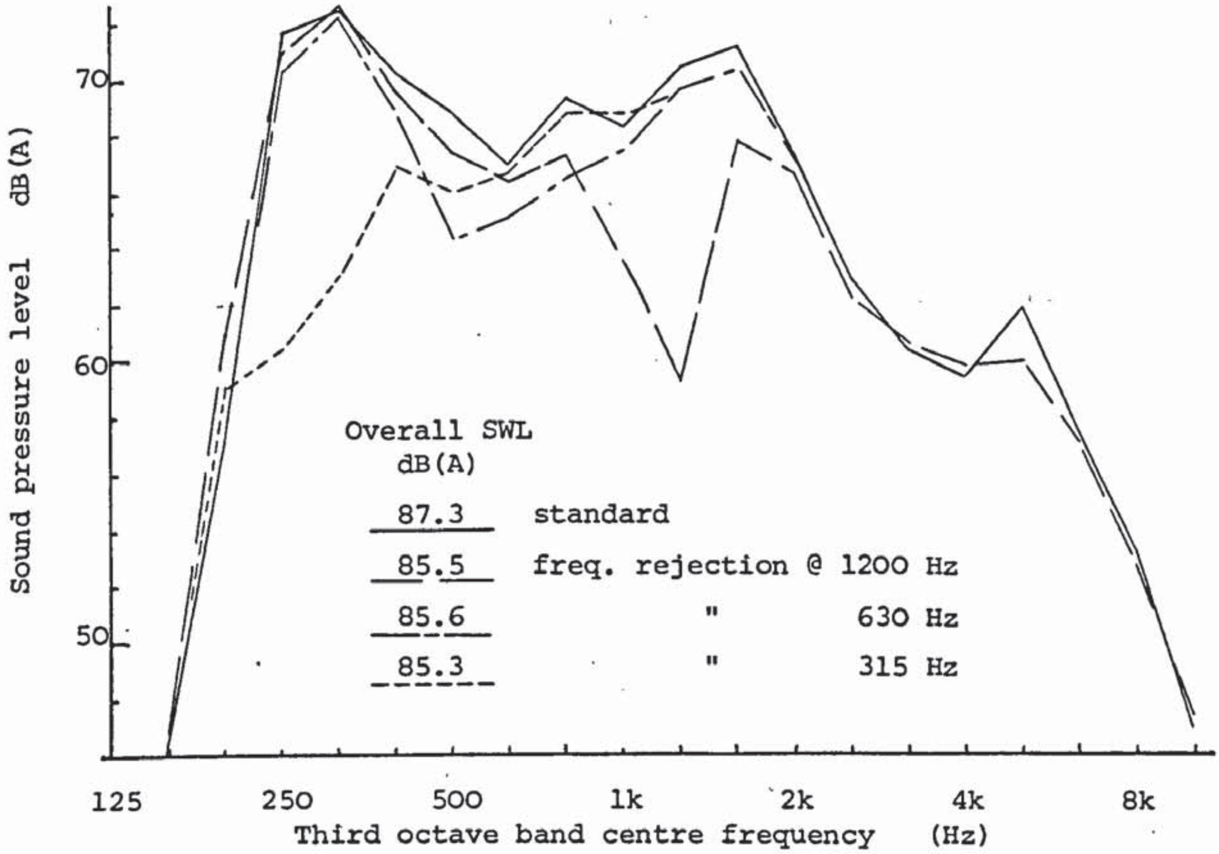
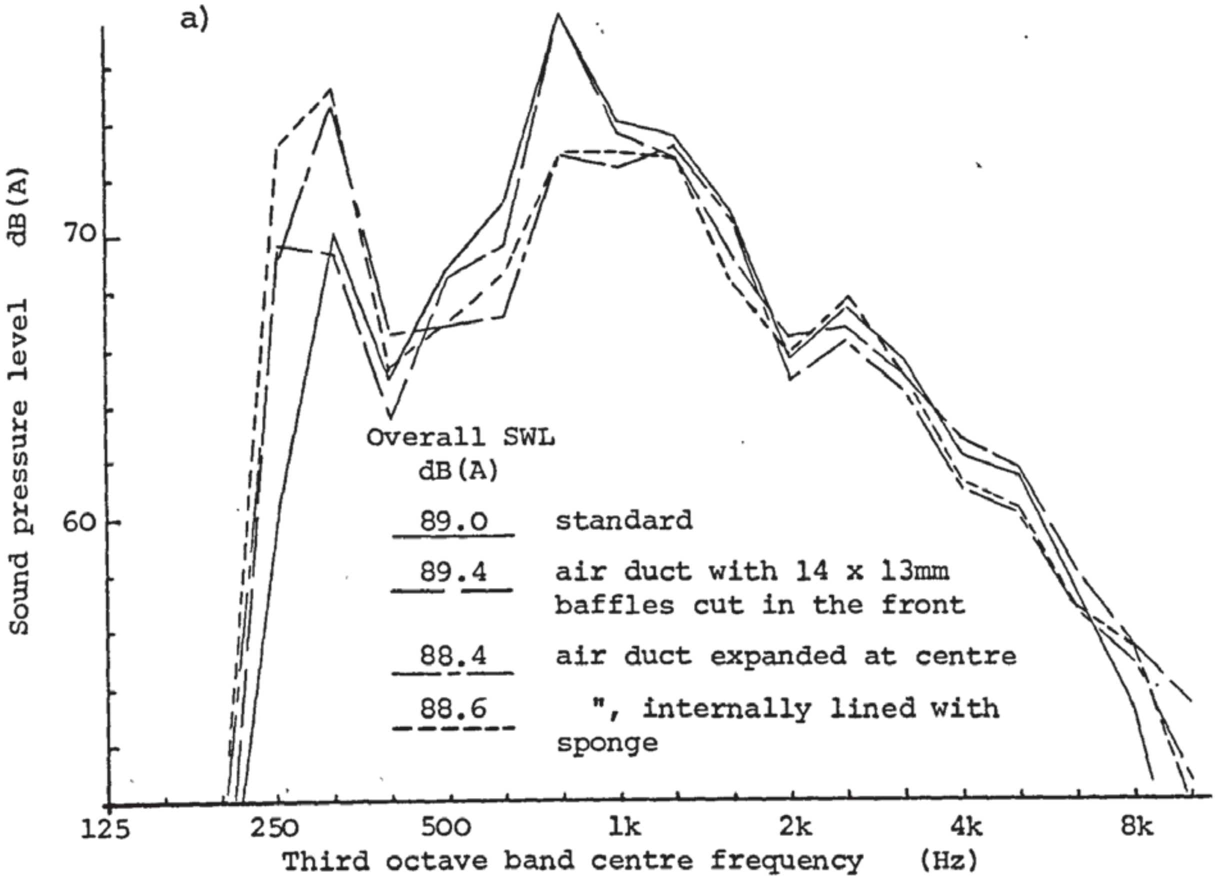


Figure 5.63 The effect of a sharp, narrow-band attenuation on a standard Turbopower cleaner noise.



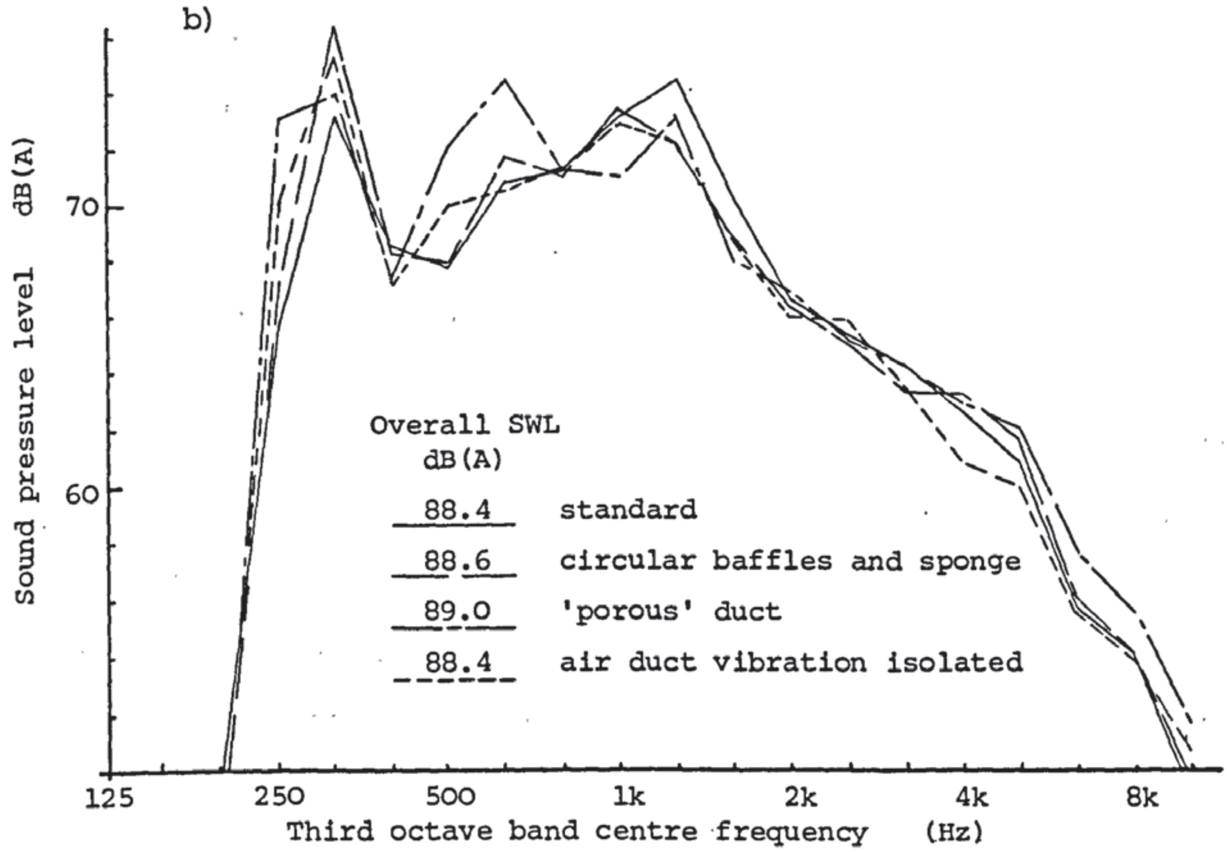


Figure 5.64 Evaluation of alternative silencers a) and b).

was undertaken to reduce the vibration excitation by isolating the centre of the air duct. The results, shown in the previous figure, indicate hardly any reduction in the noise level.

The air-duct replaced by dense foam tube

The constantly varying shape of the air-duct was replaced by a tube of similar but constant section area comprised of foam material. The foam was expected to promote sound attenuation by reducing the air turbulence but there was no reduction in noise, as shown in Figure 5.64b.

The air-duct replaced by plastic tube with baffles

The middle portion of tube comprised circular baffles covered with sponge. The result was consistent with the results of all other tests on silencers: there was no noise reduction.

In conclusion, none of the proposed silencers was effective in reducing the cleaner noise. On studying the schematic view of the exhaust side of the Turbopower cleaner, shown in Figure 5.61, it can be argued that a simple resonator system is already present. The air-duct represents the inlet to the silencer, the dust bag represents very fine perforations and terminates the inlet pipe, the bag housings provide the expansion chamber. It is possible therefore, that any additional silencers based upon the above modifications, will not make a significant difference.

5.8.11 Vibration of the hood (Figures 5.65-5.67)

The effect of the hood on the sound spectrum of a standard cleaner is shown in Figure 5.65a. The removal of the hood has two effects. Firstly, the noise response beyond 2kHz is increased and secondly, low frequency peaks, up to 2 kHz, are reduced. This implies that the hood, as a plastic sheet, offers sound absorption and insulation for frequency beyond 2 kHz, and acting as a large panel, radiates low frequency sound because of vibration excitation. The excitation of the hood due to vibration can be reduced by,

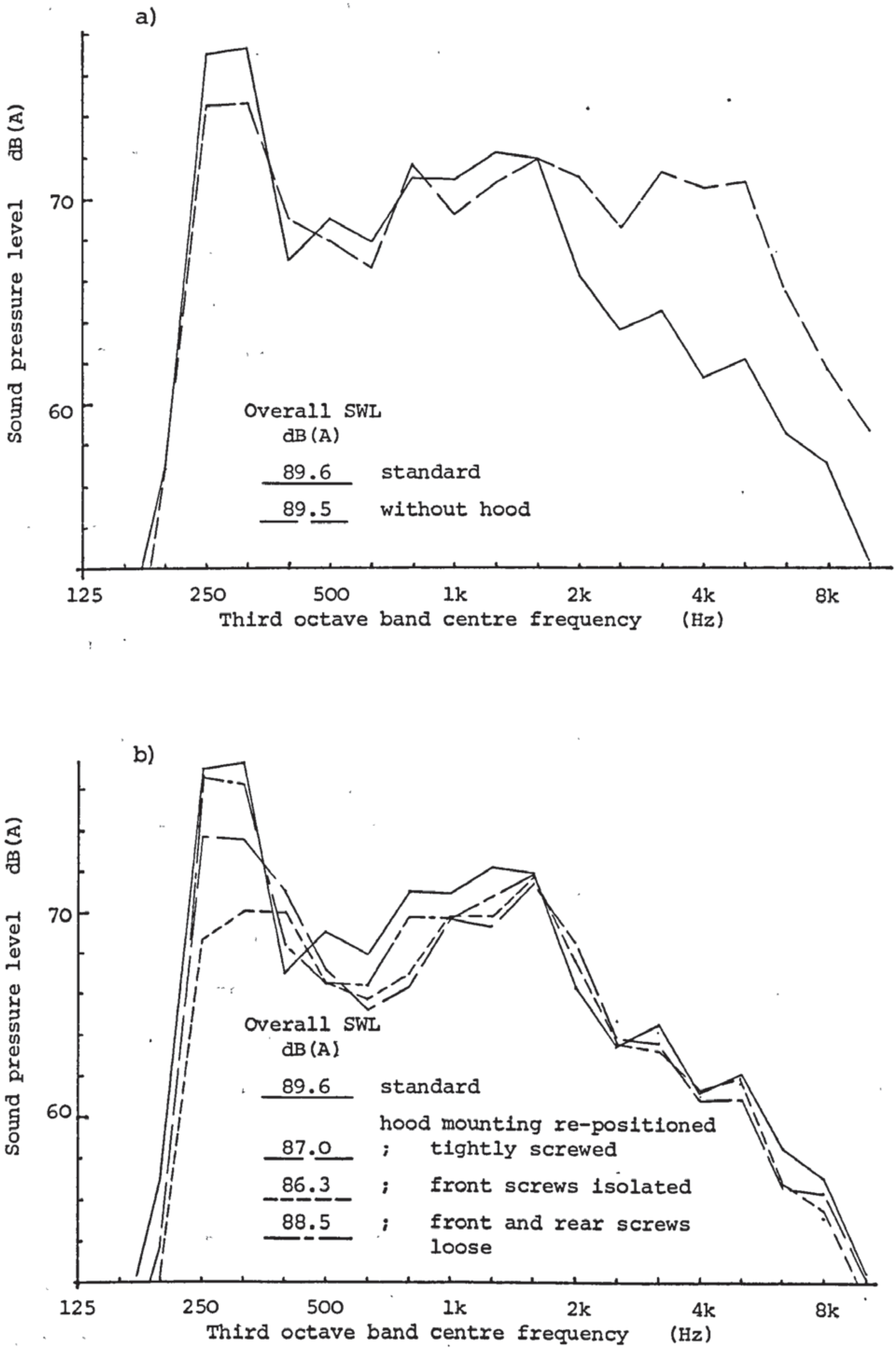


Figure 5.65 Evaluation of a) the hood and b) its re-positioned mounting on cleaner noise.

- optimising the hood mounting points,
- reducing the transmission of motor vibration,
- reducing the vibration of the hood by damping materials.

Hood mounting points

The present mounting points of the hood were thought to be close to the mounting of the motor in the chassis. So, the present two-point mounting was replaced by two at the front and two towards the rear of the cleaner. The results were as follows,

	Sound power level dB(A)	
	U2194	U2198
standard cleaner with the modified hood and chassis,	89.6	94.5
hood removed,	89.5	88.7
new mounting; loose fitting,	87.0	88.0
" ; tightly screwed,	88.0	90.5
" ; front screws isolated	86.3	88.0

For the average cleaner, model U2194, there was no change in the overall noise when the hood was removed, and there was a reduction of 2.5 dB(A) by the new mounting. For the second cleaner, model U2198, which had a sound level of 94.5 dB(A), the results showed that the removal of the hood gave 6 dB(A) reduction. The hood was, therefore, a prominent secondary sound source. The new mounting gave 4 dB(A) reduction in the noise. Therefore, this modification proved to be a success.

It was thought that the new mounting could be optimised to offer further noise reduction. Figure 5.65b illustrates that by releasing the front mounting screws, a slight reduction in noise was obtained. By loosening the rear screws there was an increase. This meant the vibration path at the front was strengthened when screws were tightened. Therefore, some noise reduction was obtained by isolating the front screws. Isolation of the rear screws was not possible in practice.

Reduction in the transmission of vibration.

There are two vibration paths to be considered, namely the motor mounting to the chassis and the chassis to the hood. The study of the former will be discussed in section 5.8.17. At present, the hood fixing screws form a direct path between the chassis and the hood and, therefore, rubber washers were used to break the vibration path as shown in Figure 5.66.

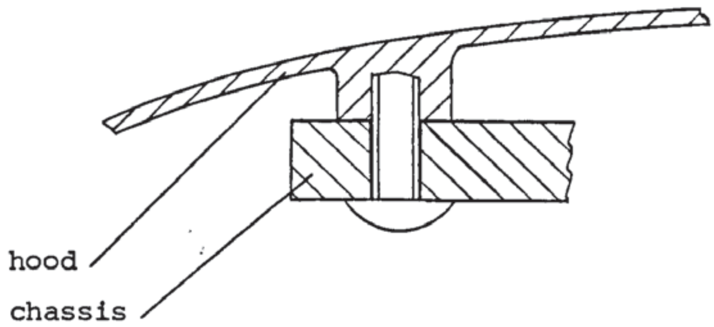
Figure 5.67a presents the narrow-band noise spectra of the standard and modified cleaners. The overall noise level was reduced from 88.1 dB(A) to 86.4 dB(A). Low frequency peaks were well attenuated and the vibration levels of the hood were reduced dramatically, as shown in Figure 5.67b.

Application of damping materials

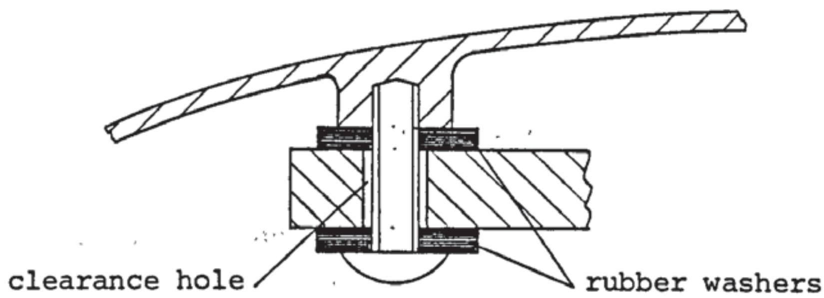
Since vibration to the hood is fed at all contact points and it was not possible to consider modification to all such points damping materials were applied. The corresponding noise results are given in the table below. There was little difference in the narrow-band results, except that the first order peak was lower for all of the modified cases. Using either a sponge or a damping pad reduced the cleaner noise by 2 - 2.5 dB(A).

	SWL dB(A)
standard cleaner,	88.2
with 8mm thick sponge under hood,	85.8
with damping pad under hood,	86.3
with sponge and damping pad	85.5

In conclusion, all of the modifications to the hood were successful. A greater reduction in the noise level was obtained on a noisier cleaner than on the standard.



Existing



Proposed

Figure 5.66 Proposed vibration isolation of the hood.

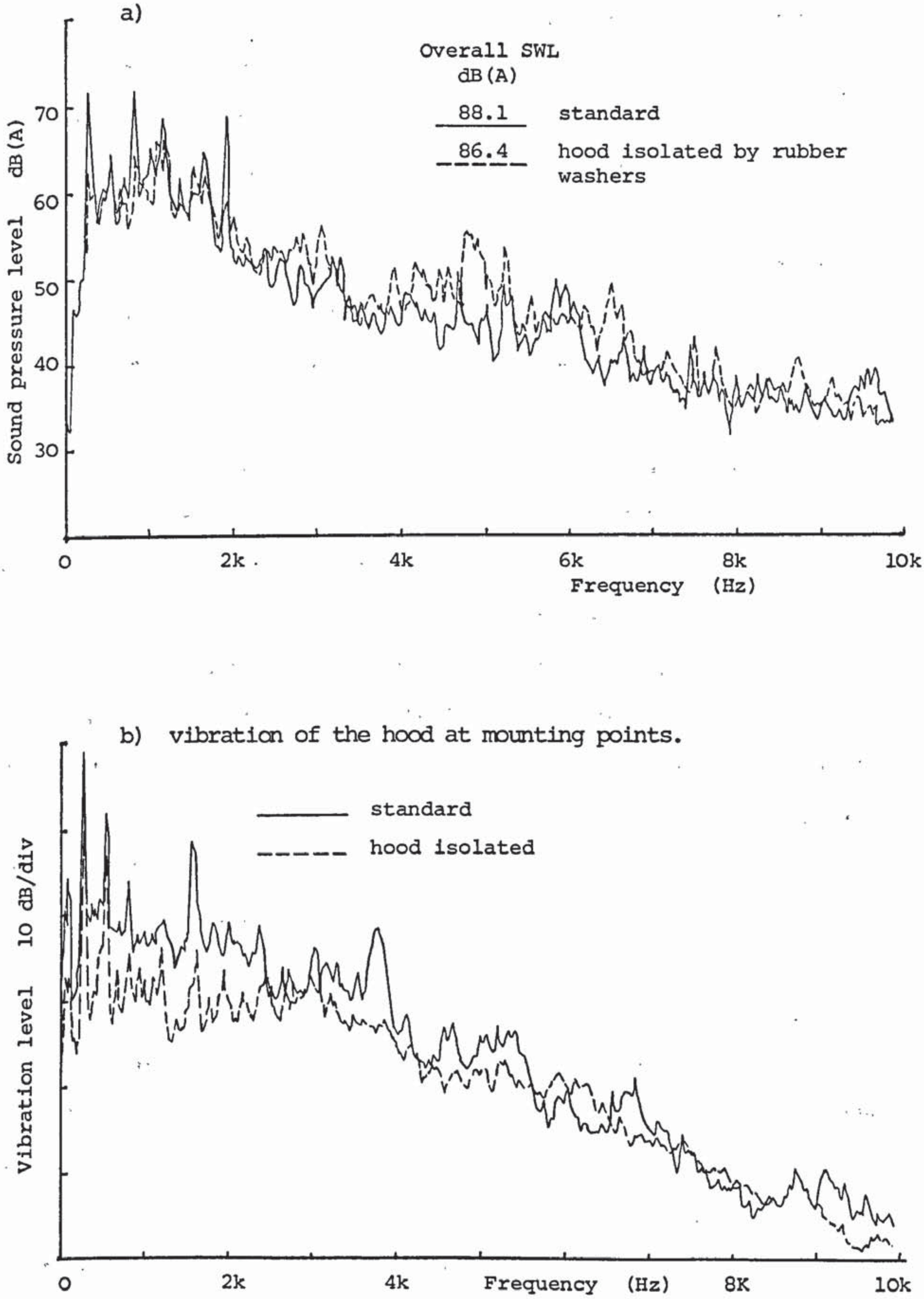


Figure 5.67 Effect of vibration isolation of the hood on a) noise spectra and b) vibration of hood.

5.8.12 Vibration of the wheel trim (Figures 5.68 and 5.69)

A loose panel chatters when disturbed by vibration. The example of the wheel trims was taken merely to illustrate the effect of vibration of a loose panel on the overall noise of the cleaner. There were other components such as the wheels and the front bag housing which rattled, but only the wheel trim was evaluated.

The wheel and the snap-fit trim are made of ABS plastic. The wheel runs on a steel pin and a slight clearance causes the wheel to shake when spun freely. The trim is fixed to the wheel at three points as shown in Figure 5.68. In many cases the wheel trims are loose and rattle when the wheel assembly is tapped. The sound levels in third octaves are plotted in Figure 5.69. The results showed a reduction of 1.0 dB(A) by removing the wheel trim, and similar reduction for a "sealed" trim. The attenuation in the frequency range of 400-800 Hz was due to the modified wheel trims.

In conclusion, there was a small but nevertheless, a definite noise reduction associated with a tight wheel trim. More significantly, the modification prevented the annoyance due to rattle. The trims can be tightened by either extending and stiffening the ridge on the attachment leg as shown in Figure 5.68, or by inserting a rubber ring between the wheel and the wheel trim.

5.8.13 Noise radiating from the bellows (Figure 5.70)

The near-field sound level above the corrugated bellows was 105.6 dB(A). This was 5.2 dB in excess of the second worst measurement. The proposed modification was for a plastic sheet to cover the bellows.

It was hoped that information of the sound transmission loss for plastics would indicate an alternative material to ABS. However, as mentioned in Chapter 3 there were no such data available for plastics. The measurement of the transmission loss was envisaged during this project, but after discussions with the supervisory team and staff of the Institute

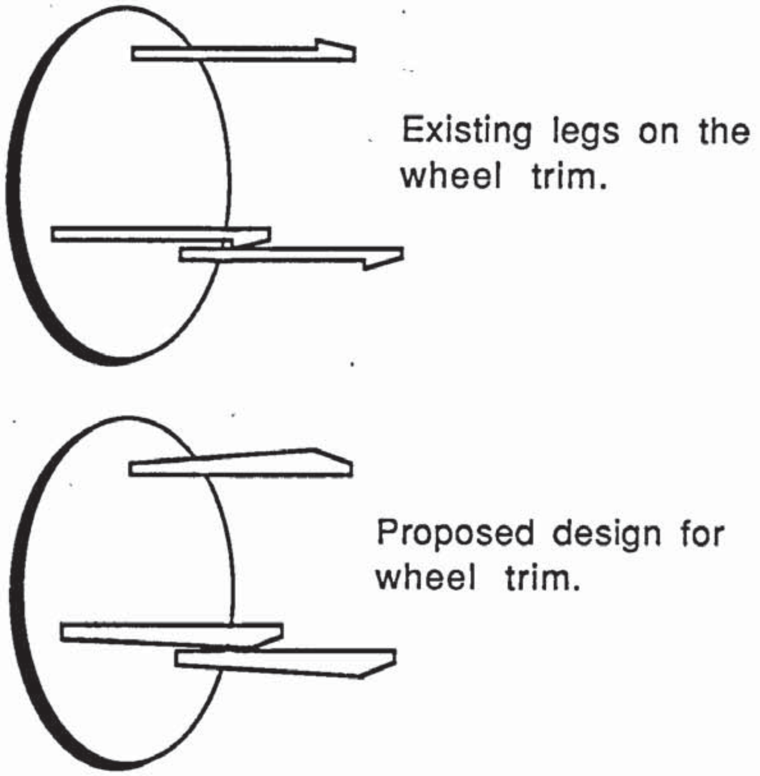


Figure 5.68 Proposed design change of the wheel trim legs.

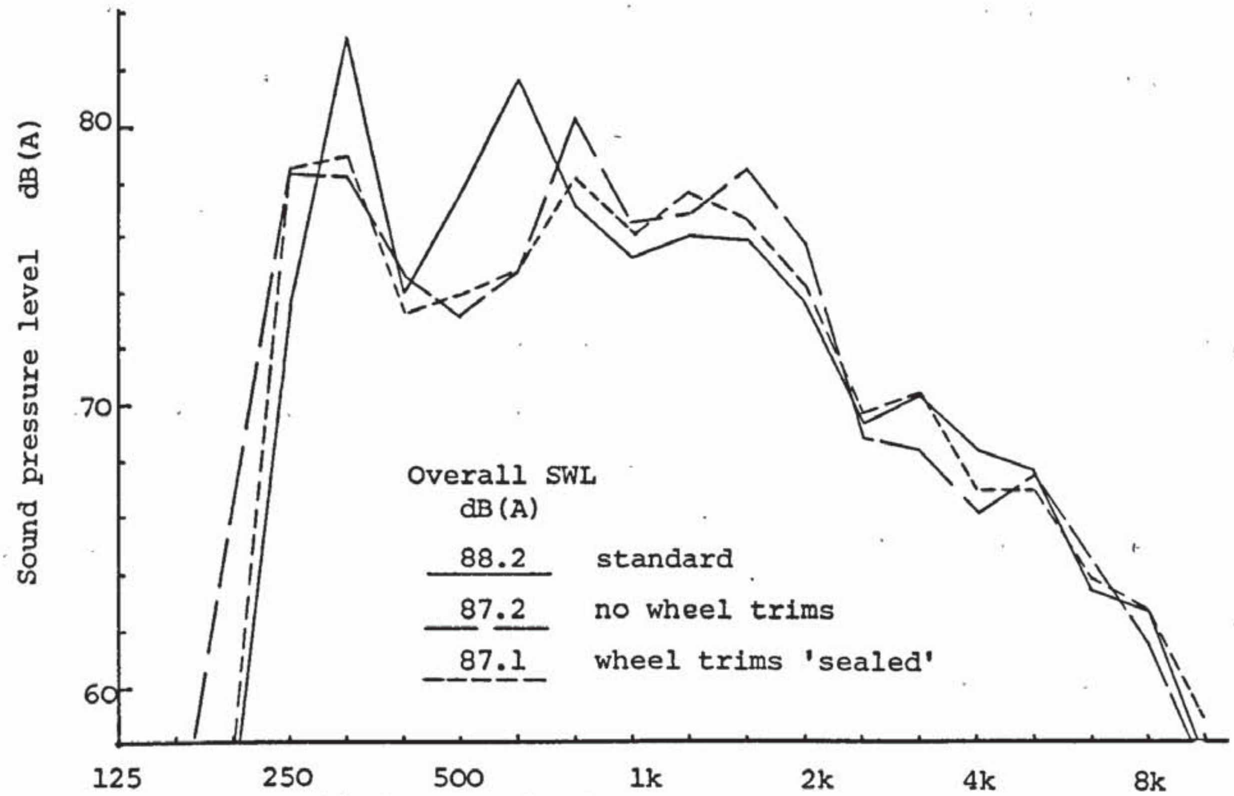


Figure 5.69 Effect on cleaner noise by sealing wheel trims.

of Sound and Vibration Research, Southampton, it appeared that the construction of a proper experimental rig was going to be time-consuming because there are many important factors to consider. Furthermore, any compromise in the set-up would cause the results to be specific rather than of general use. It was, therefore, decided not to proceed with this investigation and consequently, the noise reduction was predicted empirically and verified by experiments.

Experimental cover-up of bellows

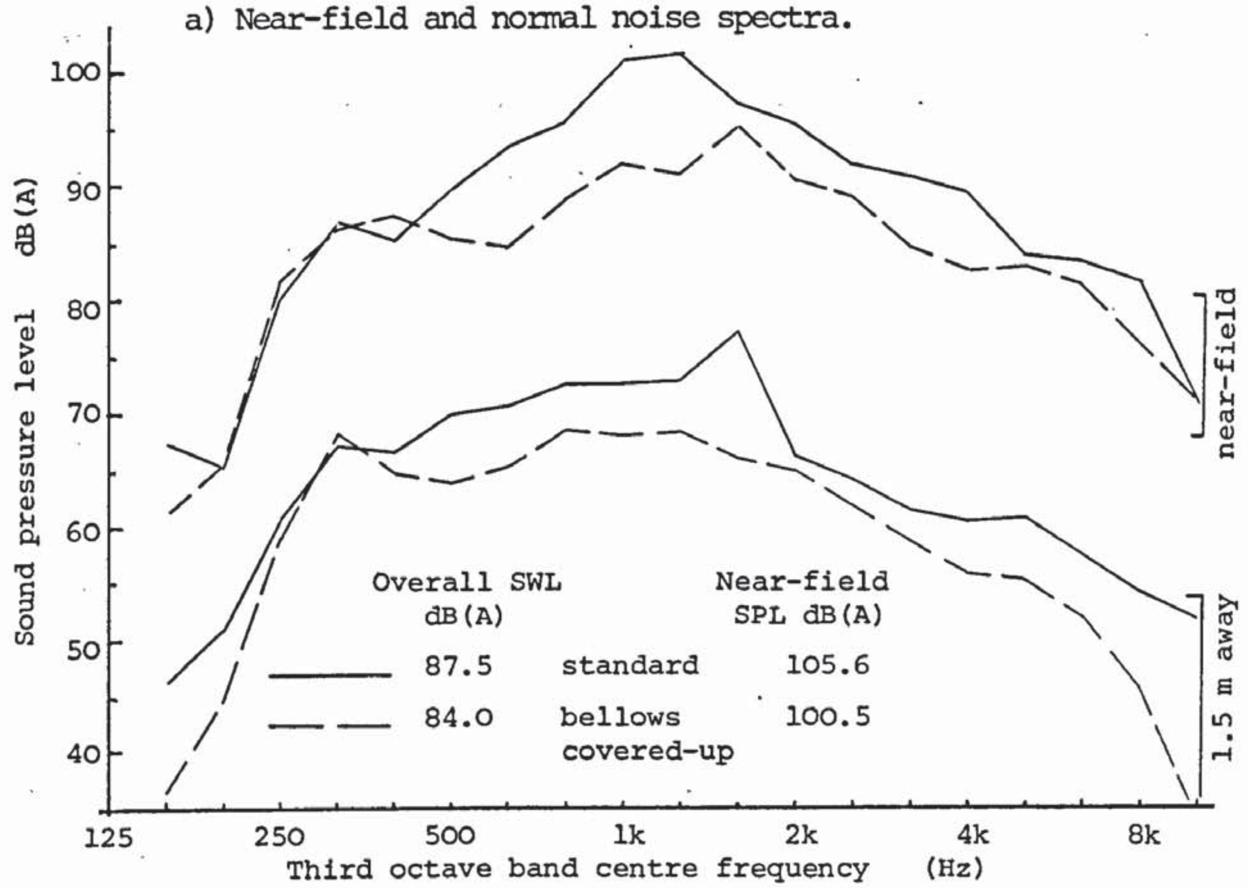
When the cleaner handle is lowered down to the floor (the position for cleaning under the furniture), the corrugated bellows stands proud of the surface of the lower cover plate. The following modifications were made to prevent the corrugations protruding beyond the cover plate,

- the output duct of the fan chamber was made shallow and broad
- the inlet end of the bellows was also made shallow and broad to fit the modified fan chamber output
- the motor was rotated to enhance lowering of the bellows, by
  - eliminating the "boss" at the joint between the fan chamber and the motor body, and
  - modifying the recess in the chassis for bellows.

A plastic sheet of 2mm thickness was fabricated and fitted. The near-field and the overall cleaner noise measurements for before and after cases are plotted in Figure 5.70a. By covering the bellows there was a 5 dB(A) reduction in the near-field sound level and a 3.5 dB(A) reduction in the overall cleaner noise. A very significant reduction in noise. The results showed that the benefit was of broad-band frequency, 500-8000 Hz.

Empirical prediction

Using the information contained in Figure 14 of Appendix 4, and that according to Guest (1986) the noise STL characteristics of glass are similar to those of plastics, the critical frequency was determined as the



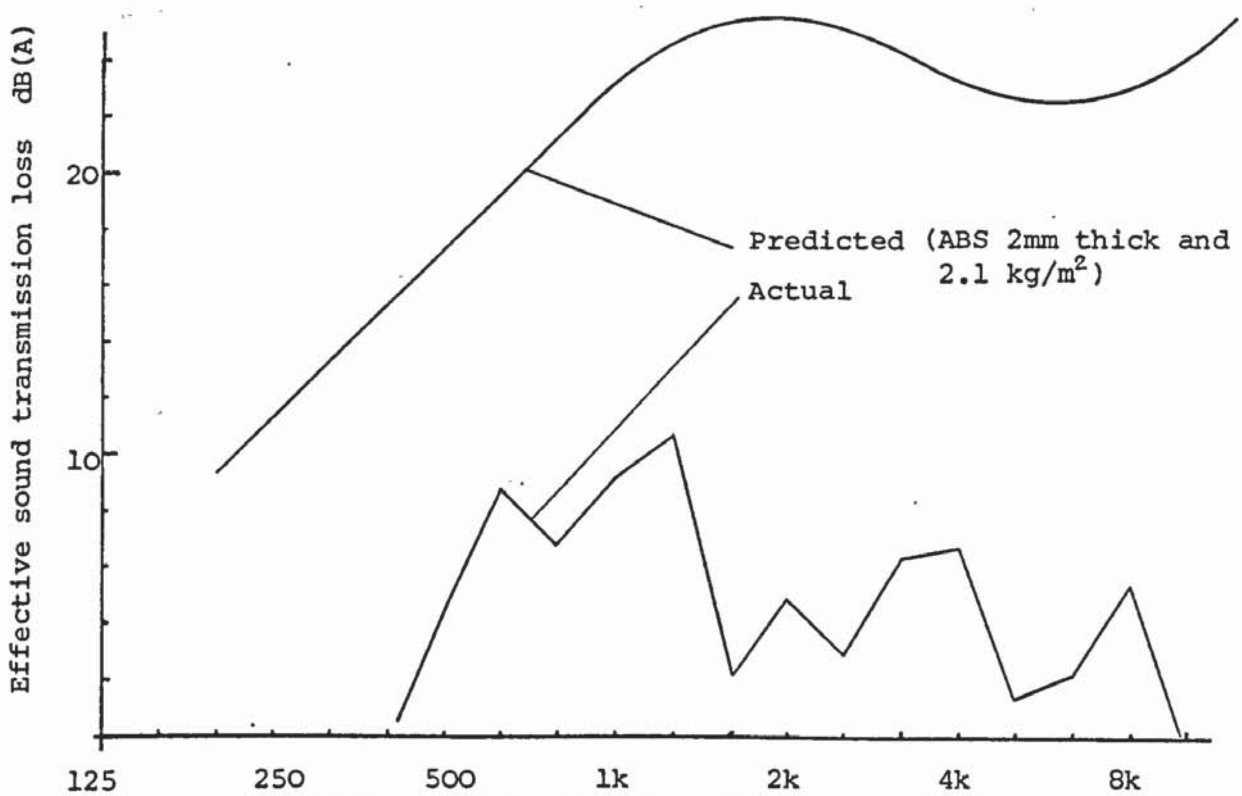


Figure 5.70 a) Sound spectra of cleaner and b) effective sound transmission loss due to cover-up of bellows.

intersection of the mass law curve and STL of 24 dB. In deriving the STL curve, the panel was assumed to be heavy, limp and large and the sound field was assumed to be reverberant on the source side. Although these assumptions do not hold well for a small plastic sheet tightly covering a relatively large sound source, it was nevertheless hoped that a prediction could be used as a guide.

There was a lot of difference between the predicted and the actual noise reduction for a 2mm plastic sheet, as shown in Figure 5.70b. This can be explained, not just by the doubt in the assumptions highlighted above but also, and more significantly, because of the contribution from other nearby noise sources eg. the orifices in the hood and the lower end of bellows. Furthermore, the incident noise was not of random frequency. In the absence of these factors the correlation would have been better.

5.8.14 Noise radiating from orifices in the hood (Figures 5.71-5.72)

The noise radiating from the orifices in the hood was the second worst noise source after the bellows. The noise levels measured above the air vents and the nozzle height adjustment slot were about 5.5 dB(A) greater than the value at centre of hood, away from any orifice.

The effective STL of the hood

The sound transmission loss due to the hood was determined by near-field noise measurement with and without the hood. These measurements were made after quietening the bellows area. The results are given below:

	Cleaner noise SWL dB(A)	Near-field SPL dB(A)
Above centre of hood	81.5	88.2
same point without hood	87.8	100.5

The noise attenuation due to the hood was broad-band as shown in

Figure 5.71a. The difference in the near-field sound level for without and with the hood is equivalent to its sound transmission loss. This is shown plotted in Figure 5.71b. The figure shows the predicted STL for 2mm sheet of ABS. The results show a much better correlation. The small difference is due to the contribution of noise from other sources, and maybe the thickness of the hood at the measurement point was less than 2.0 mm. Notice that a reduction of 12.3 dB(A) in the vicinity of the hood resulted in a 6.3 dB(A) drop in overall noise level.

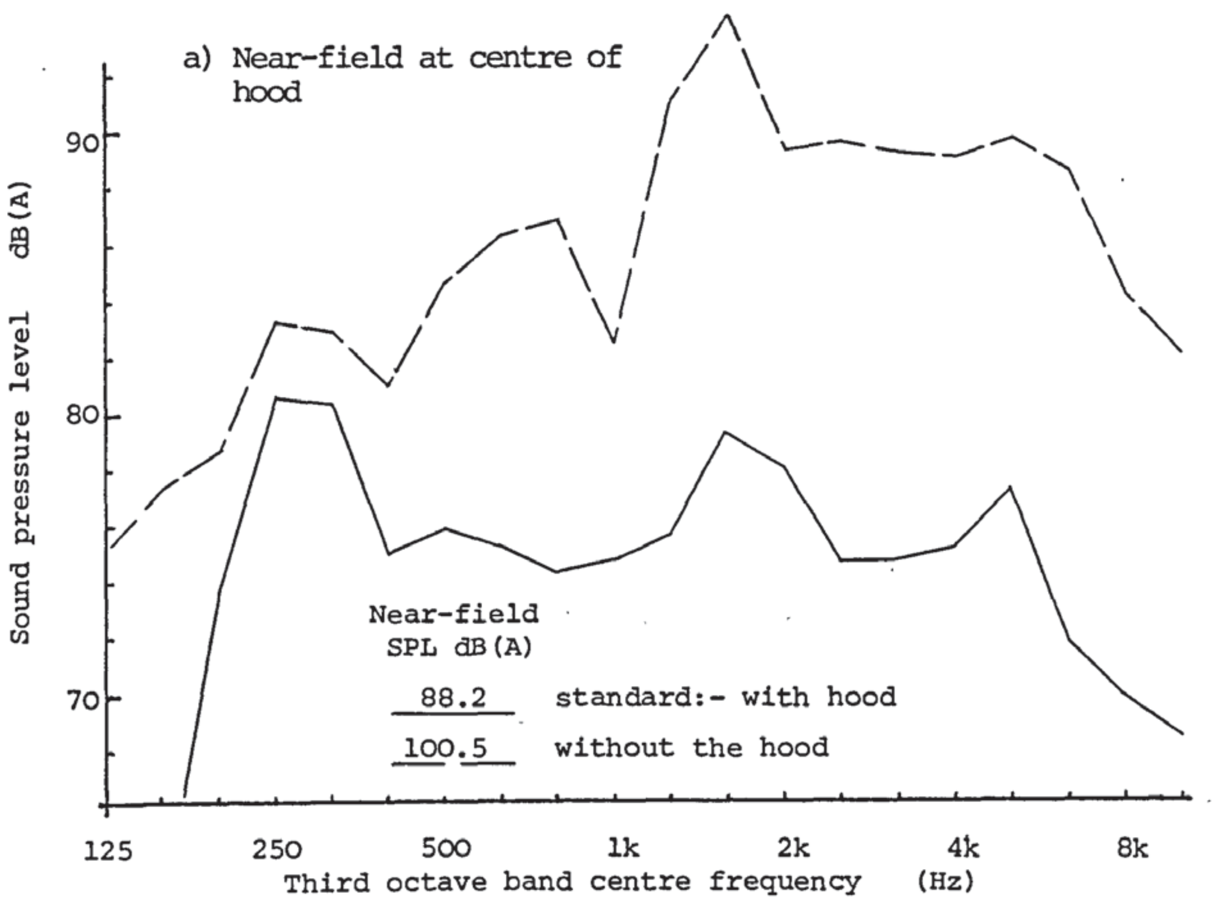
The effect of the orifices in the hood

The near-field measurements showed that the transmission loss is greatly impaired by the presence of the orifices. The aim of the following test was to investigate the potential noise attenuation by sealing or blocking the orifices.

The results, plotted in Figure 5.72, showed very little improvement in the overall cleaner noise by sealing or blocking the orifices. These results were for a standard cleaner prior to the cover-up of the bellows and it was thought this may have influenced the measurements. Under these circumstances, even a small reduction in the overall noise due to a sheet of sponge would be considered a success, because in the absence of such contribution the reduction in noise would be appreciable. The modification therefore, will be considered on the "quiet" cleaner.

5.8.15 Noise radiating between the chassis and floor (Figure 5.73)

Near-field sound measurements indicated the gap between the cleaner chassis and the floor as being a significant noise source. It was third in magnitude after the noise levels close to the bellows and the orifices. At the rear of the cleaner, a group of components are joined and due to the production tolerances and allowances for access, a number of gaps are present. Consequently, there are higher noise levels at the bottom and the rear of bellows and just in front of the rear wheels. In contrast, all of



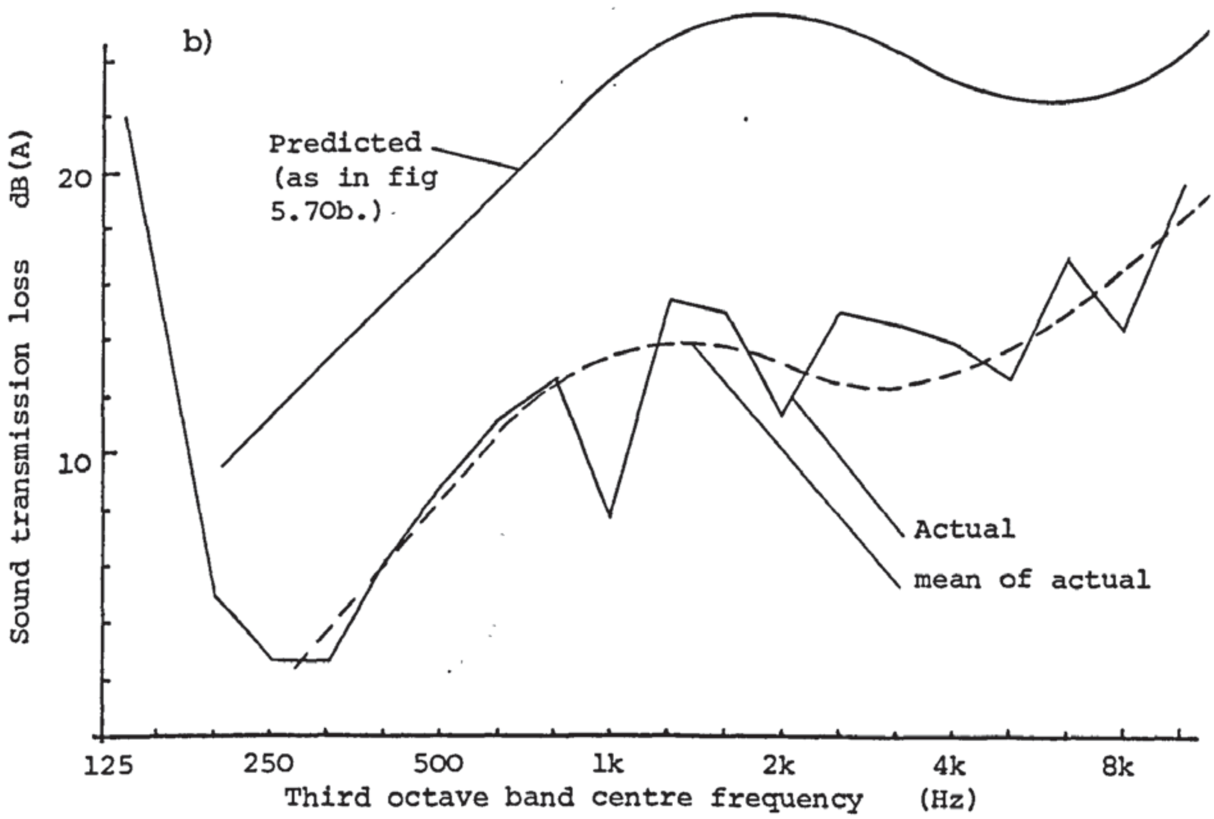


Figure 5.71 a) Near-field sound spectra with and without the hood
b) the effective sound transmission loss for the hood.

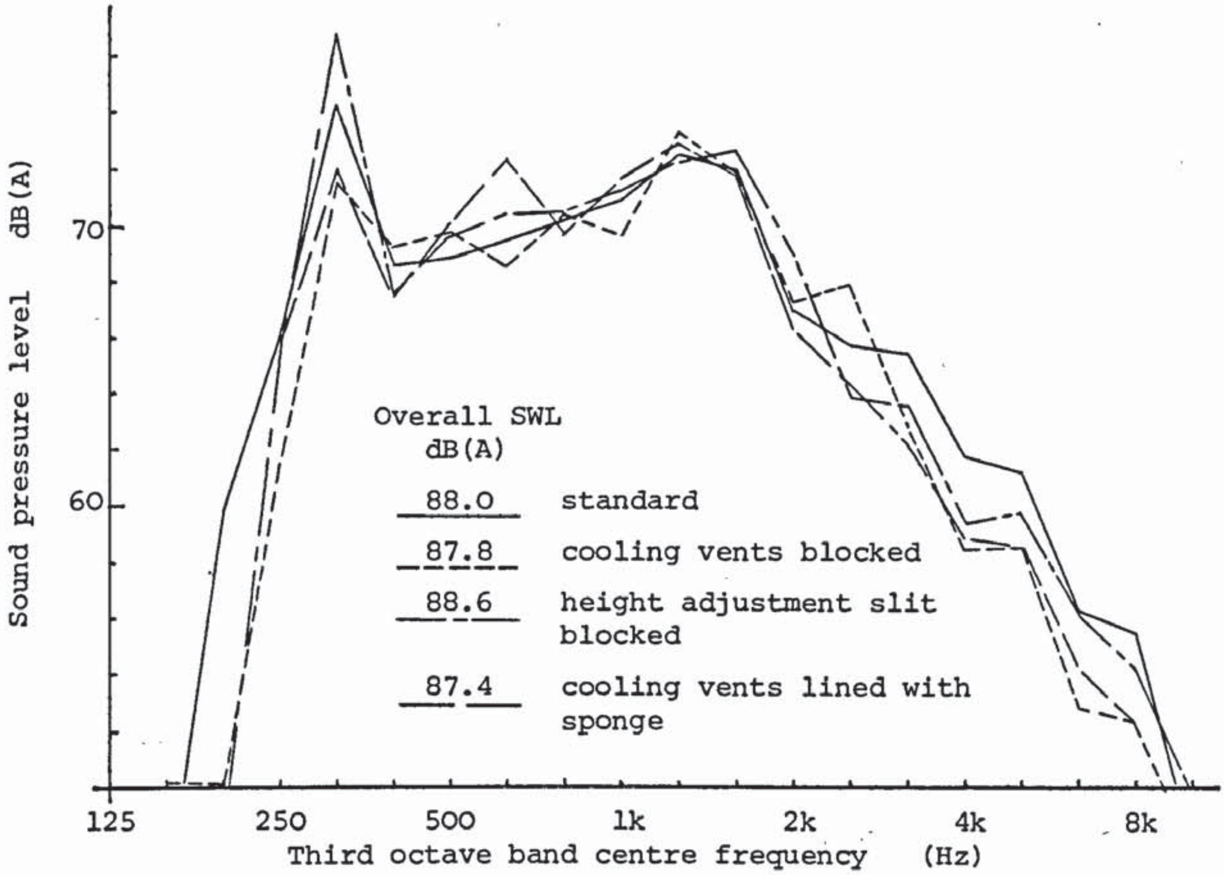


Figure 5.72 Effect of modifying orifices in the hood.

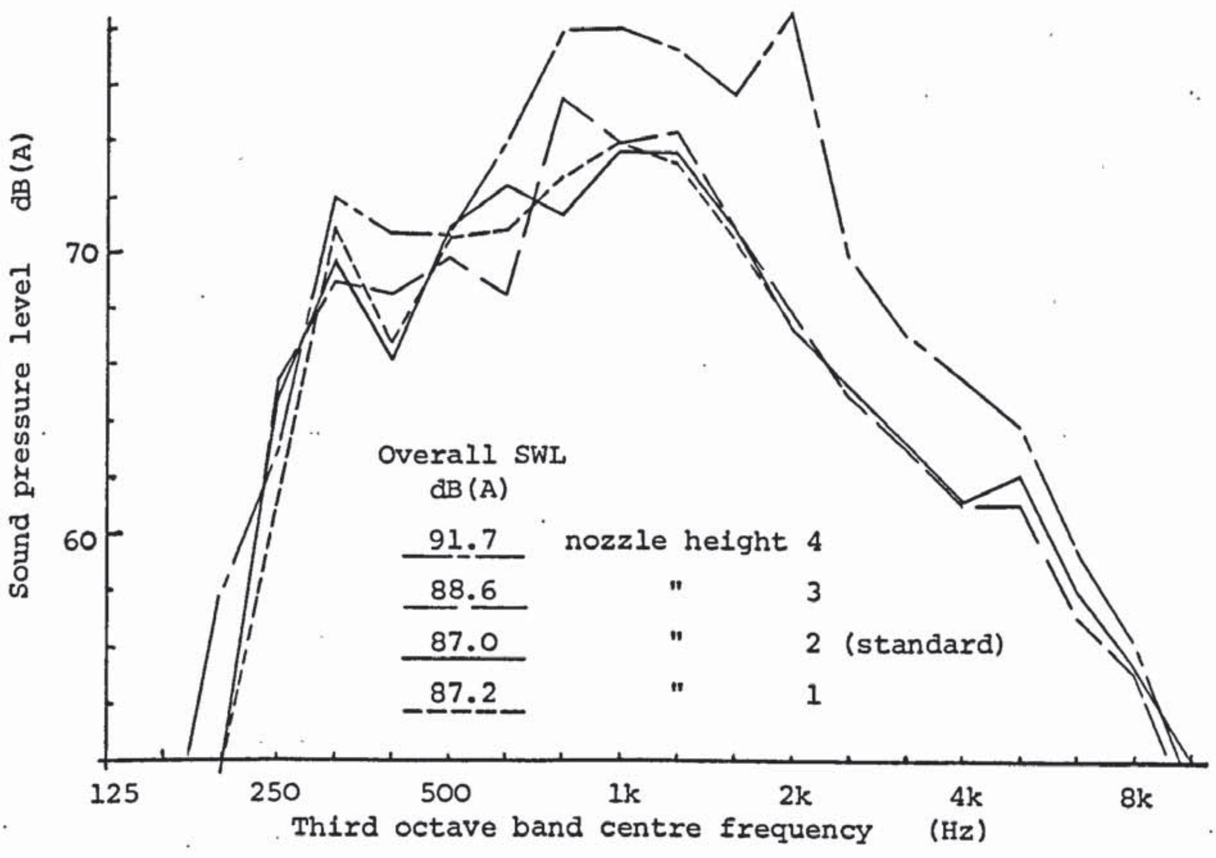


Figure 5.73 Cleaner noise spectra at the four nozzle height positions.

the joints at the front are sealed by the furniture guard.

At the front of the cleaner

The noise level at each of the nozzle height settings is summarised below:

Nozzle height setting	Cleaner noise SWL dB(A)	Nozzle height gap mm.
1 (low)	87.2	3.0
2 standard	87.0	3.8
3	88.6	8.5
4 (high)	91.7	16.0

The results showed that as the cleaner nozzle was raised there was an increase in the cleaner noise level. As shown in Figure 5.73, the increase was broad-band and therefore, most likely due to an increase in the air-flow turbulence. The noise source also becomes increasingly exposed and there is no barrier to attenuate the radiated noise. This led to the concept of an adjustable furniture guard. The gap between the bottom edge of the modified furniture guard and the carpet would be independent of the nozzle height. As a result, the noise at the higher nozzle setting was attenuated or absorbed by the guard and this led to significant noise reduction. Cleaner noise levels were as follows,

Nozzle height setting With adjustable guard:	Cleaner noise SWL dB(A)
1 (low)	87.1
2 (standard)	87.2
3	87.6
4 (high)	88.4

At the rear of the cleaner

In contrast to the front, there was no air-flow around the rear of the cleaner. Therefore, the noise at the rear is that which has radiated through the bellows and through any gaps in the chassis. There was no space to incorporate any substantial modification so the mating edges of the chassis and the hood were sealed using a soft rubber tube, and an aperture in the chassis for the air outlet/bellows was sealed using a rubber gaiter.

There was a reduction of about 1.3 dB(A) in the near-field sound level but hardly any difference in the overall cleaner noise level. It was thought that the gaiter material, being 1.5mm thick rubber sheet similar to the bellows, was not acoustically efficient.

5.8.16 Isolation of motor vibrations (Figures 5.74-5.77)

The object was to reduce the the transmission of vibration from the motor to the rest of the structure. Hence,

1) for immediate application to the Turbopower cleaner; mounts of different properties were evaluated, and
2) for the long term; a theoretical model was arranged and then used to optimise rubber mount design by utilising measured data.

5.8.16.1 Evaluation of Norsorex rubber material

Details of standard rubber mounts.

To improve the rubber mounts, properties of the existing mounts have to be determined. The small mount was designed to be much harder than the large one because of the pull of the belt. Indeed, according to the shore hardness measured by a portable gauge, the above was true. The shore hardness was measured at the room temperature, and the values were 34 for the large mount and 51 for the small.

The motor was mounted in the cleaner chassis with two rubber mounts as normal. The experimental force vs deflection curves were obtained for each mount installed in-situ. The results, in Figure 5.74a, showed the large mount to be stiffer than the small mount, a rate of 251 N/mm compared with 151 N/mm. Therefore, although the small mount was specified to be hard, this mount was softer of the two in-situ. The results corroborated the writer's opinion that the mounts were poorly designed, because,

- the large mount has very little room to accommodate "flowage" of rubber due to compression
- the damping characteristics of rubber were overlooked
- under compression, the small mount becomes very hard, thus hindering the isolation of vibration.

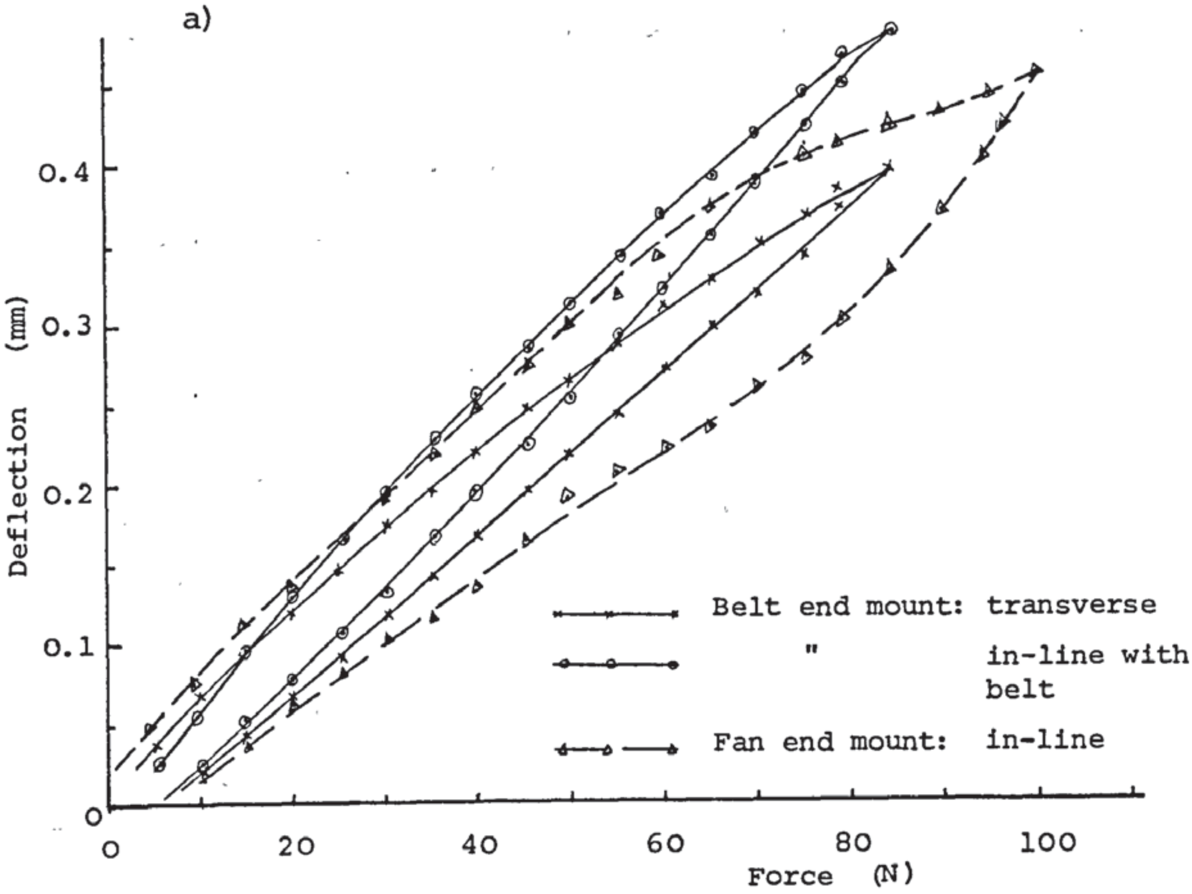
Rubber in shear undergoes greater deflection than in compression. The large mount was therefore, drilled to create open areas to accommodate rubber flow due to the shear action and to "soften" the mount in-situ. The modification resulted in a reduction of 1.5 to 2 dB(A) in the overall noise level. This set the scene for the investigation of soft mounts.

<u>Norsorex rubber compound</u>

An alternative material was required for low stiffness and higher damping. Norsorex polynorbornene (trade-mark of CdF Chimie) was one such material. It has the capacity to hold large amount of oils and fillers which directly control the stiffness and the damping properties. Four different compounds were formulated with shore hardness in the range 26 to 44. The force vs deflection curves were measured in-situ for both the Norsorex and the standard large mount. The result is shown in Figure 5.74b and showed that Norsorex was about a third of the hardness of the standard material.

<u>Cleaner noise tests using Norsorex mounts</u>

Rather than assessing the effect on the overall noise for a pair of



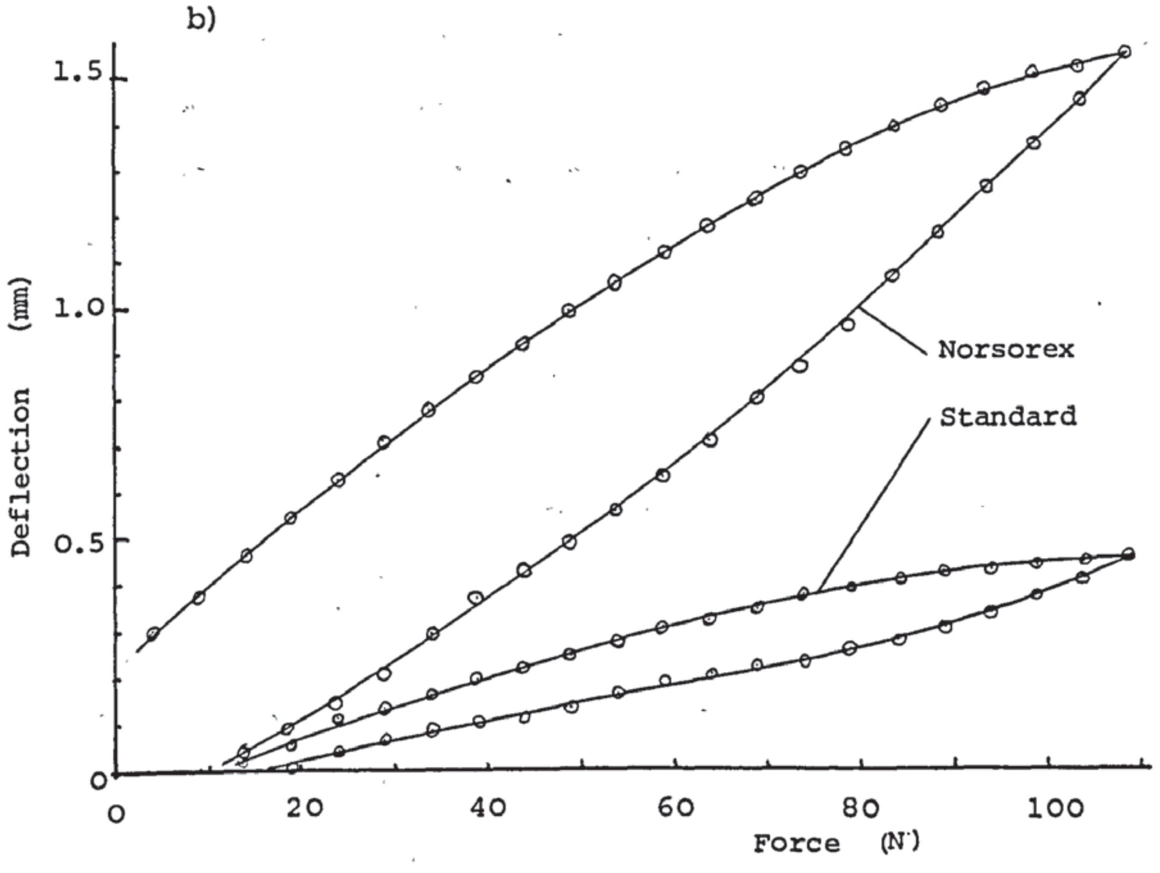


Figure 5.74 Force vs. delection for a) standard mounts and b) Norsorex.

matching mounts, a 5 by 5 matrix of the mounts available was created, as shown in the table below. The sound level for each combination was obtained with the objective of discovering whether one plane was more critical than the other. The cleaner noise levels are presented below.

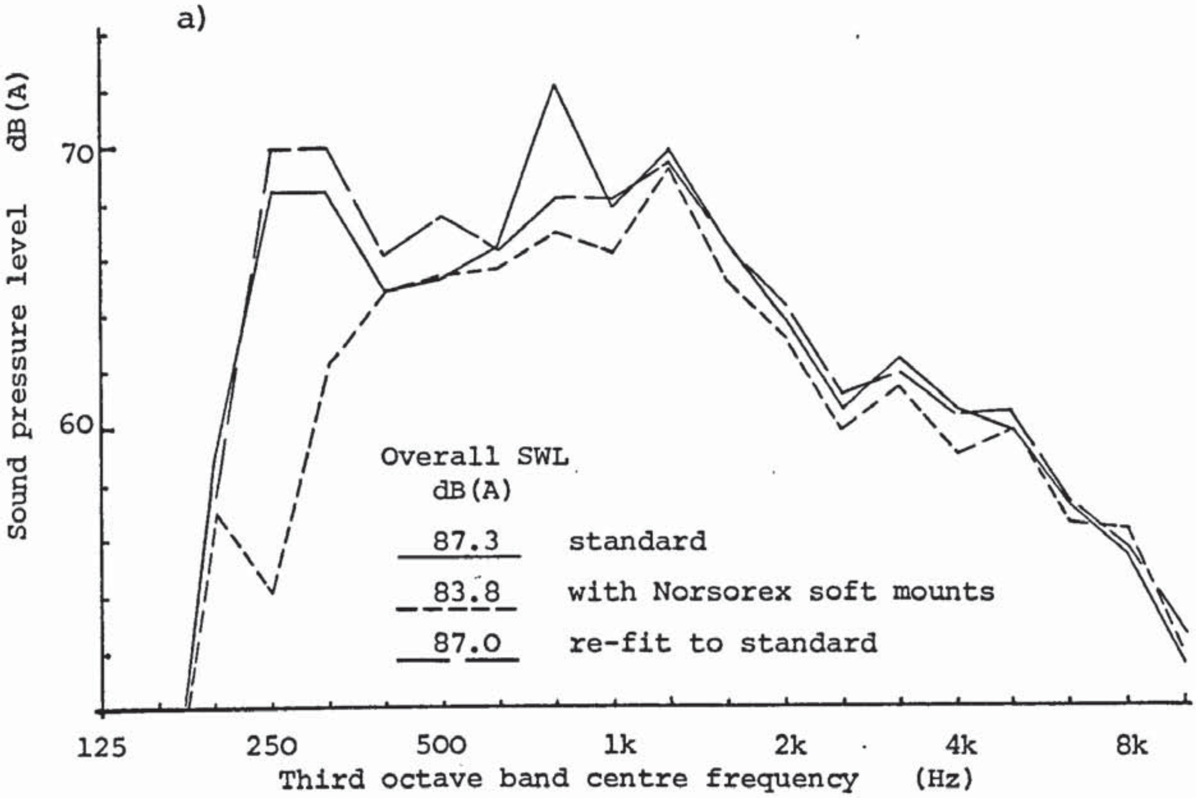
		Small belt end mount Shore Hardness				
		26	31	35	43	51*
Large	26	76.8	77.1	76.4	76.3	79.2
fan end	31	76.1	77.1	76.3	77.9	77.6
mount	34*	77.6	77.4	77.6	77.7	80.3
Shore	38	77.7	77.5	77.2	77.3	77.8
Hardness	44	77.5	78.0	77.4	77.6	77.5

SPL in dB(A)

[* = standard mounts]

The results showed consistent sound levels for the Norsorex mounts. On average there was a reduction of 2.5-3.5 dB(A) in the noise level. According to CdF Chimie, the Norsorex rubber compound had higher damping than the rubber used at present. Therefore, the damping was significant in reducing the transmission of motor vibration to the chassis and therefore, a reduction in the noise level. The soft mounts were extremely beneficial in some cases. The belt-end-mount of SH 35 gave a very "pleasant sound" for the cleaner. Third octave sound levels for Norsorex and the standard mounts are plotted in Figure 5.75a and show a large attenuation at low frequency.

The Norsorex trial was conducted on a Turbopower model U2198 which had a large imbalance peak which resulted in a 6.0 dB(A) rise in the cleaner noise level. The results are plotted in Figure 5.75b. Notice that with the soft Norsorex mounts, the third octave bands at 250 and 315 Hz were attenuated by 10dB. The overall sound level was reduced from 91.3 to



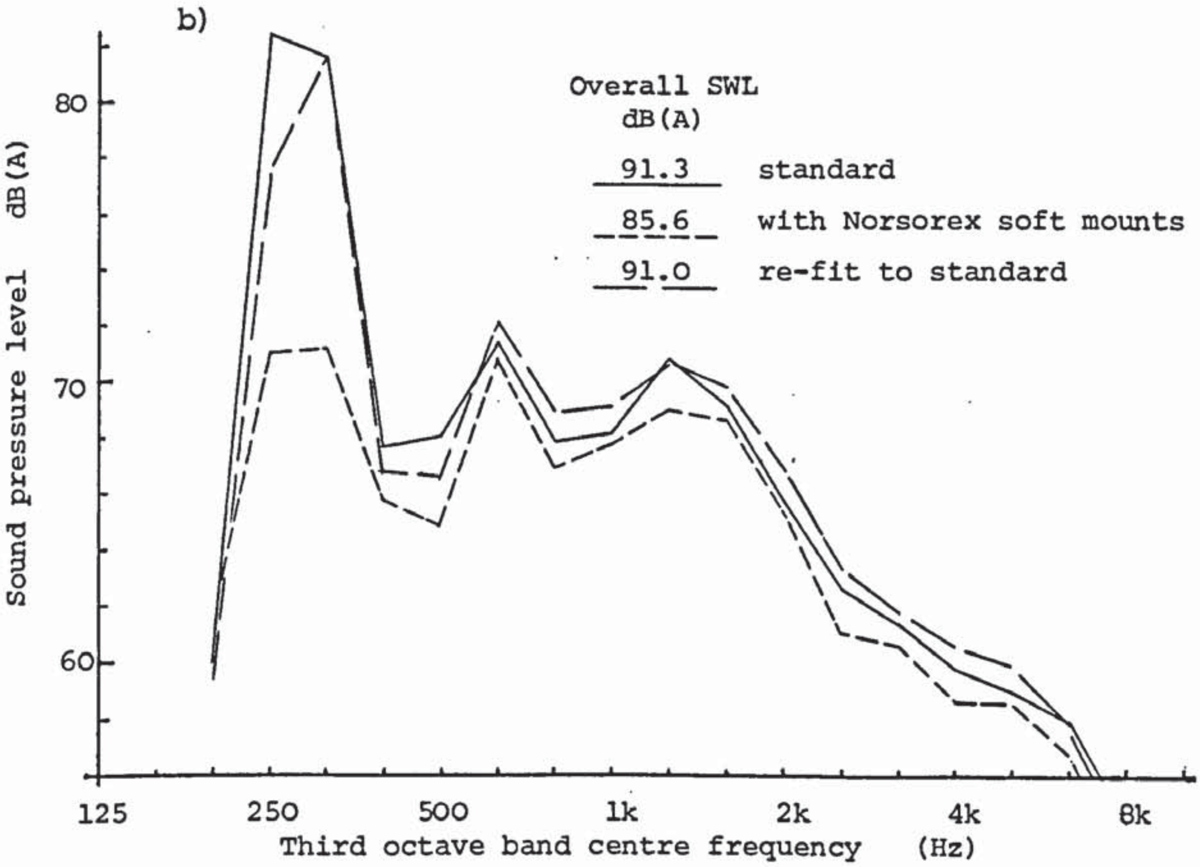


Figure 5.75 The benefit of using soft Norsorex mounts on a) U 2194 and b) U 2198 cleaners.

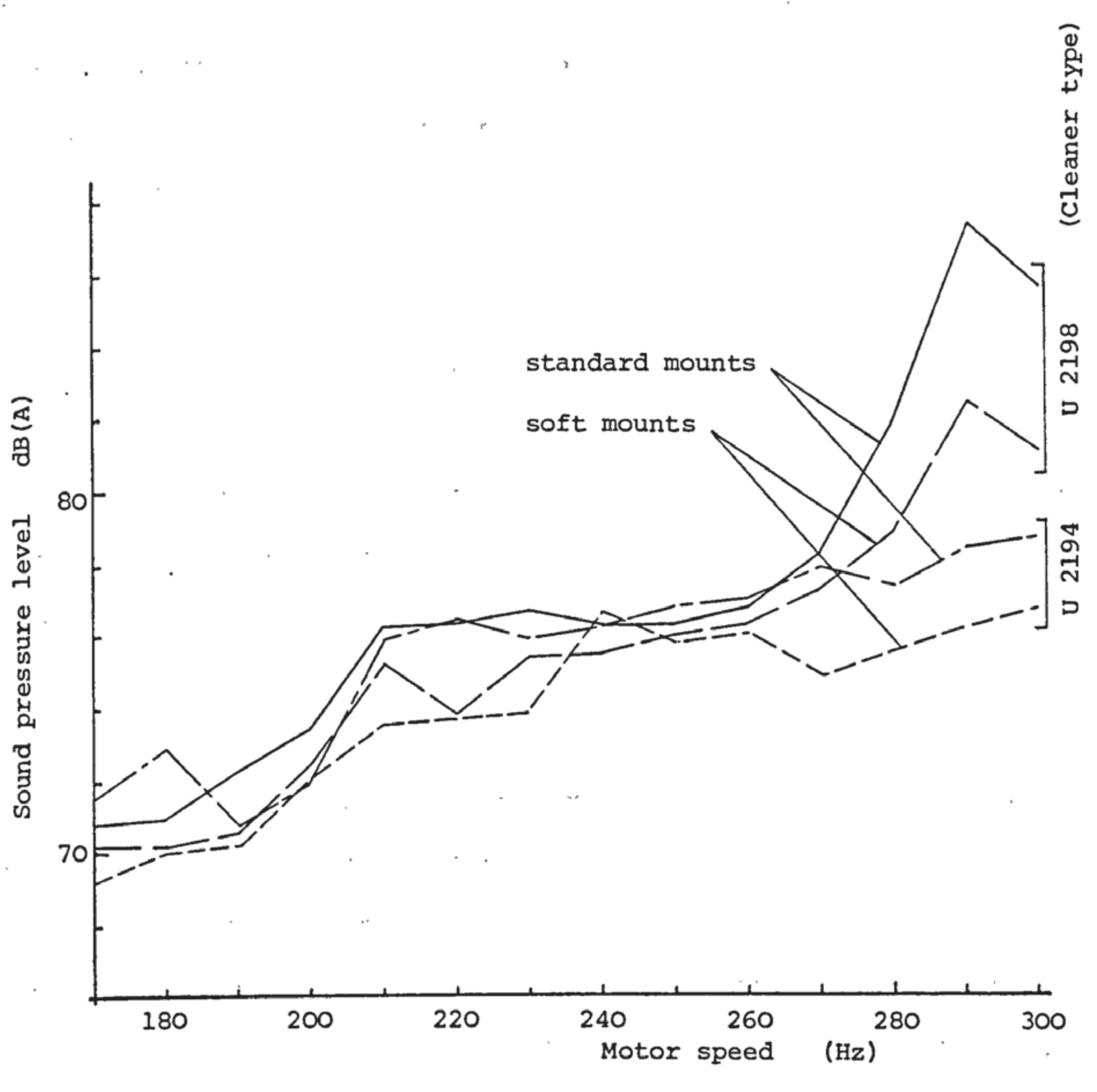


Figure 5.75 The benefit of using soft Norsorex mounts
c) cleaner SPL vs. motor speed

85.6 dB(A).

The variation in the sound pressure level with the motor speed, for both U2194 and U2198 cleaners in standard and modified form, is given in Figure 5.75c. The results showed that noise reduction is greater at higher speeds. The use of Norsorex mounts would result in a refined cleaner because the noise level of a freak cleaner, as a result of high imbalance, will be proportionally attenuated.

5.8.16.2 A theoretical model of the motor on chassis

The object of this model was to provide a better understanding of the dynamics of the motor/chassis structure, and highlight at an early stage any strong resonances. A theoretical model of the motor on a chassis was approached in two ways,

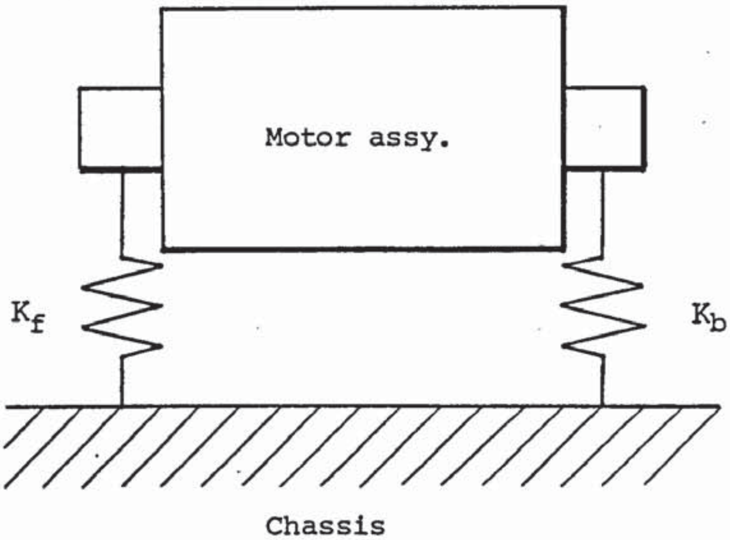
- the rigid-body motion of the motor supported by springs on a chassis, and
- a sub-system structure using mobility concepts.

These approaches are vastly different. The static data of the motor and the chassis is used for the rigid-body model whereas the measured dynamic response for the motor, the support and the chassis are used for the sub-system model. This implies the former can be used as a quick reference model while the second method, because it is complicated and requires special intrumentation, be regarded as the final optimisation model.

Rigid-body model

The transmission of vibration is generally reduced by introducing a soft support between the machine and its foundation. The support has to be designed so that the natural frequency of the machine on the support would be several times, nominally three or four, lower than the main excitation frequency. The cleaner motor, its support and the chassis can be idealised as shown in Figure 5.76.

Assuming the chassis is rigid, the resonant frequencies of the



K_f - fan side mount stiffness
K_b - belt side mount stiffness

Figure 5.76 A rigid body representation of the motor in the chassis.

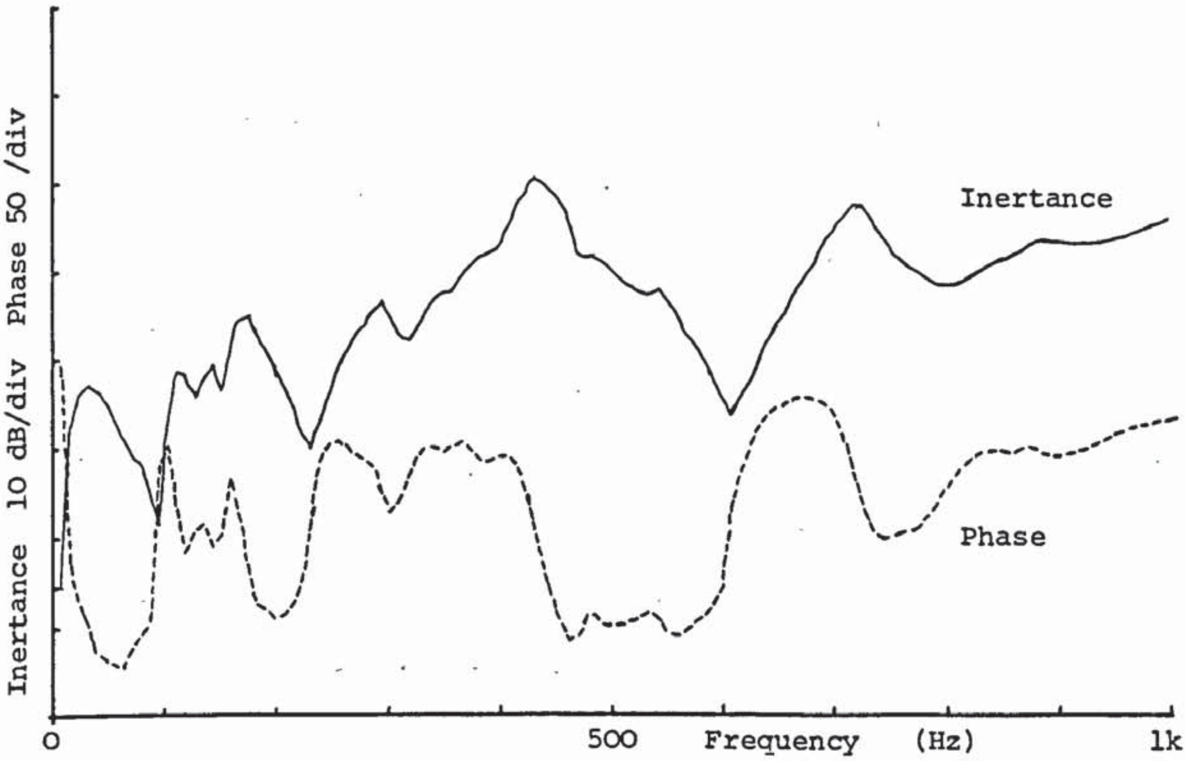


Figure 5.77 Frequency response of motor in the chassis. (measured).

motor can be predicted. A rigid-body has six degrees of freedom and therefore six natural frequencies, but in our case the most important are the bounce and pitch. The pull of the belt results in an increase in the mount stiffness in the plane of the belt. There are therefore, two cases to consider: in-line with and transverse to the belt. Derivation of the model is presented in Appendix 3. The predicted natural frequencies are summarised below.

	The plane with respect to the agitator belt axis: Transverse	In-line
Bounce	484 Hz	428 Hz
	617 Hz	696 Hz
Pitch	528 Hz	492 Hz
	1187 Hz	1310 Hz

The results can be verified by the vibration response of the motor assembly installed in the chassis. The motor assembly was excited at the belt-end mounting via a small hole cut in the rubber mount and the chassis for access. The peaks in response are shown in Figure 5.77 at 112 Hz, 144 Hz, 171 Hz, 296 Hz and 431 Hz. The discrepancy in the predicted and the measured resonant frequencies is probably due to the chassis being flexible and light, contrary to the above assumption. The figure also shows an "active region" around the normal motor speed of 285 Hz. This corroborates the finding that the cleaner noise increases for only a small rise in the motor speed. Therefore, a reduction in the motor speed would be advantegeous.

<u>Sub-system model</u>

This approach is necessary when the stiffness of the foundation is not infinite and its mass is comparable or even less than the machine. The dynamic properties are then taken into consideration. Details of the theoretical model, derivation of isolator insertion loss and development of a computer program "SYSMO" to run on a HP 9845B desk top computer are presented in Appendix 3. The frequency response functions were measured on the HP 3582A analyser and conditioned and stored on disc using the HP 9825A. Data manipulation using SYSMO was to be performed on the HP 9845B. However, because of the hardware problems with the HP 9825A and HP 9845B system, it was not possible to verify the theoretical model.

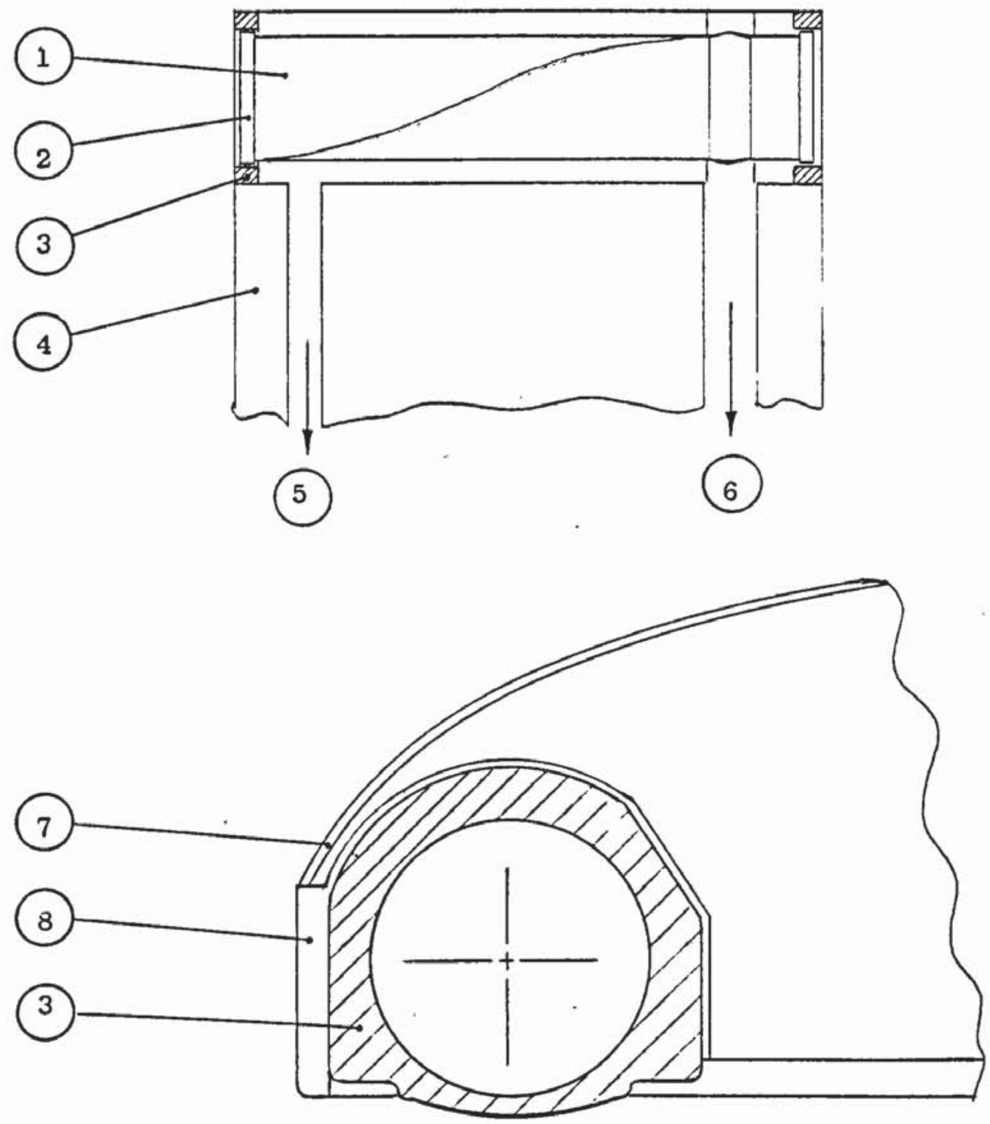
The object was to design an isolator such that its insertion loss was high across a broad frequency range, and especially at the frequencies corresponding to the active region of the motor.

<u>5.8.17 Isolation of the agitator vibration</u> (Figures 5.78-5.79)

At present, the agitator is installed via the hub of the thread guards which fit directly into the slots in the chassis. There is therefore, a direct vibration path. A new method of assembly was conceived to break this direct contact and hence reduce the vibration transmitted to the chassis. The hubs at the agitator ends were isolated from the chassis as follows;

- each thread guard hub was cut and filed so as to be flush with the external face, and
- a new housing was fabricated from ABS plastic as shown in Figure 5.78. The hole was large enough to accommodate the thread guard with a 3.5mm thick ring of rubber around its circumference.

On installation, it was ensured that the end faces of the guards were not touching the chassis. The results of the cleaner noise levels for with and without this modification are plotted in Figure 5.79. Notice a noise reduction of 2 dB(A) by isolating the agitator-to-chassis vibration



1 Agitator
2 Agitator Thread Guards
3 Thread Housing
4 Cleaner Chassis
5 Air Inlet To Motor
6 Belt To Motor
7 Cleaner Hood
8 Furniture Guard

Figure 5.78 Details of modifications to the agitator thread guard housings.

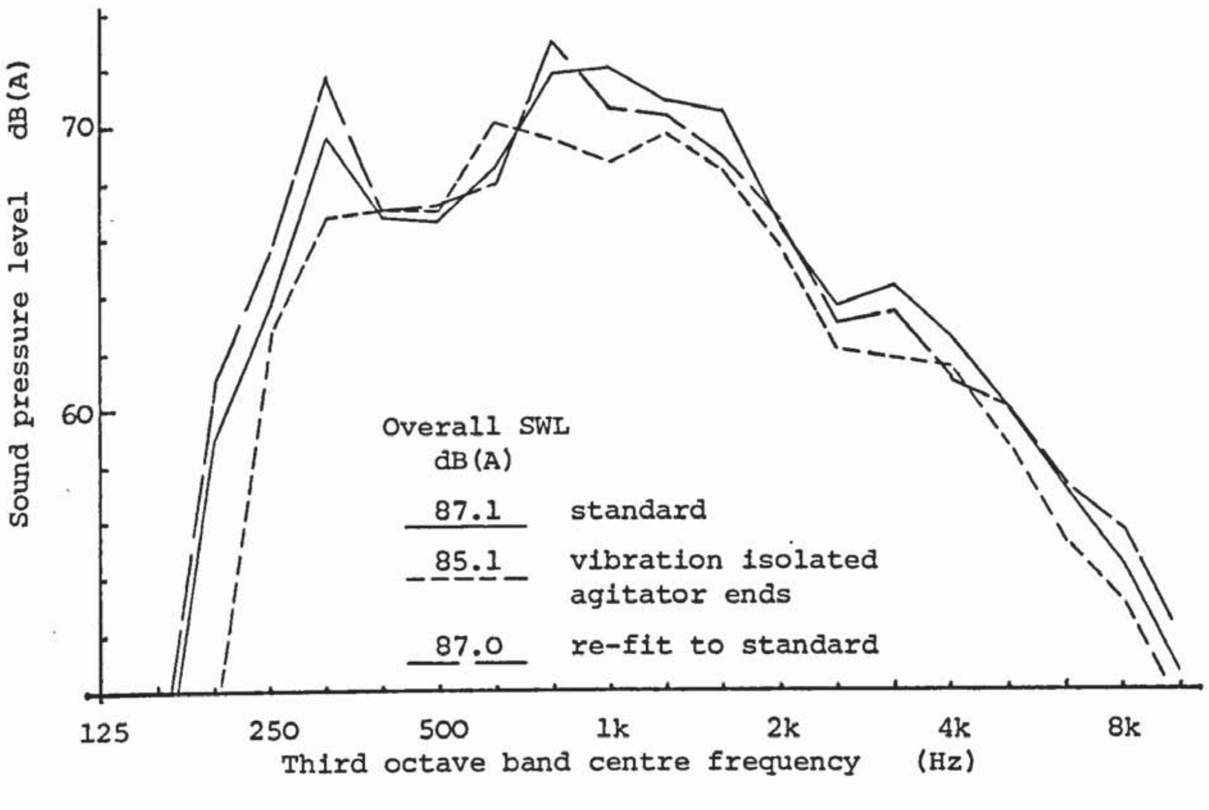


Figure 5.79 Noise spectra for vibration isolated agitator thread guards.

path. The reduction in the noise is concentrated around 315 Hz and around 1000 Hz; there was also some reduction at higher frequencies. In conclusion, the reduction in the vibration transmitted from the agitator to the chassis gave particularly beneficial results. The overall noise level was reduced by 2 dB(A) and the greatest reduction was concentrated at the desired frequencies.

Observations have shown that if the agitator is tight between its thread guards, the bearings get hot and this eventually leads to a badly deformed chassis close to the agitator mountings. Consequently, the agitator becomes misaligned and subsequently, the belt slips off the motor spindle. A further advantage of the modification is that it provides a cure for this problem.

5.9 Review

This chapter has documented the experimental results of the correlation of the acoustic room and the calibration of sound, vibration and performance measurements. After identifying and ranking the major sources of noise, noise reduction principles and treatments were applied and results were presented. Some of the treatments were beneficial while others were not. The selection and the evaluation of positive treatments with regards to their cost-effectiveness and practicality for application to the Turbopower cleaner will be presented in the next chapter.

CHAPTER SIX: CONCEPT OF A QUIET TURBOPOWER CLEANER.

Contents

6.1 Introduction

In Chapter 5 many ideas and practical modifications which resulted from the outlined hypotheses were investigated. Some of these proved to be potential solutions to the noise problem. This chapter examines these potential solutions and briefly assesses their merits for application to either the current cleaners or the next generation.

It is very difficult to appreciate the reduction in the noise of a vacuum cleaner by simply viewing graphs and results. Lower decibels do convey the obvious meaning; that the appliance is less noisy, but portray little or no subjective information of the sound. An exercise to promote the subjective results of this project, was through the construction of a "Quiet Cleaner". Brief details of the fabrication, the sound power and performance measurements and the subjective assessment of a prototype modified cleaner are now presented, and the chapter closes by considering the cost of the proposed modifications.

6.2 List of the solutions to the noise problem

In order to achieve a significant noise reduction, it is necessary to attack the noise problem at many points, not just one or two. The combined effort, therefore, included treatment of noise sources and modification to the vibration transmission paths and the radiating surfaces.

The noise treatments which provided a positive benefit to the Turbopower cleaner were,

- high production quality standards by improved inspection
- post-assembly balance of the motor
- reduction in the operating speed of the motor
- cover-up of the bellows area
- control of the exhaust air

- extended and moveable furniture guard
- isolation of the motor vibration
- isolation of the agitator vibration
- isolation of the hood from the chassis vibration
- prevention of wheel and wheel-trim rattle.

6.2.1 Evaluation of solutions

Product quality

At first sight, the Turbopower cleaner looks (and is) attractive, well made and contains all the major features within the model range. On closer srutiny however, the attention is drawn to badly fitted or poorly designed components such as the loose dust bag housing covers and wheel-trims. When operating the cleaner, the user detects the existence of stray vibration signals through the handle. The subjective assessment is heightened by "panel-rattle", because noticeable beats are produced.

The scatter in the sound power level of production cleaners was substantial. This spread, from 86 dB(A) to 94 dB(A), was primarily caused by the variation in the motor out-of-balance, as shown in the table below. The case of the extreme out-of-balance is a feasible situation because the imbalance of some of the suction fans produced in the cavity tool exceed the limit. Consequently, by maintaining the suction fan out-of-balance at its design limit, a significant reduction in the cleaner noise can be expected.

Armature imbalance:	Cleaner SWL dB(A)
best balanced	86.8
[average cleaner	87.5]
max design limit	90.4
twice the design limit	94.1

Post-assembly check on the out-of-balance of the armature

Currently the rotor is balanced while it comprises the cooling air fan, commutator, laminations and the windings, but without the suction fan, bearings and washers. At present there is no post-assembly check. The results of the experiments in Chapter 5 showed that imbalance at the cooling-fan-end bearing plane had the greatest influence on the overall cleaner noise. For maximum imbalance (within limit) at the vent-fan-end only (the suction fan plane being balanced) there was a noise increase of 5.2 dB(A), with respect to the best balanced armature. On the other hand, an increase of less than 2 dB(A) resulted when the imbalance was at the suction-fan-end plane alone.

Reduction in the motor speed

There was very little change in the cleaning efficiency for up to 3.5% reduction in the motor speed. A lower speed is associated with lower imbalance forces, which reduced the noise level of a cleaner by 2 dB(A). This was primarily as a result of the operating speed moving away from an active structural resonance in the cleaner.

Cover-up of the bellows.

The near-field sound measurements indicated the bellows area as the worst noise source; 5 dB(A) in excess of the second worst. By shielding the bellows with a sheet of ABS plastic, the near-field sound level was reduced by over 5 dB(A) and the cleaner sound power level was reduced by 3.5 dB(A). This is a significant reduction. Furthermore, it was obtained across broad-band frequency.

Control of exhaust air

The most practical method of reducing the air-flow turbulence noise was by sealing the peripheral gap between the dust bag housing covers, then the exhaust air was allowed to pass through two micro-filters and out via an opening at the bottom rear of the cleaner upright. Furthermore, the bag housings were strengthened by the seal and this consequently reduced their noise radiating capability at the important low frequencies. The reduction in noise was 2.1 dB(A).

Extended and moveable furniture guard

By constructing an adjustable furniture guard, a noise level of approximately 87dB(A) was maintained up to the third highest position of the nozzle. At the standard nozzle position, however, there was no advantage, so this modification will not be implemented in the "Quiet Cleaner".

The isolation of motor vibration

Imbalance of the motor is the source of all vibration. A reduction in the transmission of motor vibration, by using soft rubber mounts, was found to reduce the noise of the cleaner by up to 6 dB(A). The reduction in noise was proportional to the amount of the motor imbalance.

The isolation of agitator vibration

By preventing the direct transmission of vibration from the agitator to the chassis, a reduction of 2 dB(A) was obtained in the overall cleaner noise level. The modification also provided a solution for avoidance of chassis deformation close to the slots due to the heat generated by the agitator bearings.

The isolation of the hood from the chassis

The removal of the hood from some of the cleaners actually reduced the overall noise level. This highlights the present unsatisfactory design of the cleaner foot. Three approaches were investigated and conclusions were that,

1. by moving the hood mounting points away from the motor, the noise of the cleaner was reduced by 2 dB(A),
2. by avoiding the vibration transmission between the chassis and the hood using a sandwich of two rubber washers, a reduction in excess of 1.5 dB(A) was obtained,
3. the use of sponge or damping material led a to noise reduction

of 2-2.5 dB(A).

For each of the these modifications, there was greater noise reduction for cleaners whose motors had greater out-of-balance.

Prevention of wheel trim rattle

Some of the wheel trims were extremely loose and this gave rise to a noticeable "rattle". A tightly fitted trim resulted in a noise reduction of 1 dB(A). Nevertheless, the prevention of "rattle" was subjectively more significant.

6.2.2 Noise reduction trials at Hoover plc

Hoover plc's desire for a quieter range of products was at its highest at the time of the publication of findings of the market research survey in April 1980. Since then and during the course of this project, their desire has gradually declined. Consequently, the development work on noise reduction by Hoover plc during this project, was as follows:

i) a two month exercise to determine quick noise reducing solutions; this was an ad-hoc approach,

ii) trials on Norsorex rubber compound,

iii) a re-consideration of the vent-fan-end plane imbalance on the Turbopower,

and, iv) a noise reduction exercise on the new wet and dry cleaner.

The first item was undertaken by the writer in early 1985. The exercise was timed so that modifications could be evaluated and introduced with the facelift of the Turbopower cleaner which was targeted for September/October 1985. At that time no simple solutions were at hand. In the event, Hoover plc introduced a two speed facility which gave the user an option to lower the noise with a corresponding drop in the performance. The work on the wet and dry cleaners was undertaken by a development engineer within the Floorcare Lab. and will not be discussed here. The remaining items were in effect Hoover plc's response to the outcome of the present project.

Trials on a Norsorex rubber compound

A development project was undertaken to ascertain the usefulness of the Norsorex rubber material. Neill (1985), Chief Engineer Floorcare, suggested that, if Norsorex was successful, its introduction would enable Hoover plc to "satisfy the German requirement for less noisy Turbopower cleaners".

Four Turbopower cleaners were selected at random from the production line, checked to be manufactured to specification and then subjectively rated as regards noise level. The airborne noise was measured accurately for each "standard" cleaner at the NEL. The Norsorex mounts were then fitted and the noise re-measured. The results are shown in the table below. It was concluded from the exercise (by a development engineer at Cambuslang), that the average drop in the overall noise of the cleaner for Norsorex was 0.5 dB(A). However, the engineer did notice a consistent drop in the sound level of the octave band which contained the fundamental frequency. Unfortunately, for a reduction of 0.5-1 dB(A), the on-cost of the Norsorex material could not be justified.

	Cleaner SWL dB(A)	
Cleaner no.	Standard mount	Norsorex
1	86.99	86.34
2	86.43	86.19
3	87.03	86.45
4	87.12	86.71

The reason for the low reduction in noise was that the sample of cleaners was not "average". These cleaners had been pre-screened by the subjective noise test. A purely random sample however, would increase the likelihood of a freak cleaner, that is one comprising a higher imbalance. According to the experimental results, a greater noise reduction would have resulted from a random sample. So, perhaps the Norsorex mounts should be re-tested.

Re-consideration of the vent-fan-end plane imbalance.

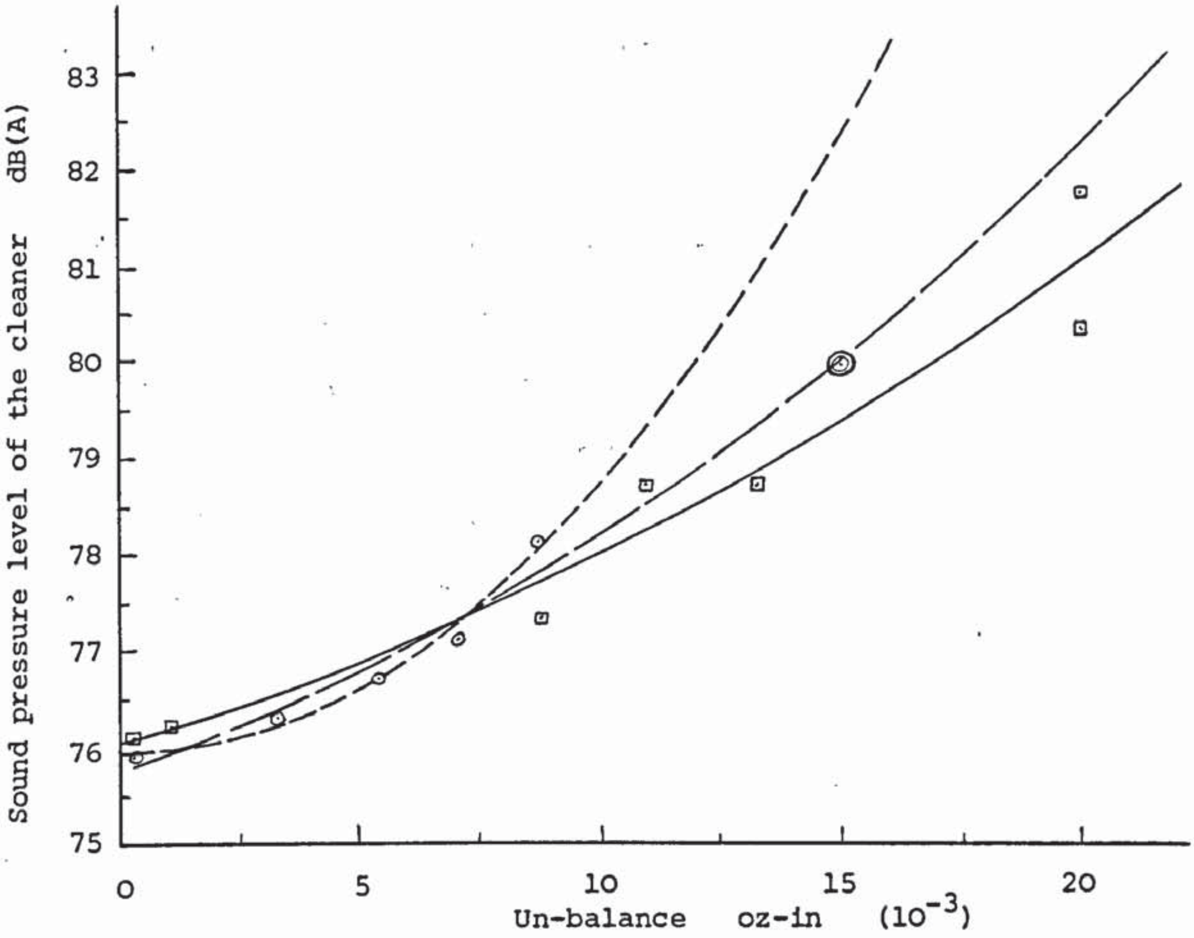
The notion that the armature out-of-balance affects the noise level of the cleaner is not new. The issue of the motor imbalance has been raised many times at Hoover plc. However, the experimental work on the armature imbalance was mainly concentrated at the suction-fan-end. The reasons were twofold: firstly, there was difficulty in holding the suction fan imbalance to the design limits because of production problems and the asymmetric nature of the fan and secondly, the apparent ease of holding the vent fan imbalance to the design limits. Novak (1986a) confirmed that

> "the overall cleaner noise is relatively insensitive to residual imbalance at the vent-fan-end plane".

This project proved that the vent-fan-end imbalance was critical to the overall noise of the cleaner. This finding aroused fresh interest in the subject of armature imbalance and following the release of this information, Novak (1986b) reflected on the change of opinion, as

> "the effect on overall noise of the vent-fan-end unbalance and the suction-fan-end unbalance seems to be roughly equivalent".

Analysis of the data which accompanied the second memorandum, given in Figure 6.1, however, confirmed the present author's findings. This figure shows 6 measured data points for the vent-fan-end imbalance. One point, at an imbalance of 0.015 oz in, is seen to be "out of step" compared to the other five. With or without the specified datum, the extrapolation of the curve fitted to the data indicates, for imbalance greater than the design tolerance, the vent fan end as having greater influence upon the noise of the cleaner than the suction fan end. By excluding the specified datum, the effect of the vent fan end imbalance is greatly increased.



▫	Suction fan end un-balance		————
⊙	Vent fan end un-balance	6 data points	—— ——
⊙	Vent fan end un-balance	5 data points excluding ◎ at 0.015 oz-in	— — — —

Figure 6.1 Effect of the suction and vent fan end plane un-balance on the cleaner noise.

6.3 The "Quiet Turbopower Cleaner"

Since, a change of 3 dB(A) is only just perceptible, the intention was to construct a "Quiet Cleaner" to demonstrate the full value of the findings of this project. The "Quiet Cleaner" was a standard Turbopower incorporating the noise reducing treatments.

6.3.1 Fabrication

The following features were implemented to creat a "Quiet cleaner",

- the armature was balanced to the design limits
- the bellows was covered up
- the hood was isolated using rubber washers
- soft Norsorex rubber mounts were used
- vibration isolation was applied to the agitator ends
- the exhaust air path was modified
- wheel-trims were tightened.

Details of the actual modifications that were made to the components to accommodate these solutions have already been presented in Chapter 5.

6.3.2 Evaluation of the "Quiet Cleaner"

The best method to evaluate a "Quiet Cleaner" was to repeat the sound measurements which were made to identify the noise sources and then compare these results with those of a standard cleaner. The following measurements were made,

i) the near-field noise levels

ii) the overall sound power level of the cleaner

iii) the suction/airflow performance and IEC 312 dust removal.

Near-field sound measurements

The table below illustrates the ranking order for noise sources of a standard cleaner (as presented in Chapter 5) with the corresponding results for the "Quiet Cleaner". There was a significant reduction in the noise level close to each source.

Noise source:	Near-field SPL in dB(A) Cleaner: standard	"Quiet"
Above the bellows	105.6	97.3
Hood: above orifices	100.5	93.8
Hood: away from orifices	94.8	88.2
Agitator bearing end	98.1	95.3

The overall sound power level

The average sound power level of a standard Turbopower hardbag cleaner is 87.5 dB(A) and the "Quiet Cleaner" before modification had a level of 87.3 dB(A). After five modifications, namely the cover-up of the bellows, the isolation of agitator ends and the hood, the use of Norsorex mounts and re-direction of the exhaust air, the noise level of the cleaner was reduced to 81.5 dB(A). When the motor speed was lowered to 275 Hz, the noise level of the "Quiet Cleaner" was reduced to 80.4 dB(A) and this reduction was broad-band. Third octave sound levels for lower motor speeds are plotted in Figure 6.2.

A comparison of the sound levels for the standard cleaner at 285 Hz and the "Quiet Cleaner" at 275 Hz are given in Figure 6.3. Notice the standard cleaner does not show a pronounced out-of-balance peak, which implies that its balance was better than average, but despite this its noise level was 87.3 dB(A) SWL. The results show that the "Quiet Cleaner" exhibits noise reduction at all frequencies.

As mentioned in Chapter 5, the sound measured at the single-point in the Perivale acoustic room was likely to be influenced by the characteristics of the acoustic room. Consequently, for a fair comparison, the sound levels on all sides of the cleaner were measured. The results

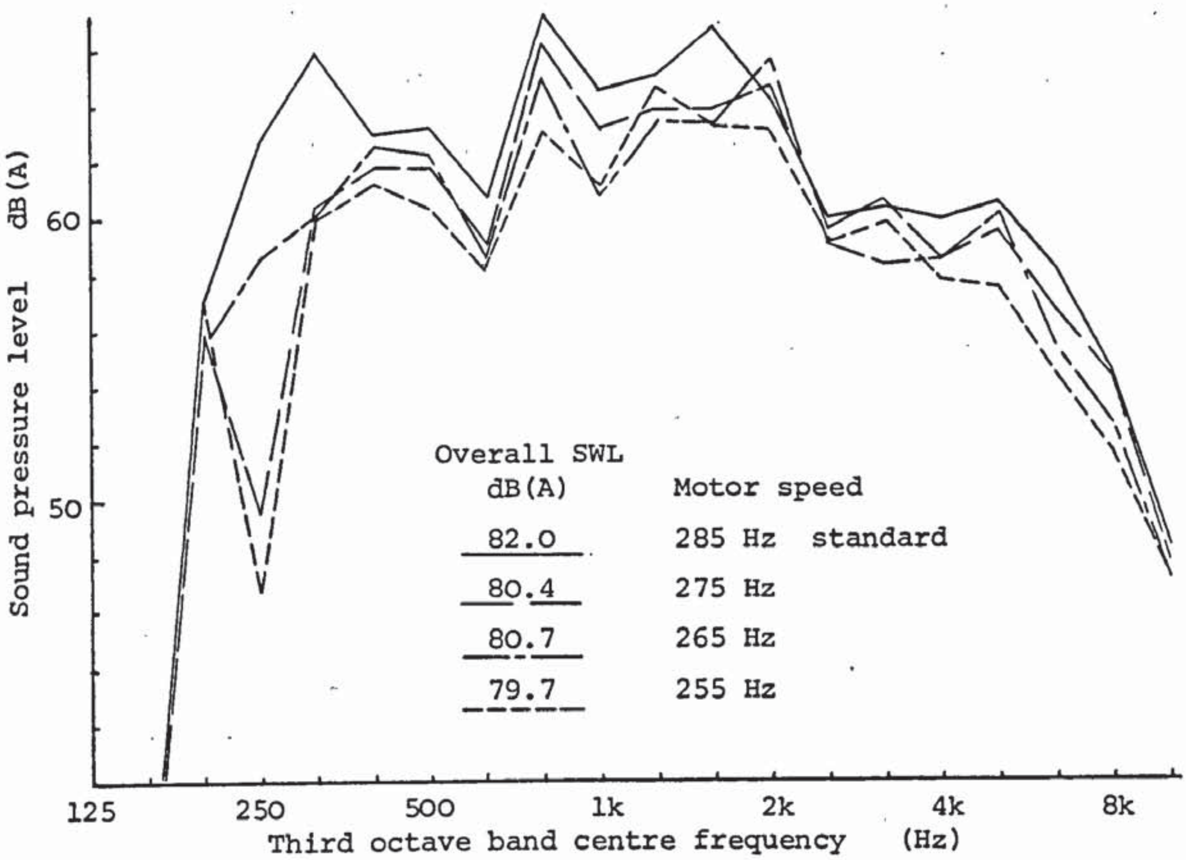


Figure 6.2 Noise spectra for the quiet cleaner at different motor speeds.

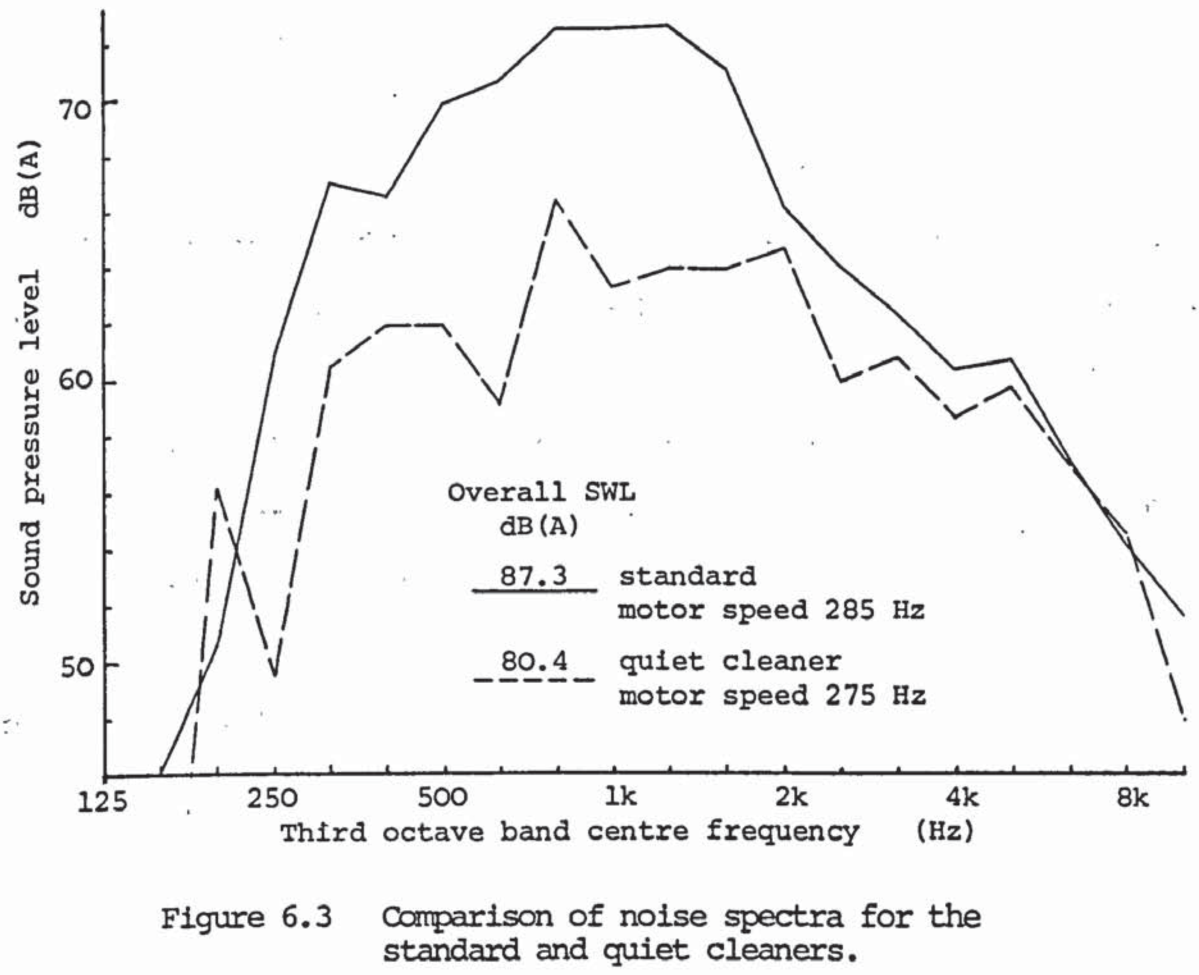


Figure 6.3 Comparison of noise spectra for the standard and quiet cleaners.

are summarised below. A more satisfactory conclusion, however, would have been to evaluate the "Quiet Cleaner" in the anechoic room at the NEL, East Kilbride. Unfortunately there was no opportunity to do this.

Measurement point:	Sound power level dB(A) (at motor speed of 285 Hz)	
	Standard	"Quiet" [*]
Front	87.7	83.2
Left hand side	87.9	83.7
Rear	88.9	82.8
Right hand side	87.7	82.8

* The "Quiet Cleaner" had been stored for six months and the difference in sound levels shown here and those quoted earlier is probably due to deterioration in the isolation properties of the rubber mounts and washers.

The suction, airflow and dust removal performance.

The performance of the cleaner may be affected by the change in the air exhaust route. This was the only modification which would affect the performance. The other changes were structural and away from the air passage and were, therefore, not expected to influence the performance. The results are summarised at the top of the next page. Therefore, after sealing the peripheral gap in the bag housings and introducing two apertures, the suction, air-flow and the dust characteristics remain comparable to a standard cleaner.

	Cleaner:		
	Standard	"Quiet" (at 285 Hz)	
Maximum air-flow	39.5	36.2	10^{-3} m^3/s
Maximum suction	570	600	mm water
Maximum suction power	54.9	56.1	watts
Motor speed at open flow	17000	17100	rpm
Dust removed 1st cleaning	54.0 %	57.6 %	
" 5th	75.3 %	75.9 %	

6.3.3 Noise and vibration criteria

In Chapter 2, a number of criteria which can be used to evaluate the effects of noise were discussed. These can be factors such as noise exposure, speech interference, perceived loudness, annoyance and vibration exposure. The intention of this section is to evaluate the results of this project in terms of these factors. It was appropriate to introduce a competitor to the Turbopower cleaner so that the noise ratings can be judged from a noise-labelling standpoint. An Electrolux Z610 upright cleaner was chosen.

Hearing damage risk

The service life of a cleaner is assumed to be 500 hours (as used in life-testing) and this is based over a time period of 10 years. Therefore, the cleaner usage is approximately 1 hour per week and hence 10-12 minutes per day. The equivalent continuous noise level over 8 hours Leq (8 hr) was obtained using the nomogram in HMSO (1981). The results for the Turbopower cleaners ("Quiet", standard and typically noisy) and the Electrolux Z610 upright are summarised at the top of the next page. Under the present limit of 90 dB(A) Leq (8 hr), therefore, there will be no risk of hearing loss for normal use of any of these cleaners.

Cleaner	SPL [*] dB(A)	Leq (8 hr) dB(A)
"Quiet"	73.4	58.5
Standard	80.5	65.0
Noisy	88.0	73.0
Electrolux Z610	75.5	61.0

* The sound pressure level at a distance of 1.5m from the centre of the foot of the cleaner in a semi-reverberant sound field.

Loudness and annoyance

The third octave sound levels for the Turbopower ("Quiet", standard and typically noisy) and the Electrolux Z610 are tabulated in Table 6.1. A procedure to determine the "loudness" of a sound is given in the ANS Standard S3.4 (1968). It involves the conversion of the linear (that is not A-weighted) sound pressure level in each octave band into a loudness index. From the loudness index, the total loudness in "sones" was computed, and then the loudness in "phons" was obtained using a nomograph. The perceived loudness level, measured by "noys", was determined using the data in Beranek (1960). Table 6.2 summarises the overall results.

The phon and the sound power level are both logarithmic scales and therefore, there is no benefit in using phons because its derivation is the more complex of the two. Moreover, the phons scale compresses the gap in the results. In contrast, the sone scale is linear and able to portray the subjective "noisiness" better. It does this by "expanding the gap" in the overall results as shown in the table.

Perceived noisiness

The loudness level in "sones" was based upon a reference sound; that of a pure tone of 1000 Hz. Common sounds are rarely discrete. Therefore, a reference sound comprising a collection of frequencies was

			THIRD OCTAVE BAND CENTRE FREQUENCY (Hz)																					SWL dB(A)
			100	125	160	200	250	315	400	500	630	800	1000	1250	1600	2000	2500	3150	4000	5000	6300	8000	10000	
Turbopower cleaners;	"Quiet"	dB(A)	44.2	23.3	33.4	56.0	49.6	60.4	61.8	61.8	59.1	66.3	63.2	63.9	63.9	64.6	59.8	60.6	58.6	59.6	56.8	54.4	47.9	80.4
		dB	63.3	39.4	46.8	66.9	58.2	67.0	66.6	65.0	61.0	67.1	63.2	63.3	62.9	63.4	58.5	59.4	57.6	59.1	56.9	55.5	50.4	
		Octaves		63.5			70.3			69.6			69.8			66.9			63.6			59.7		
	Standard	dB(A)		33.9	46.2	50.8	60.7	67.0	66.5	69.8	70.8	72.5	72.5	72.6	71.0	66.2	64.0	62.4	60.3	60.6	57.2	54.0	51.6	87.3
		dB		50.0	59.6	61.7	69.3	73.6	71.3	73.0	72.5	73.3	72.5	72.0	70.0	65.0	62.7	61.2	59.3	60.1	57.3	55.1	54.1	
		Octaves		60.0			75.2			77.1			77.5			71.8			65.1			60.6		
	"Noisy"	dB(A)	41.0	45.6	39.3	45.2	79.6	81.4	66.4	65.7	71.2	68.1	70.6	70.9	72.8	68.0	62.8	61.5	61.0	61.5	58.0	51.8	40.8	92.1
		dB	60.0	61.7	52.7	56.1	88.2	88.0	71.2	68.9	73.1	68.9	70.6	70.3	71.8	66.8	61.5	60.3	60.0	61.0	58.1	52.9	43.1	
		Octaves		64.2			91.1			76.2			74.8			73.2			65.2			59.0		
Electrolux	Z610	dB(A)	26.1	37.8	38.3	54.7	58.5	60.2	63.7	68.5	65.9	65.0	64.4	62.0	64.6	61.0	62.5	60.2	63.6	60.0	58.5	57.6	54.2	82.5
		dB	45.2	53.9	51.7	65.6	67.1	66.8	68.5	71.7	67.8	65.8	64.4	61.4	63.6	59.8	61.2	59.0	62.6	59.5	58.6	58.7	56.7	
		Octaves		56.0			71.3			74.5			68.9			66.7			65.6			63.0		

Sound pressure levels.

Table 6.1 Third octave band sound pressure levels for the "Quiet", standard and "Noisy" Turbopower cleaners and the Electrolux.

Cleaner			Octave band centre frequency Hz 125	250	500	1000	2000	4000	8000	Overall SWL dB(A)	Loudness "Sones"	"phons"	Perceived loudness "Noys"	"PNdB"
Turbopower	"Quiet"; at 275 Hz	SPL dB	63.5	70.3	69.6	69.8	66.9	63.6	59.7	80.4				
		L Index	3.1	6.4	7.3	8.8	8.8	8.6	8.1		21.5	84.3		
		"noys"	2.3	7.0	8.2	8.3	9.1	12.0	16.1				30.1	89.2
	Standard;	SPL dB	60.0	75.2	77.1	77.5	71.8	65.1	60.6	87.3				
		L Index	2.4	8.4	11.1	14.0	11.7	9.4	8.6		29.5	88.8		
		"noys"	1.6	9.9	13.8	14.2	12.4	13.2	17.2				36.7	92.1
	"Noisy",	SPL dB	64.2	91.1	76.2	74.8	73.2	65.2	59.0	92.1				
		L Index	3.3	23.2	10.7	11.7	12.8	9.5	7.8		39.9	93.2		
		"noys"	2.5	30.2	12.9	11.7	14.2	13.3	15.4				51.2	96.9
Electrolux;	Z 610	SPL dB	56.0	71.3	74.5	68.9	66.7	65.6	63.0	82.5				
		L Index	1.8	6.8	9.6	8.3	8.6	9.7	9.9		23.5	85.7		
		"noys"	1.1	7.5	11.4	7.9	9.0	13.7	20.0				35.2	91.5

Table 6.2 Subjective sound levels for the "Quiet", standard and the "Noisy" Turbopower cleaners and the Electrolux.

used by Kryter and Pearson (1963). They defined the perceived noise level, PN, as the sound level of a band of noise (the reference) from 910-1090 Hz that sounds as noisy as the sound in question. The perceived "noisiness" in "noys" is calculated in a similar way to its counterpart, the "sone". The results were presented in Table 6.2 and shows that the "noys" scale "expanded the gap" in the results even more than the "sones". The "noys" and the "sone" scales are linear, and therefore, the result suggests that the noisy Turbopower cleaner is subjectively almost twice as loud as the "Quiet Cleaner".

In conclusion, therefore, the "subjective noisiness" scale of "noys" was preferred when evaluating household appliances. According to Lord et al (1980), the "sone" scale was previously used as an index for rating appliances but unfortunately, it did not find widespread acceptance because of the complex procedure. In the author's opinion, however, the derivation of "noys" or "sones" is not particularly difficult. Considering that a value in "noys" is required only for the final evaluation, the intermediate comparisons can still be made using the A-weighted sound level. A case is made herewith that the "noys" scale should be adopted for product noise-labelling.

Speech interference level

The preferred speech interference level, PSIL, is the average of the sound level in the octave bands centred on 500 Hz, 1000 Hz and 2000 Hz. The results are summarised as follows;

Cleaner	Octave band SPL dB			PSIL
	500	1000	2000 Hz	(dB)
"Quiet Cleaner"	69.6	69.8	66.9	68.8
Standard	77.1	77.5	71.8	75.5
Noisy cleaner	76.2	74.8	73.2	74.7
Electrolux Z610	74.5	68.9	66.7	70.0

Note that the noisy cleaner had high sound levels at low frequency, mainly at the motor speed, see Table 6.1. These frequencies, however, were outside those considered for speech interference and therefore, this cleaner was "rated" as being similar to the standard cleaner. The results are superimposed onto the rating chart, as illustrated in Figure 6.4. The following conclusions can then be made:

- a conversation with normal voice level is possible up to a distance of 1 metre when operating the "Quiet cleaner",
- the speaker has to raise the voice level to near shouting for a conversation over the same distance when operating the standard or "noisy" Turbopower cleaners.

Vibration criteria

The vibration levels at the handle for the standard and the "Quiet" cleaners are greatest in the fore-aft axis. The major frequency components were around 250-1000 Hz. The rms vibration levels were measured and given below. By considering the frequency components of the vibration and the duration of the use of the cleaner, as shown in Figure 2.7, it follows that the vibration levels are within the limits.

Cleaner:	Vibration level (g)
Standard	6.3
"Quiet" (at 275 Hz)	3.4

6.3.4 Subjective assessment of the "Quiet Cleaner"

The "Quiet Cleaner" has been shown to be comparable to a standard cleaner as regards dust removal performance and much superior as regards the sound power level. A subjective assessment of the "Quiet Cleaner" was sought from reponsible personnel. Two opportunities presented themselves.

Firstly, at the 8th meeting of the Hoover plc/IHD Noise Control

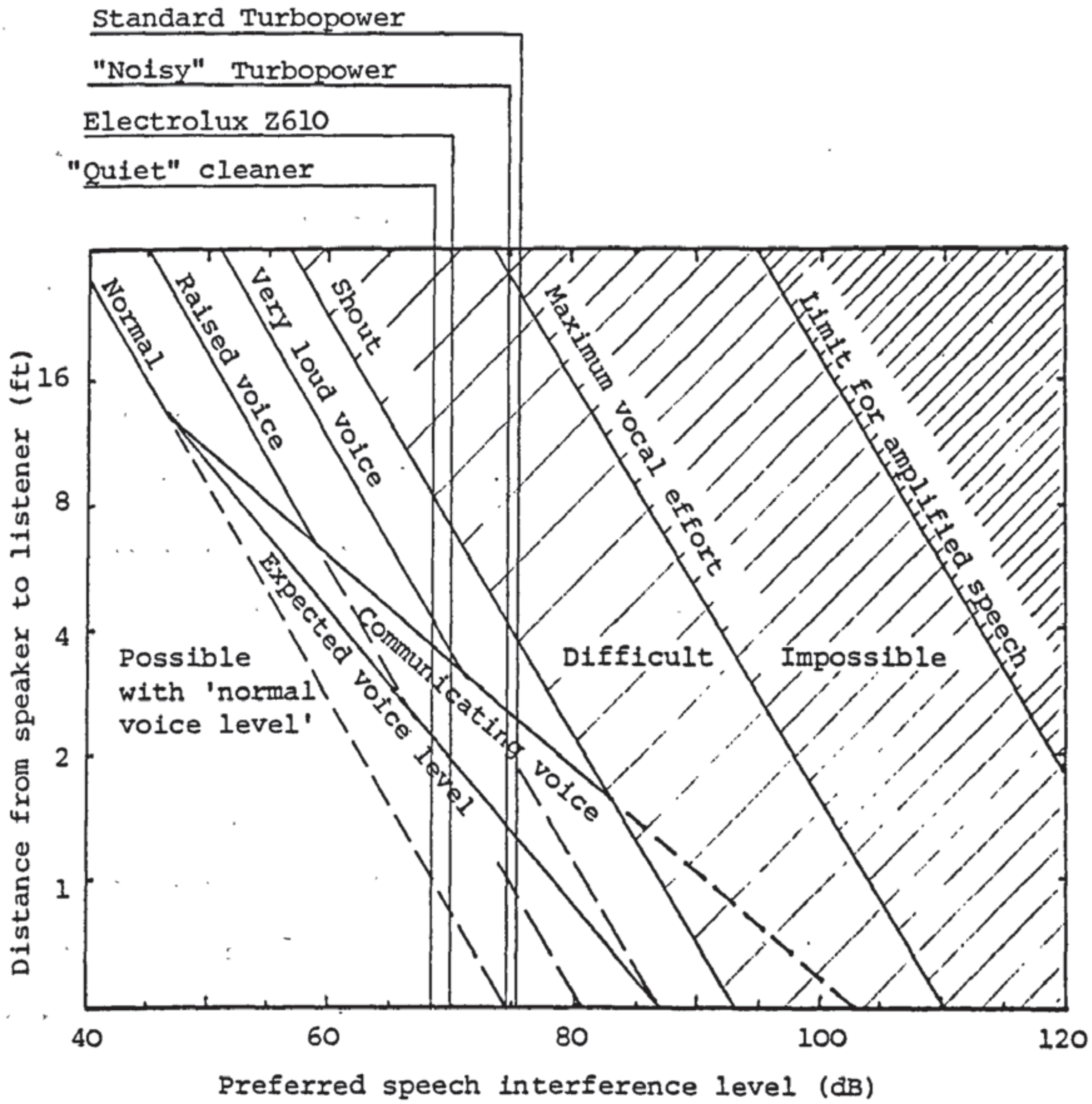


Figure 6.4 Evaluating speech communicating capability for cleaner noise levels.

Committee (the project steering group organised by the author), the "Quiet Cleaner" was compared to a standard Turbopower. The meeting was attended by the Chief Engineer in addition to the project supervisory team. All members of the team evaluated the two cleaners and noticed a significant reduction in the noise level of the "Quiet Cleaner" (running at a motor speed of 285 Hz (17100 rpm)). Although, there was caution concerning the practicality of covering the bellows in the way it was fabricated, there was a general agreement that detail design would cure the problem.

Secondly, the author gave a seminar to the Mechanical and Production Engineering Department, University of Aston. Following the seminar, the "Quiet" and a standard Turbopower cleaners were presented to the audience comprising members of staff and other research students. The audience remarked that there was a significant reduction in the noise level of the modified cleaner.

6.4 Economic consideration

Noise reduction is not a particularly difficult task, it is often quite easy to over-quieten an appliance. The "art" of noise control, however, lies in effecting economic solutions. The economics of cleaner production - as with all mass produced products - is an important factor in choosing the amount of noise suppression to be applied.

The cost of noise reduction as seen by Hoover plc, is a combination of the actual price of the component, that is, the extra material such as any brackets or screws necessary, and two additional costs called the "fixed burden" and the "variable burden". The fixed burden is a proportion of the fixed company overheads attributed to any manufacturing process. The variable burden relates to the time taken to complete an operation. The following cost estimates were obtained from Hoover plc's Cost and Purchasing Department at Perivale. Only the estimate of the direct unit cost and the necessary tooling investment of the noise treatments can be made, it is very difficult to evaluate the indirect costs and benefits, such as the product image, lost sales, extra sales effort and the warranty expense.

6.4.1 A cover for the bellows

Current:	The lower cover plate, D 37257-509, covers the electrical cord just before the cord passes through the trunnion. The cover plate contains a thin sheet of ABS which runs along the bellows to form a box section.	
Proposed:	The thin sheet described above to be removed and replaced by a thicker sheet to cover the bellows. A screw fixing will be required.	
Tooling:	New lower cover plate tool (2 cavity)	£ 23000
Unit cost:	Extra assembly + screw	+3 p/cleaner
Comment:	There will be some material saving.	

6.4.2 Isolation of the hood

Current:	There is a direct contact between the cleaner chassis and the hood at the attachment points.	
Proposed:	A rubber washer to be used under each screw head.	
Tooling:	Changes to two chassis tools.	£ 5000
Unit cost:	Two rubber caps including tool amortisation	+6 p/cleaner
Comment:	The screw receptacle on the hood may have to be recessed. This will save some material.	

6.4.3 Norsorex rubber mounts

Current:	Neoprene rubber mounts are used for the motor.
Proposed:	A softer Norsorex rubber compound is proposed.
Tooling:	Nil.

Unit cost: Large mount on-cost 9.5 p/cleaner

Small mount on-cost approx 2.5 p/cleaner

Comment: Extra material can be removed from the large mount for enhanced resilience and cost saving.

6.4.4 Isolation of the agitator ends

Current: The two thread guards have a spigot which slots into the corresponding cut-out in the chassis.

Proposed: Eliminate spigot at each end. Thread guard to be located at the circumference, hence a cut-out of 50mm diameter is required in the chassis, which will be slightly reinforced at these points. A rubber ring to be sandwiched in-between.

Tooling:	New thread guards	£ 2000
	Chassis changes (2 tools)	£ 5000
	Agitator changes (3 tools)	£ 6000
Unit cost:	2 rubber mounts plus tool amortisation	11 p/cleaner

Comments: There will be some material saving.

6.4.5 Control of the exhaust air

Current: There is a gap between the bag housings which allow the air to flow out.

Proposed: The gap to be sealed. This requires close fitting housings and a strip of closed-cell sponge to seal the peripheral gap. Two apertures, with microfilters, at the bottom of the rear bag housing to allow the air to flow out.

Tooling:	Changes to front bag door (3 tools)	£ 15000
	Changes to rear bag door (2 tools)	£ 8000
	Microfilter support moulding (2 cavity)	£ 12000
Unit cost:	1.3m of self-adhesive seal	+25 p/cleaner
	Strip extrusion and assembly	+11 p/cleaner

Microfilter +25 p/cleaner

Comment: There will be material some saving from the two apertures.

6.4.6 Tighten wheel-trims

Current: The wheel trims are loose in the wheel assembly.

Proposed: Slight re-inforcement of the three attachment legs.

Tooling: Changes to wheel trim (2 tools) £ 6000

Unit cost: Virtually nil.

6.4.7 Reduction of the motor speed

Current: Speed of 285 Hz is close to structural resonance activity,

Proposed: Motor speed of 270-275 Hz (16500 rpm approx.)

Tooling: Virtually nil except for re-setting.

Unit cost: 3.8% increase in length of copper wire.

Copper wire on the armature costs 13p - will cost 13.5p

Copper wire on the field costs 28p - will cost 29.1p

Total increase in cost +1.6 p/cleaner

The details are summarised in the table on the next page. The reduction in the noise level apertaining to each solution and the introduction, whether immediate or long term, of each treatment is also given.

Even after determining and analysing the noise drop per unit cost, it is not easy to decide which solution to adopt, because in some ways the choice is dependent upon the effect a solution has on the variation in the sound level. In fact, it is up to the senior management strategist to specify, not just the absolute noise level, but also the "limits" on the variation. The most suitable noise reducing treatment can be proposed in the light of this information.

Noise treatment:	COSTS Tooling £	Pence per cleaner	Drop in SWL dB(A)	Noise treatment intro.
Bellows covered	23000	3	3.5	LT/facelift
Isolation of hood	5000	6	2.0	immediate
Norsorex mounts	-	12	2.5	immediate
Isolation of agitator ends	13000	11	2.0	LT/facelift
Exhaust air re-directed	35000	60	2.1	LT/facelift
Tight wheel trims	6000	-	1.0	immediate
Lower motor speed	-	1.6	1.9	immediate

6.5 Review

This chapter evaluated those treatments which gave a significant reduction in the overall noise of the cleaner. The treatments were fabricated into a standard Turbopower cleaner. The objective evaluation and the subjective trials proved that the "Quiet Cleaner" was successful, both in terms of the lower sound level and comparable performance. The noise treatments were feasible for production. The subjective loudness of the cleaner was rated in a number of ways, and the "noys" scale was found to be the best suited for product noise-labelling.

The total noise reduction of 7.1 dB(A) cost 93.6 pence per cleaner. Without modifying the exhaust air route, a noise reduction of 5.7 dB(A) would cost just 33.6 pence per cleaner.

CHAPTER SEVEN: DESIGNING FOR QUIETNESS

Contents

7.1 Introduction

Downham (1965) recognised that;

"the trouble with acoustics and vibrations as a subject is that in many cases no one wants to know about it until it is too late. As a result, very often the noise and vibration engineer is looked upon as a trouble-shooter to be kept in a back room until circumstances dictate that he shall be released".

Noise reduction is generally considered to be a process that is normally applied as an after-thought; that is prescriptive or curative. The more enlightened view, however, should be one of "noise control" achieved by following the appropriate steps in the design phase itself.

This chapter analyses the aims of a design strategy and then discusses the current design strategy of Hoover plc. The integration of the noise control guidelines should prevail at the outset of a new product design. The concept of "sonance" design, that is, designing for quality sound, is reviewed to this effect. For completeness, the influence on the noise level of the manufacturing variations in the cleaner and motor assemblies at Cambuslang are discussed. In the light of these experiences, an outline of the proposed changes to the design strategy of Hoover plc are given. The major emphasis must be on providing adequate check-points from start to end of the design process, and greater interaction between all of the disciplines which are important to a successful design, namely marketing, quality control and production.

Essential to any design strategy is the availability of correct and concise information and theoretical guidance for the incorporation of disciplines involved in a given design. A gap in knowledge was identified at Hoover plc. Members of the design team were not familiar with the fundamental principles of "designing for noise control". Consequently, a handbook of noise control guidelines was produced. The feedback arising from this handbook was brief but encouraging and is presented in this chapter.

7.2 A design strategy

Turner (1966) defined a strategy for effective design, as the art of conducting the design process efficiently within an organisation. Moreover, design is a complex activity which requires the collaboration of many specialists in different disciplines. The commercial success of a product is greatly influenced by its design which predetermines such factors as the performance, appearance, reliability, maintainability and production cost. The management of design is, therefore, important and fundamental to the creation of new products. This view is identified and shared by The Engineering Council (1986), which suggested that,

> "the board members of a company should be aware of the far-reaching impact of design and ensure that the general management principles are rigorously applied to the management of design".

Figure 7.1 highlights the key role played by "design" in the total business cycle. The interaction between the different departments and the information flow are clearly indicated. The customer and the design are in the mainstream of translating ideas into hardware. The phases involved in the introduction of a new product can be shown by the flow diagram in Figure 7.2. It emphasises the need for providing adequate check-points throughout the complete operation.

Turner (1966 p253) stated that a policy is not a directive or a command, but a guide. Subsequently, a company design policy exists to clarify and disseminate the viewpoints of the top management concerning the direction. This provides a pattern or a framework within which the design group may operate. Such a guide is used to establish the scope of product design decisions and anticipate the future trends and conditions.

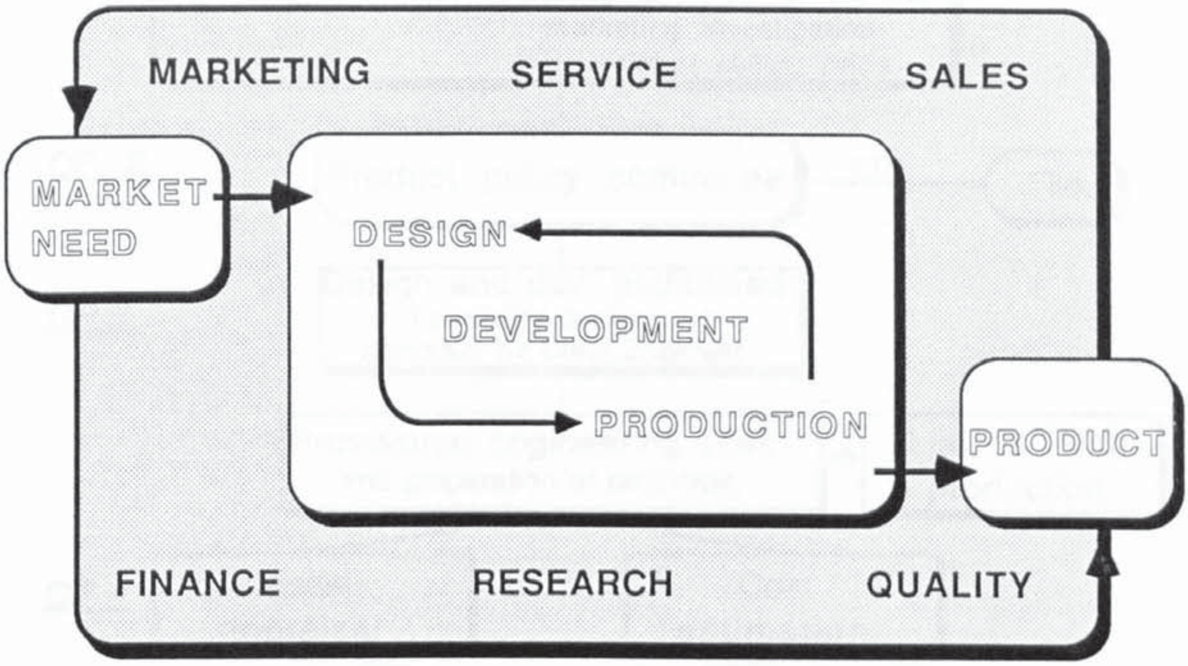


Figure 7.1 The role of design in a business cycle.

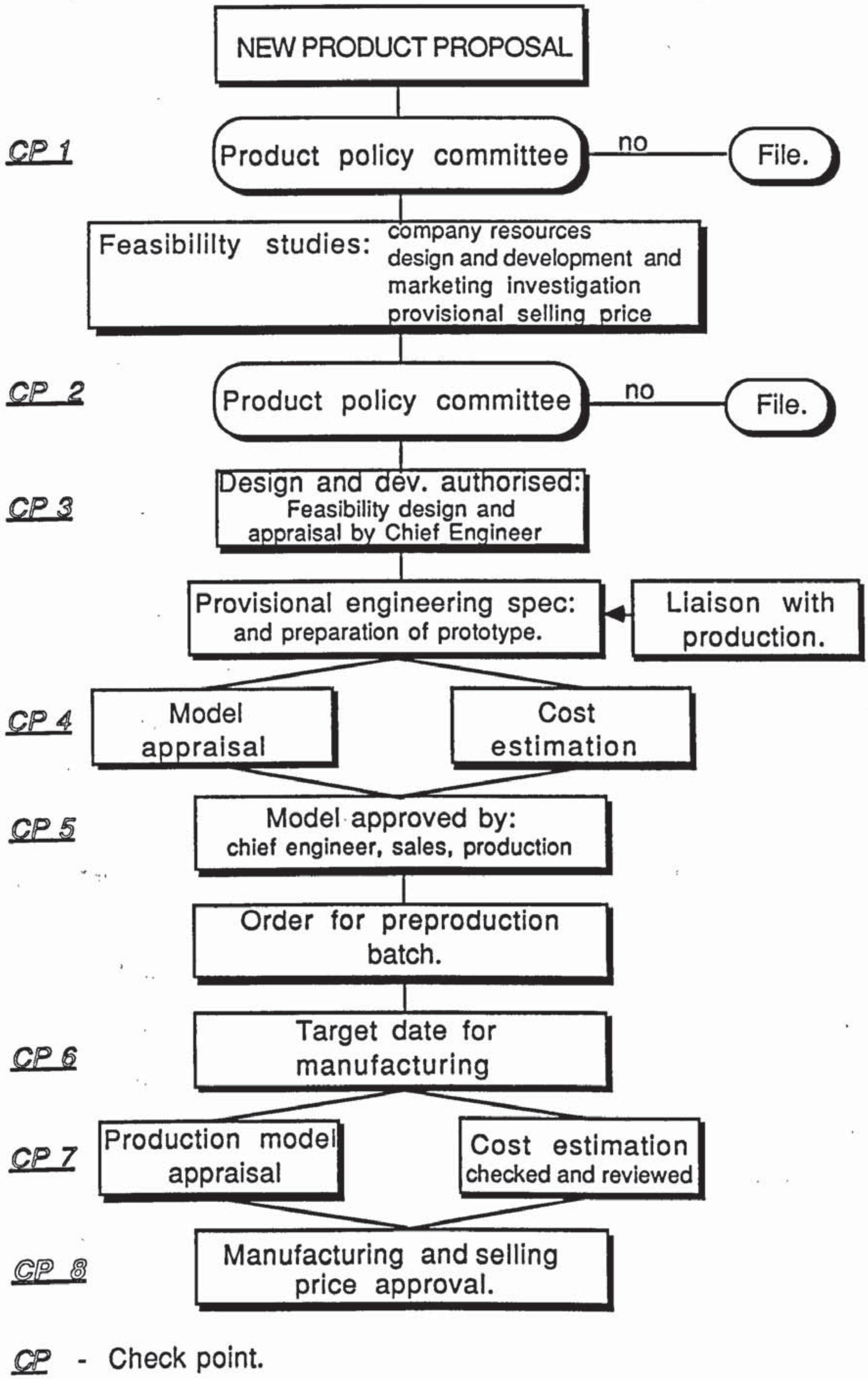


Figure 7.2 Introduction of a new product emphasising check-points.
(Adapted from Turner (1966))

The design process and particularly the mix of the various company activities such as basic research, added value, development and quality control are dependent upon the overall strategy for the product.

7.3 The current design policy of Hoover plc

The design department at Hoover plc, Perivale, is responsible for the overall design of white goods and floorcare products. There is a large design department at the Hoover headquarters at North Canton, USA.

A small survey of designer's experience and attitudes was conducted at Perivale. The aim was to determine the current design policy. A list of appropriate key questions was formulated and from this a questionaire, shown in Table 7.1, was drawn up.

There may be difficulties when a researcher seeks information from personnel in an organisation in which the researcher is not a permanent employee. Information given may reflect the designer's perception of the use the information will be put. In many cases the researcher is often treated with caution, and in some quarters he may even be considered a management "spy". Consequently, in an attempt to gain useful and correct information, time was spent in gaining the confidence of designers by ensuring casual contacts in the period preceding the survey. In the event, it was gratifying to observe how forthcoming were the people interviewed. Nevertheless, some of the information has to be disregarded due to the "distortion of facts". By cross referencing the information obtained with that from other sources, it was ensured that only good corroborated evidence was used.

7.3.1 Major findings

None of the designers had any education in acoustics and vibration. Many had been out of formal education for many years and consequently, claimed a long job experience. Naturally, designers have most of the

General data; Job specification :
Qualifications :
Area of responsibility :
Experience in the area :

Noise aspects; Novice 1 2 3 4 5 Expert
(tick as applicable)
familiarity of acoustics fundamentals
experience in noise control
how was this knowledge of noise acquired external/internal courses
who do you turn to for advice external/internal contacts

Do you consider reduction important? yes/no
if noise reduction does become important,
what information, in your opinion, is essential... (list in order of importance)
1..
2..
3..
4..
5..

would you prefer; to be sent on courses yes/no
a handbook of design guidelines yes/no
better noise analysis facilities yes/no
what else? ....
....

How significant are the following to noise reduction;
material selection 1 2 3 4 5
critical speeds 1 2 3 4 5
machining tolerances 1 2 3 4 5
limits and fits 1 2 3 4 5

What sort of information would you like to see in the handbook...bear in mind the number of books already available.
....
....
....

List the five noisiest household appliances, worst first..
....
....

Briefly, what in your opinion is the current design policy?

Table 7.1 The questionaire.

"relevant" information close at hand, either formally in manuals or informally by consultations with their colleagues. Surprisingly, a few claimed that there was no need to approach an "outsider" for noise related information because "common-sense prevails". However, if noise reduction did become important, designers preferred to be sent on appropriate external courses. A majority recognised the shortcomings of the facilities for sound measurement and analysis presently available at Hoover plc, and expressed the need for better facilities.

The general consensus was that noise was important. High noise level was considered to be a defect in a cleaner. In attempting to reduce the noise level, cost, closely linked with the demands of the marketing department, was the first prompted answer followed by a search for information on the previous cleaners, of which there was plenty within the company. Most of this information, however, is stored informally as experience and knowledge of other employees and only some of it is formally recorded in reports. It seemed strange, to the author, that Marketing was in such a strong position.

Machining tolerances and limits were considered as part of the design process. These factors are indeed taken into account at the design stage, but from a cost of manufacturing viewpoint and not, the author deduces, from their noise emission consequences. The need to select alternative materials was considered important for noise reduction but again, cost was stressed to be the ultimate decider. The design staff appreciated the negative effects of resonances close to the operating speed. However, a point was made that the motor speed was stipulated by the motor laboratory (to meet specifications, such as the performance and cost of manufacture). Any change in the motor speed is at the discretion of the senior management.

The vacuum cleaner was cited as the noisiest appliance in a household followed by the washing machine (in high speed spin mode) and the grinder-blender.

7.3.2 The current design process

The process for the design of a new product is initiated by a product requirement. This can be in the form of a brief from the Board of Directors or Marketing to supply a new market, or as a result of a market research survey. The design process is depicted in Figure 7.3 and is self explanatory. The management of a new product is with senior management up to the Industrial Design stage and transfers to a designated Project Engineer thereafter until the whole project is given the go-ahead for production. Then the responsibility is with the Production Engineering Department. At each crucial design or decision stage the emphasis is on low cost, ease of service, reliability and attractive external appearance.

Policy of tests for vacuum cleaner noise.

On the author's visit to The Hoover Company Headquarters in Ohio, their check list for noise tests was obtained. This check list, see Table 7.2, had been formulated and implemented some years ago. However, the Acoustics Laboratory manager confessed that,

> "he would be surprised if the quality control engineers on the production line actually adhere to it, because they will not be able to provide the relevant information".

No such check list was discovered at Hoover plc in Perivale, where, only after the prototypes of a cleaner have been made, is the cleaner noise level determined for the first time. If the noise level of a cleaner fails to meet the specification, then Engineering will put forward a table of costs related to certain modifications which, from experience, offer a certain noise reduction. The writer (1986a) discovered that this exercise is a compromise between noise reduction, weight, performance, styling and cost.

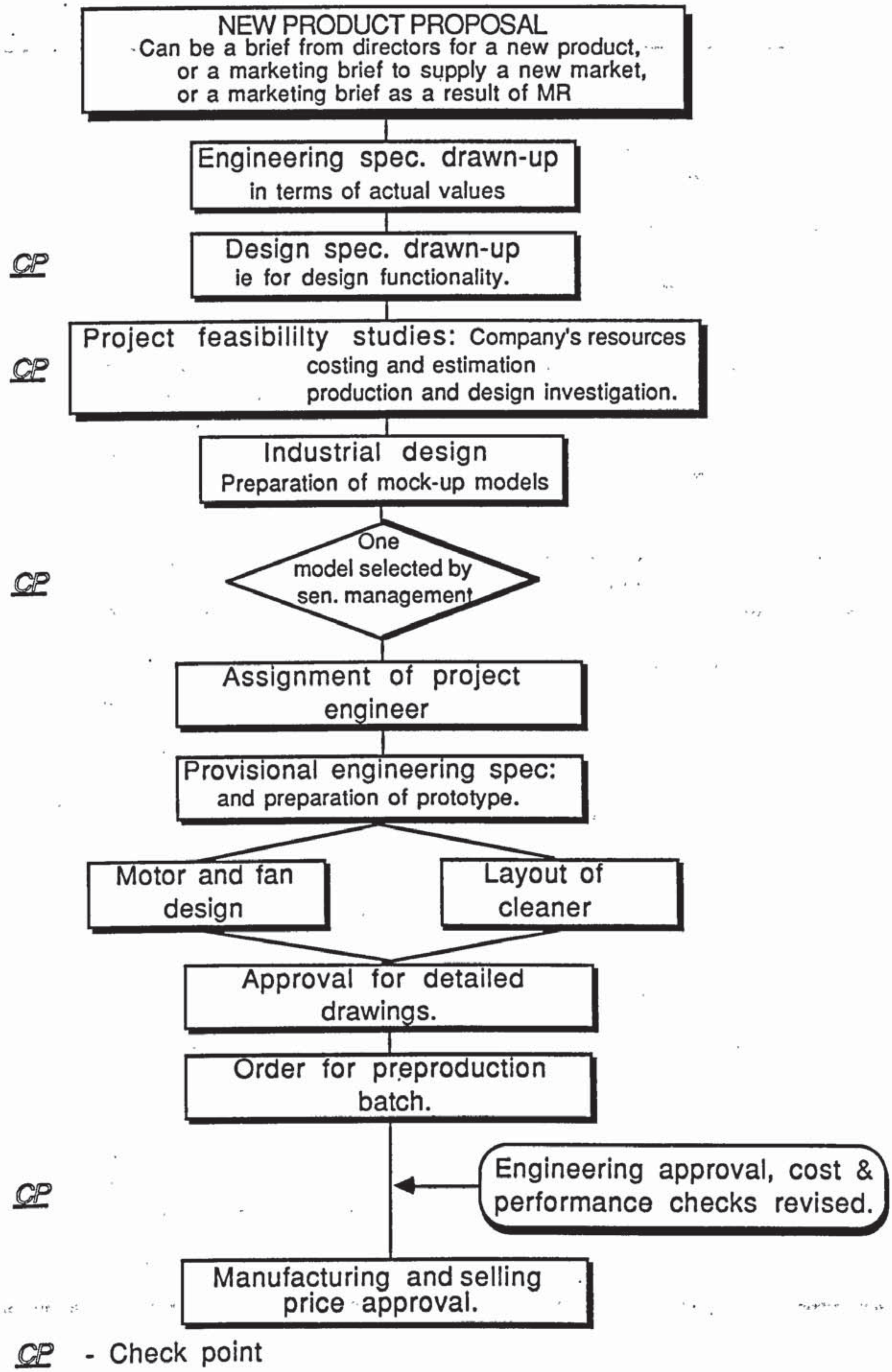


Figure 7.3 The current procedure for new product development at Hoover plc.

North Canton Noise-check Test-list.

1. Early development stages - proposed design on paper.
   - 1.1 Motor design: run UNBANAL analysis, determine balancing hardware,
   - 1.2 Motor mounting evaluation,
   - 1.3 Exhaust evaluation and design of inexpensive muffler features, such as bends and spaces for foam,
   - 1.4 Interface noise: evaluation of agitator design and speed.

2. Early performance models.
   - 2.1 Motors: check motor noise for open airflow and evaluate the speed and airflow influence on noise; are these with targets?
   - 2.2 Motor mounts: evaluate trasmissibility if possible,
   - 2.3 Interface noise,
   - 2.4 Exhaust: determine exhaust contribution and develop recommendations,

3. Shop detail models.
   - 3.1 Transmission paths: analyse contributions of various paths to overall noise and check transmission path for pure tones,
   - 3.2 Standard cleaner performance: note cleaner noise in standard operation and check noise variation with speed and airflow,
   - 3.3 Motor mounts: check mounting arrangement and material,
   - 3.4 Motor: as in 2.1, and develop unbalance vs. motor mount vs. noise based on predicted unbalance and proposed structure, then recommend rotor out-of-balance,

4. Detail models from production tooling.
   - 4.1 Structural noise: evaluate structural noise vs. unbalance and predict likely statistical parameters for cleaner noise, also check for strengthening or damping if required,
   - 4.2 Transmission paths: as in 3.1.

5. Trail and pilot.
   - 5.1 Population estimates from a sample of cleaners, obtain mean and standard deviation,
   - 5.2 Transmission paths: recommend changes if production cleaners are not in line with predictions,

6. Miscellaneous product tests.
   - 6.1 Hoses: check for flow sensitive whistles,
   - 6.2 Straight air nozzles: check airflow vs. noise,
   - 6.3 Lightbulbs: check for resonances,
   - 6.4 Check vibration at the cleaner handle grip and circuit board mounting,
   - 6.5 Electrical leads: check for resonances.

Table 7.2 The North Canton noise check test-list.

7.4 The general principles of noise control by design

Noise should be an important consideration in the design process. A vacuum cleaner should have a low noise level and this sound should be "non-offensive". Designing for quiet, pleasing sound was poetically termed "sonance" design by Sawyer (1955).

Sonance design must start with the initial design ideas, because improper initial design may make it impossible ever to achieve a satisfactory control at a reasonable expense. A designer should therefore try to incorporate those ideas that have led to good "sonance" results in the past. The success of noise control depends on the decisions which have to be made at a very early stage. Consequently, previous experience is a useful source of information. So, the designer should be aware of and appreciate the proven noise control principles and noise reduction treatments. In addition, a knowledge of the acoustics theory is required to estimate the noise level of a given source and the potential noise attenuation of common noise reduction treatments. An understanding of the noise criteria (speech interference and annoyance) can help to decide the qualitative effects of the design changes.

The above qualities form an integral part of the noise control plan shown in Figure 7.4. The plan emphasises frequent checks, feedback and the required interaction between various disciplines of the design function.

Prediction of noise levels

The aim is to estimate the approximate sound power output of the major noise sources and then derive the overall sound power level. The sound pressure level at any point in the vicinity of the cleaner can then be determined. For an accurate estimate, the designer should use measured data wherever possible. Consequently, a database of each major noise source needs to be compiled and then regularly up-dated.

Alternative designs which can reduce noise emission at source need to be evaluated. There are very few basic source treatments other than

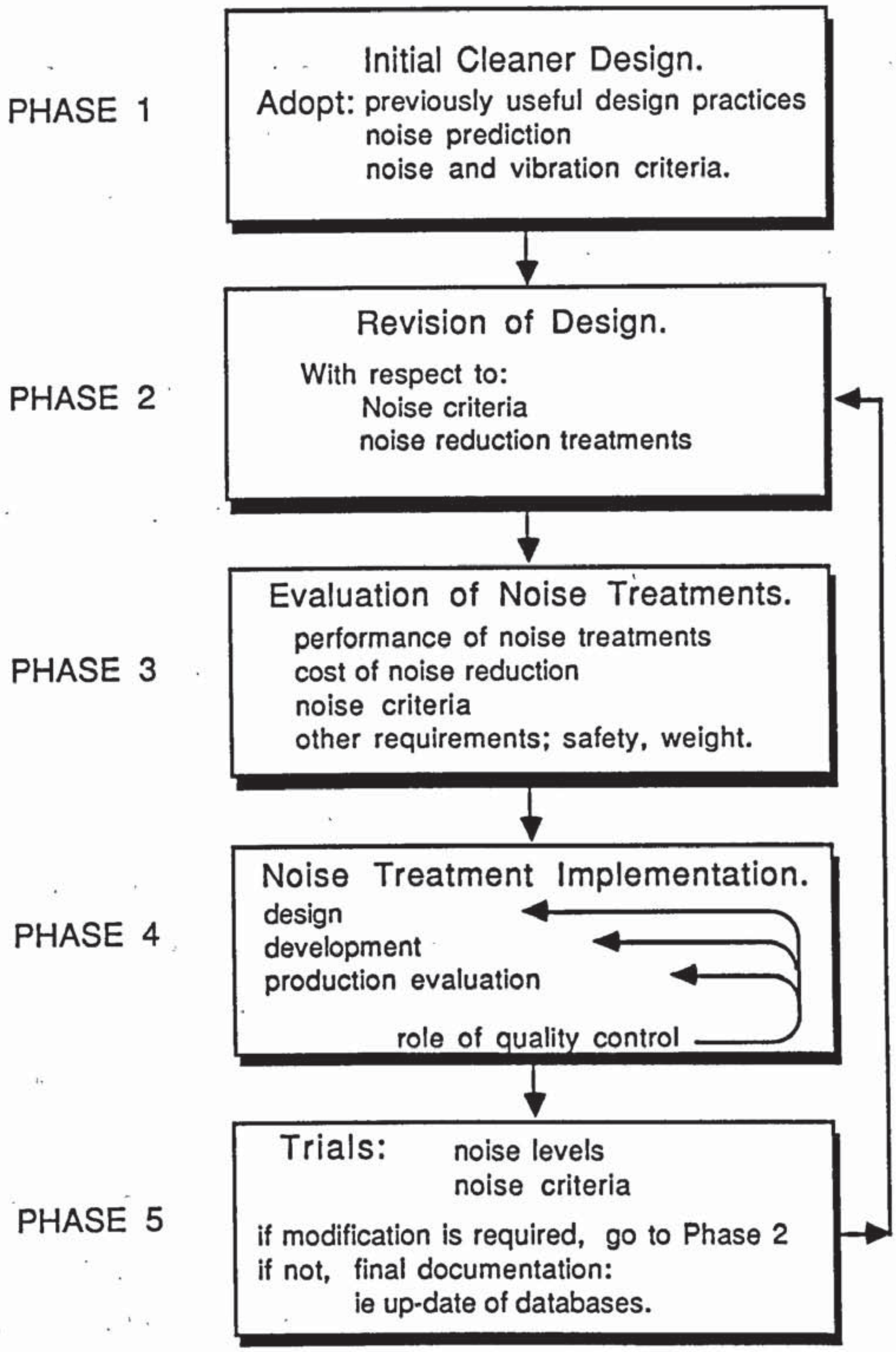


Figure 7.4 Phases in a noise control plan.

those that can be applied in the design stage itself. For instance, a phosphorus-bronze bush complemented by superior assembly methods, is much better than the present arrangement for the agitator bearings on the Turbopower. As a further example, a vacuum cleaner with a "clean-air" system can be considered more refined than one with "dirty-air". In the "clean-air" system, the dust is deposited in the bag before the air passes over the suction fan. Subsequently, the loud impacts caused by hard deris colliding with the fan blades are avoided. These impacts can disturb the balance of the fan-armature assembly, thereafter, leading to "harsh" vibration and unbearable noise. Indeed, previously when the suction fan was made of a zinc alloy blades had been known to disintegrate upon impact with hard debris. Further advantages of the "clean-air" system are that the motor can be post-assembly balanced; also the motor can be cooled by the main air-flow after depositing the dust, and a separate cooling air fan is not required.

Sound propagation and noise control treatments

A model representing the noise sources with their output can be constructed using the database containing the noise levels of the major noise sources. The airborne noise level in the space surrounding the source is the combined result of the transmission of sound through the air and the structure as illustrated in Figure 7.5. This requires a knowledge of the sound transmission loss of basic elements such as a silencer duct and a plastic cover, with or without openings. Transmission paths may contain several elements, in which case, the sound loss associated with each element along the path to the receiver are subtracted from the source level to yield the sound level at any desired point.

A competent acoustics engineer can "condition" the sound emitted by a cleaner. This conditioning is what is implied in "sonance" design. The following basic properties are used.

- the amount of sound insulation and absorption provided by

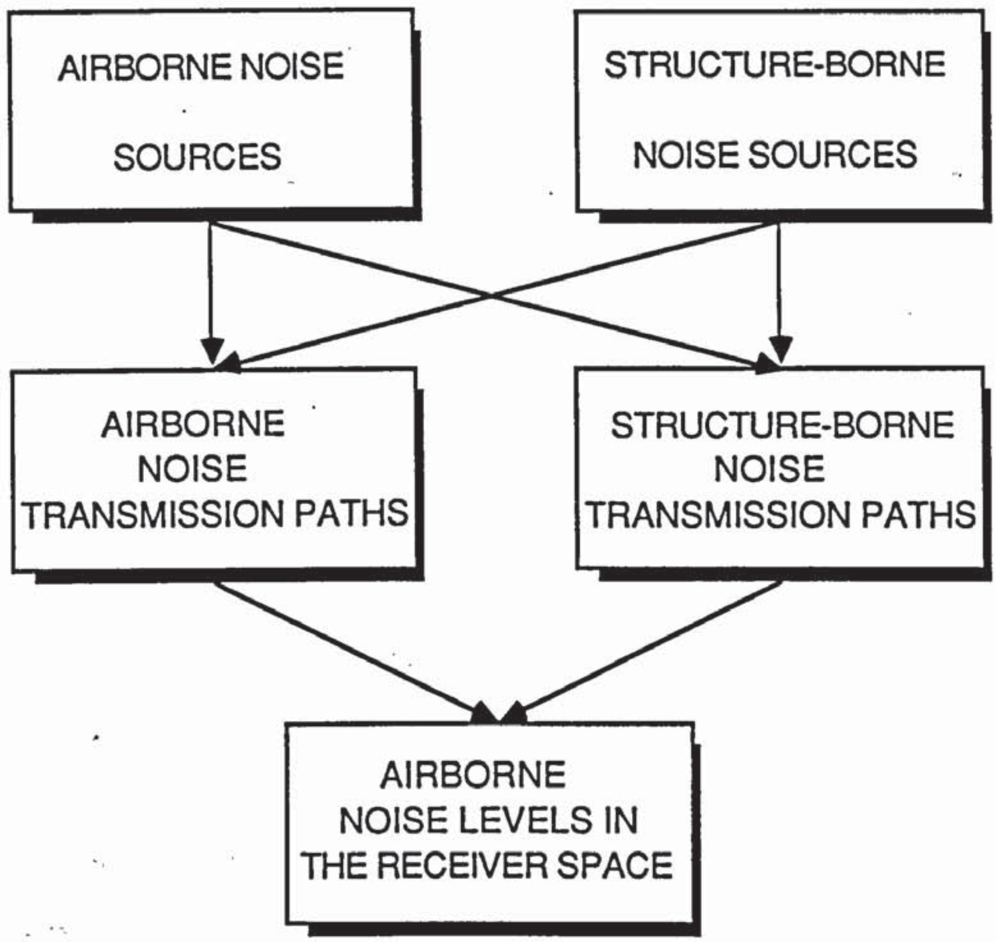


Figure 7.5 Elements in noise radiation and transmission.

acoustic materials is usually greatest in the middle to high frequency range and less effective below 1000 Hz. The heavier and denser an insulating material, the more effective it is at low frequency.

- the sound isolation due to a panel, a duct or an enclosure is greatly enhanced by lining with absorbent material. The sound attenuation increases, albeit at higher frequency.
- the low frequency components are generally due to vibration. The isolation of the vibrating source is the best means for reducing noise in this region. The isolator can be optimised to concentrate the vibration isolation at the desired frequency.
- low frequency noise is also caused by structural vibration. By changing the mass, stiffness or the damping, the sound radiating ability of a panel can be affected. The mix of the treatment can be optimised for attenuation at the desired frequency. Vibration measurements should be made of the sub-assemblies to determine the active regions and highlight the vibrating components.

7.4.2 Good acoustical design practice

Prime mover related

1 Select a low speed motor which is generally quieter than a high speed one.

2 Improve dynamic balance. This decreases the rotating forces and hence reduces the structure-borne sound.

3 Reduce the ratio of rotating masses to fixed masses.

4 Improve lubrication. Inadequate lubrication is often the cause of bearing noise and vibration, which can lead to structural noise and resonances.

5 Maintain closer design and manufacturing limits for bearings and moving parts.

6 Install bearings correctly. Improper installation aggravates bearing noise and vibration and shortens its working life.

7 Mount the prime mover close to a node (position of least vibration) of the foundation or chassis.

8 Maintain an adequate gap between the operating speed and the known lateral, torsional or structural resonances.

9 Select asymmetric fans so that they are least likely to excite strong noise tones or vibration.

10 For the same blade tip speed, select a low shaft speed and a large impeller diameter because this combination produces lower blade passing frequency tone than a high shaft speed and a small fan.

11 Avoid a close gap between the moving elements and those which are stationary because this is a major source of discrete tone.

12 Select a quiet fan and one with a low blade frequency increment because some fans are noisier than others.

13 Operate a fan at its peak efficiency because fans that operate away from their peak efficiency generate higher noise levels.

14 Specify quiet motors and fans if these are bought from external suppliers.

<u>Structure related.</u>

15 Avoid structural resonances by ensuring that the operating speed is away from major resonances. Structural resonances can be determined by vibration measurements of the sub-assemblies; resonances can also be excited by impacts, or by sliding or rubbing contacts.

16 Eliminate or reduce any impact force by reducing the impact mass or the striking velocities. Alternatively, change or add material to dampen impacts.

17 Reduce large, flexible noise radiating areas. Surfaces can be

split into smaller areas which have lower radiation efficiency.

18 Isolate possible sound-radiating panels from vibration. Avoid creating a direct transmission path.

19 Avoid any openings when using enclosures because orifices seriously degrade the sound transmission loss.

20 Use a lining of sound absorbing material on the internal surfaces of sound enclosures as this improves the sound transmission loss.

21 Use heavy panels to insulate the noise source. Panels with greater damping also give superior noise reduction.

22 Take advantage of all directivity effects wherever possible by directing the inlet and discharge openings away from the direction of the listeners.

23 Cover up acoustically transparent areas or components. These areas can be highlighted by near-field sound measurements.

24 Use interference fit to avoid the irritating "rattle" noise due to loose components.

Air-flow related

25 Minimise the flow velocity through passages and bends, where possible. A 15-18 dB increase in the noise level is associated with the doubling of air speed; that is a 2-3 dB increase with each 10% increase in the airflow speed.

26 Avoid abrupt changes in the cross-section of ducts and bends.

27 Avoid a poor inlet to the fan as it can cause significant excess noise.

28 Use gradual bends in the ducting as they provide some natural noise attenuation. Their effectiveness, however, can be improved by lining the duct with sound absorbent material. In contrast however, sharp bends and elbows cause turbulence which directly affect noise generation. The air speed is the vital

factor which tips the balance between the noise attenuated due to the duct and the noise generated by turbulence. In the case of a vacuum cleaner where the air speed is high, generation of noise is predominant.

Resilient mount related

29 Ensure that the mobility of an isolation mount is much more than that of the motor and the chassis (or foundation).

30 Check for and avoid rigid structural or flanking paths between the isolated components. For example, the electrical leads and rigid accustic materials.

31 Design the isolator so that the natural frequencies of the motor on isolation mounts is much lower than the lowest excitation frequency, such as the rotational speed and its harmonics.

32 Place all vibrating sources on a common sub-base which can subsequently be isolation mounted.

This list is not exhaustive but intended to be stimulating and thought provoking. For maximum benefit, these recommendations should be introduced at the initial design stage and carried throughout the design process. The aim of designing for quietness is to use good acoustical design practices so that ultimately the number of post-prototype noise treatments are minimised.

7.5 The influence of manufacturing tolerances on the noise level

A number of cases are apparent which demonstrate that, high noise levels of the cleaner can be due to poor manufacture. Brown (1984, 1985) reported of cleaners which had originally generated high noise levels (and had caused their owners to complain). These cleaners were stripped and re-assembled, sometimes with new suction fans or bearings. In all of the cases, the subsequent noise level of the cleaner was lower, and close to the average sound power level of 87.5 dB(A). It follows, that the source of the scatter in the noise level are the mechanical variations in the assembly.

The effect of the manufacturing tolerances is to introduce a spread around the mean value as shown in Figure 7.6. The shaded region represents the proportion of excessively noisy cleaners which can arouse complaints. There are two methods to reduce this proportion;

- by reducing the mean sound level. Thereby shifting the curve towards a lower sound level as shown in Figure 7.6a. This can be achieved by implementing the noise reduction treatments,
- by reducing the standard deviation. The mean value will remain same, but the curve will have a narrow distribution, as shown in Figure 7.6b. This can be achieved by reducing the production manufacturing variations.

The maximum manufacturing variations which were found in the main parameters, and a summary of the design limits, are shown below.

Component:	Design limits.	Manufacturing variations (max)
Suction fan out-of-balance	0.432 gm-cm	1.34 gm-cm
Cooling fan out-of-balance	0.432 gm-cm	0.23 gm-cm
Dynamic armature out-of-balance	0.34 gm-cm	2.5 gm-cm
Bearing misalignment	0.1 mm	0.3 mm
Bearing journal	0.006/8 mm	0.015 mm
Nozzle height (position 2)	3-4 mm	6 mm
Sound power level of cleaner	87.5 dB(A)	96 dB(A)

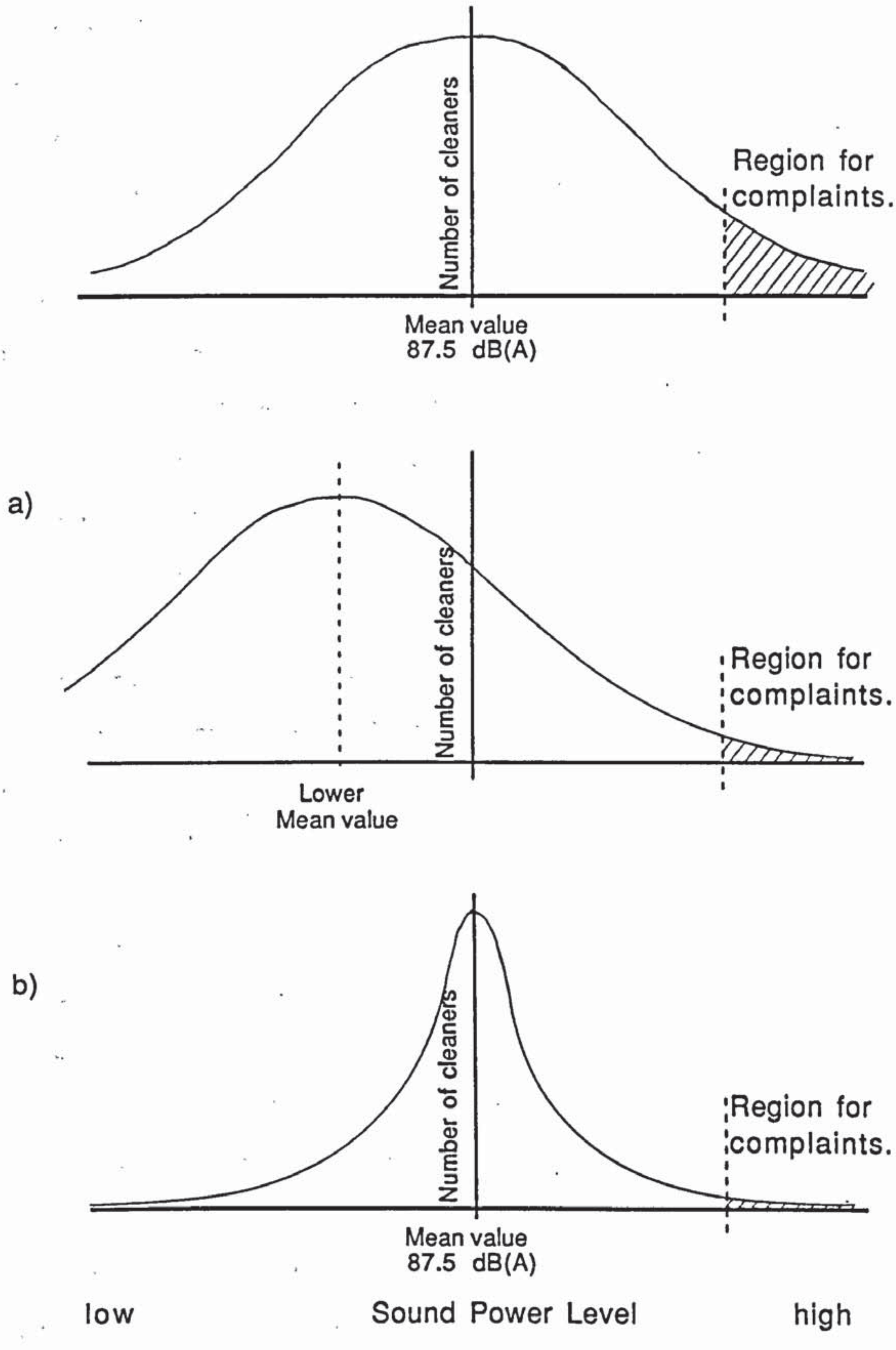


Figure 7.6 Ways of reducing the number of complaints due to excessive noise level:
a) better design but current manufacturing,
b) current design but improved Quality Control.

Higgs (1981) suggested that the shop floor problems which lead to these variations fall into two categories, subjective and objective. The former are associated with the fundamental manufacturing concepts and are common to any industrial organisation. The objective problems are mainly the technical or engineering troubles known to exist on the shop floor.

7.5.1 Objective manufacturing problems

a. Plastic moulding shop

The operators are paid a "piece work" system as an incentive for achieving high production levels. As a result, the machine cycle times and temperature settings are sometimes reduced by the operators in order to increase the rate of production, and hence their pay. Consequently, these mouldings subsequently cool in crates rather than in the tooling cavities. This results in distortions, which cause problems on assembly. Infrequent quality audits by inspectors meant that the operator visually checked the mouldings for burns or other surface defects.

Variations in the moulding thickness are not checked either. In the case of the field winding cage, this caused poor alignment between the field and the armature and resulted in a variation of the rotor/stator air gap, which was shown experimentally to influence the magnitude of the pure tone generated by the chopping effect between the rotor and stator slots. The distortion and the variable thickness of the plastic motor body casings also cause misalignment in the bearing housings and hence the armature shaft; these factors also generate noise.

The suction fan is another item beset by manufacturing problems. The fan is injected moulded in a four cavity tool. Checks on the fan out-of-balance revealed that only 75% of the mouldings are regularly within design limit. The out-of-balance of the fans is not checked on the shop floor. Therefore, it must be accepted that defective fans are assembled into the motor and cleaner. This situation undermines any effort to obtain lower cleaner noise levels.

b. Machine shop

Problems are associated with poor machine tool settings. Although electronic and ordinary gauges are supplied to check the key dimensions, these gauges are not suitable because the gauges were easily knocked out of true or lost and thus some operators do not use them. Also, these gauges are not suitable for identifying tapers or lobing of shaft elements. Lobing is caused by improper centreless grinding operation and is due to the operator malpractice of decreasing the grinding wheel dwell time. The centreless grinding operation is such that the work piece will only be truly round after the dwell cycle has been completed. Lobing and taper causes slop in the bearings and consequently affects the armature unbalance.

Mis-handling of the armature shaft before and between machining operations has been known to cause bending. A bent shaft increases misalignment of bearings and shaft unbalance.

c. Armature manufacture

The production of the armature is fully automatic except for the initial operation where the laminations are pressed on. Most of the problems found were due to defective armature shafts arriving from the machine shop. There were some maintenance problems, such as worn tools, un-cleaned V-blocks and a frayed belt (used in rotating the armature during balancing, causing it to jump or run out of true) which causes eccentricity and bad surface finish. Armatures are dynamically balanced but this operation is followed by a final machining of the commutator.

In many cases, the Quality Inspectors were aware that armatures were being produced which did not meet the design limits, but allowed the situation to persist since the rejection of defective components would have caused a bottle-neck in production. The armature balance limit was originally specified as 0.22 gm-cm, but because production problems led to a bottle-neck, this limit was relaxed to the present value of 0.34 gm-cm. Therefore, product quality considerations had become second to attaining production quotas.

d. Final assembly and test area

Assembly is by operator flow line. Motors are assembled first and then loaded on a rack where they are connected to the mains and allowed to run-in for around 40 minutes. The motors are then removed and tested for sealed suction, open airflow, flash test for the leakage current and quality of commutation. Final assembly of the cleaner is broadly similar to motor production.

The tension of the agitator belt is only checked periodically. Quality control procedures for all bought-in parts need to be improved because, for example, the shore hardness of rubber mounts has never been checked and yet the hardness of mounts was conclusively demonstrated by experiments to affect the noise of the cleaner.

As stated already, the assembled motors and cleaners are tested for performance. Noise level tests for the motor and the cleaner can be implemented at the test stations. However, the initial operator reaction to the inclusion of a noise test was unfavourable because a proposal for any additional test would mean more failures and thus a lower productivity bonus. Defective cleaners are checked and subsequently rectified.

7.5.2 Subjective manufacturing problems

The progressive attitude of the management at Cambuslang was reflected in the concept that operators were responsible for ensuring the quality of their output. This meant the functions of quality control and manufacture are interrelated and not separate. However, quality control was not treated adequately.

a. Quality Control problems

Foremen have a wide freedom to make value judgements. They also have a responsibility to maintain production quotas, consequently, faulty components are often used as a compromise, undermining the Quality Control concept. Bilkhu (1985, p447) suggested that in many instances, batches of rejected components found their way back into the production area. Due to a shortage of Quality Inspectors, audit controls were infrequent allowing such batches to accumulate. Subsequent pressure from the production department led to unchecked or even faulty components to be used, unless evidence of non-acceptability could be provided.

Operators are paid on a "piece work" system. Moreover, they are paid the same rate whether the components produced are good or defective. Often the operator makes his own value judgement as to how serious a particular problem is and whether or not to seek rectification. Of course, if attention is required, then his work stops until repair work and tool setting can be carried out. During this time, the operator is paid a "down-time" rate which is lower than the rate he would have obtained had he continued to produce defective components. Highlighting a problem is therefore, against the operators' interest. In addition, due to a shortage of Quality Inspectors, many of the operators malpractices go unnoticed and there is a consequent increase because the operators realise how vulnerable the system is.

To reduce the variation in the factors which influence noise of the cleaner, the Quality Control function must be authoritative and well staffed with personnel who are familiar with the purpose of noise control treatments.

b. <u>Fault prevention problems</u>

In many instances, poor tool setting and maintenance of machines are the cause of defective components because of the ad-hoc approach. Maintenance staff were summoned when a machine was identified to be faulty or needing repair. Inevitably, poor quality components were produced before any action could be taken. Furthermore, there were no records kept of checks on tool setting or machine condition.

The quality and training of the tool setters is also of importance. Bilkhu noted that the shop floor operators with the longest service were promoted to a machine setter, with total disregard to their aptitude or skills. Thus, the majority of the machine setters may lack the necessary skills to use the tool setting equipment, gauges and to read production drawings. They appear to have little appreciation for the effects of the manufacturing variations on the cleaner assembly and consequent performance.

7.6 The handbook of noise control guidelines to fill a gap in knowledge

Noise reduction in engineering was one of the four main problems in industry identified by a recent survey done by the Institution of Mechanical Engineers (1986a). This underlines the fact that many design engineers do not have the necessary knowledge of the basic principles of acoustics to arrive at an optimum solution to a noise reduction problem. Hoover plc designers are not alone: the need to train design engineers in acoustics and vibration was identified by Lyon (1978). Lyon also suggested cross-fertilisation of the noise control methods from the high technology industry, such as aerospace, to the appliance industry.

A vast pool of the relevant information is available in a growing number of publications. Nevertheless, the Institution of Mechanical Engineers (1986b) identified a need for,

> "a handbook that is well signposted and cross-referenced which could be accessed directly to solve noise problems because the designers did not want or have the time, to read a textbook from which they could pick out or derive the facts and formulae".

Consequently, a handbook was required to fill a gap in knowledge by providing the most relevant information through a collection of commonly used concepts, facts and guidelines. Or alternatively, as expressed by the Chief Engineer at Hoover plc in Sagoo (1984);

> "a guide or designer's manual explaining the doctrine of a Quiet Vacuum Cleaner would be really beneficial".

7.6.1 Aims of the handbook of noise control guidelines

To help the designer to develop acoustical features in order to reduce the airborne noise and meet the criteria for annoyance, speech interference and the proposed noise legislation, the handbook contains;

1. an introduction to acoustics and vibration
2. procedures for predicting the sound level of primary sources
3. methods to estimate the effectiveness of noise control treatments
4. a list of guidelines for good acoustical design practices
5. a case study; application of above techniques and treatments.

In order to restrict the size of the handbook to a manageable length, a compromise on the range and depth of the contents was inevitable. The range of topics, therefore, was suited to the particular needs of Hoover plc design engineers and specifically inclined towards vacuum cleaner design. The handbook was intended to complement, rather than compete with available literature. Subsequently, the main aim of producing this handbook became an original attempt to transfer new technology into Hoover plc. The handbook of guidelines is attached in Appendix 4.

7.6.2 Summary of feedback from Hoover Design Engineers

In its first draft form, this handbook of guidelines was structured as follows;

Part 1: Noise reduction summary and recommendations.

Part 2: Fundamentals of acoustics.

Part 3: Noise level estimation and reduction.

The initial layout was deliberately unconventional. The writer's aim was to present the principles and recommendations of good acoustical design at the beginning in an abbreviated form to enable the designer to save time. It was hoped to expand the content by evaluating equations and comparing with measured data where possible. A brief summary of Chapter 5 would form the case study. In the event, the overall concensus was a desire for a return to a conventional layout. The significant points arising from the feedback were;

- the order of main text should be conventional, that is, an introduction, the body of text and a final summary with recommendations
- a need for careful wording, especially of the recommendations, to avoid mis-interpretation
- that a clearer interpretation of the analytical content would come from more detailed explanation or examples,
- practical examples should be given of the use and limitations

of the various techniques outlined.

7.7 Proposal for modifications to Hoover plc's design policy

To quieten noisy products by design demands a creative and a seriously considered approach based on human perception. Since, Guz et al (1983) found that annoyance was a more sensitive measure of the subjective reaction than complaints, the designer should carry out the objective and subjective research to predict whether or not the populace would be annoyed when exposed to a specific noise source. Therefore, a new design policy should co-ordinate the marketing, design, development and production functions with a unifying aim of designing-out noise.

7.7.1 Marketing

The design process begins and ends with the customer. Consequently, market research is a valuable source of feedback. Plunkett (1958) reasoned that,

> "earlier vacuum cleaners were expected to have a powerful sound, otherwise it would be suspected of not doing its job".

In a more sophisticated society, however, it has to be expected that noise cannot add sales appeal. According to Clayton (1982),

> "in the highly competitive market of domestic appliances a product won't sell properly if it makes more noise than the salesman".

In a noise reduction project, the unknowing elimination of customer feedback may result in an "overkill", which may be equally catastrophic as excessive noise levels. For instance, Fleischman (1983) noted that a replacement of a noisy bearing in a prosthetic elbow resulted in the user experiencing great difficulty in manipulating the quiet elbow, and Horvat (1984) pointed out that a salesman encountered disappointment on the faces of potential customers when unveiling a company's latest super-quiet diesel engine. "How can it have any power if it makes no noise?". The essential feedback was missing. A similar fate may await a super-quiet vacuum cleaner.

Therefore, although the design or the product can be an engineering success from a technical standpoint, the product may become a failure for human-factor reasons. Psychological meaning is embedded in sound. Some sounds are perceived as belonging to products with a high quality or status image. Therefore, a more structured market research is required to determine, in detail, the requirements of a customer in terms of the acoustics information. The analysis should account for a "description" of the emitted sound; that is, purr/hum, quality/cheap, rough/smooth, simple/complex, clunky/tinny/ squeaky in addition to loud/soft. These subjective impressions of noise are quantifiable and their knowledge can form an important input to the design process.

7.7.2 Design

A qualified noise and vibration engineer should be added to the design and development team. He should co-ordinate other departments for an "inter company" noise control effort. He should,

- estimate the sound power level of each major noise source, and consider from the outset how the estimated noise can be reduced or eliminated and suggest appropriate methods of noise control
- anticipate the environmental effects of noise when the product is in operation
- supervise frequent checks to evaluate the noise level, from the initial design stage through to production and after sales.

Alternatively, designers should be given training to appreciate the principles of noise and vibration control.

7.7.3 Development

In the writer's opinion, the A-weighted sound level is not the best scale for evaluating cleaners. This is because the components of discrete frequency are not represented clearly. Discrete frequency components cannot be ignored, because a cleaner having a lower overall noise level may actually be more annoying than a cleaner of a higher noise level. For these reasons, the "sone" or the "noys" scale should be used frequently during development to estimate the subjective response.

The development department should also,

- evaluate the first production models and design-in further noise reducing treatments, if required,
- test a production sample and identify the major noise sources by using the techniques and following the flow-chart in Figure 7.7,
- evaluate the contribution of manufacturing tolerances to the variation in the motor and cleaner noise level, and if necessary, take steps to correct machining operations and operator malpractices to ensure that the essential parameters which influence noise are kept within the design limit,
- create and up-date a database of major noise sources,
- create and up-date a database of noise reduction treatments and benefits.

7.7.4 Production

The combined efforts of good acoustical design, low noise sources and a proper choice of noise control treatments, can all be nullified by improper installation of the treatment or poor quality control. Therefore the Production Department should;

- train all key production and Quality Control personnel to understand the value and the purpose of noise and vibration control principles,

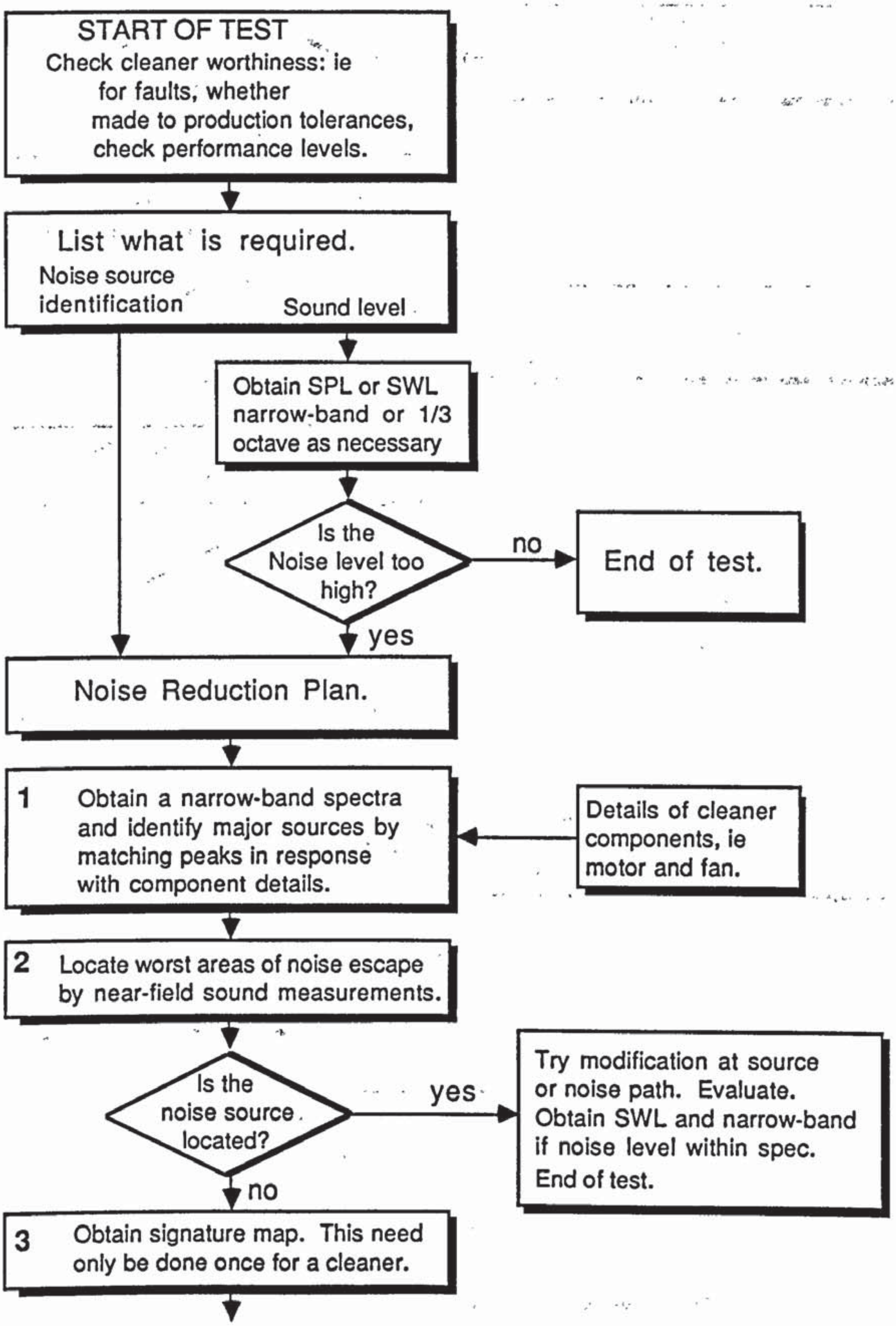


Figure 7.7 Proposed Noise Control Programme.

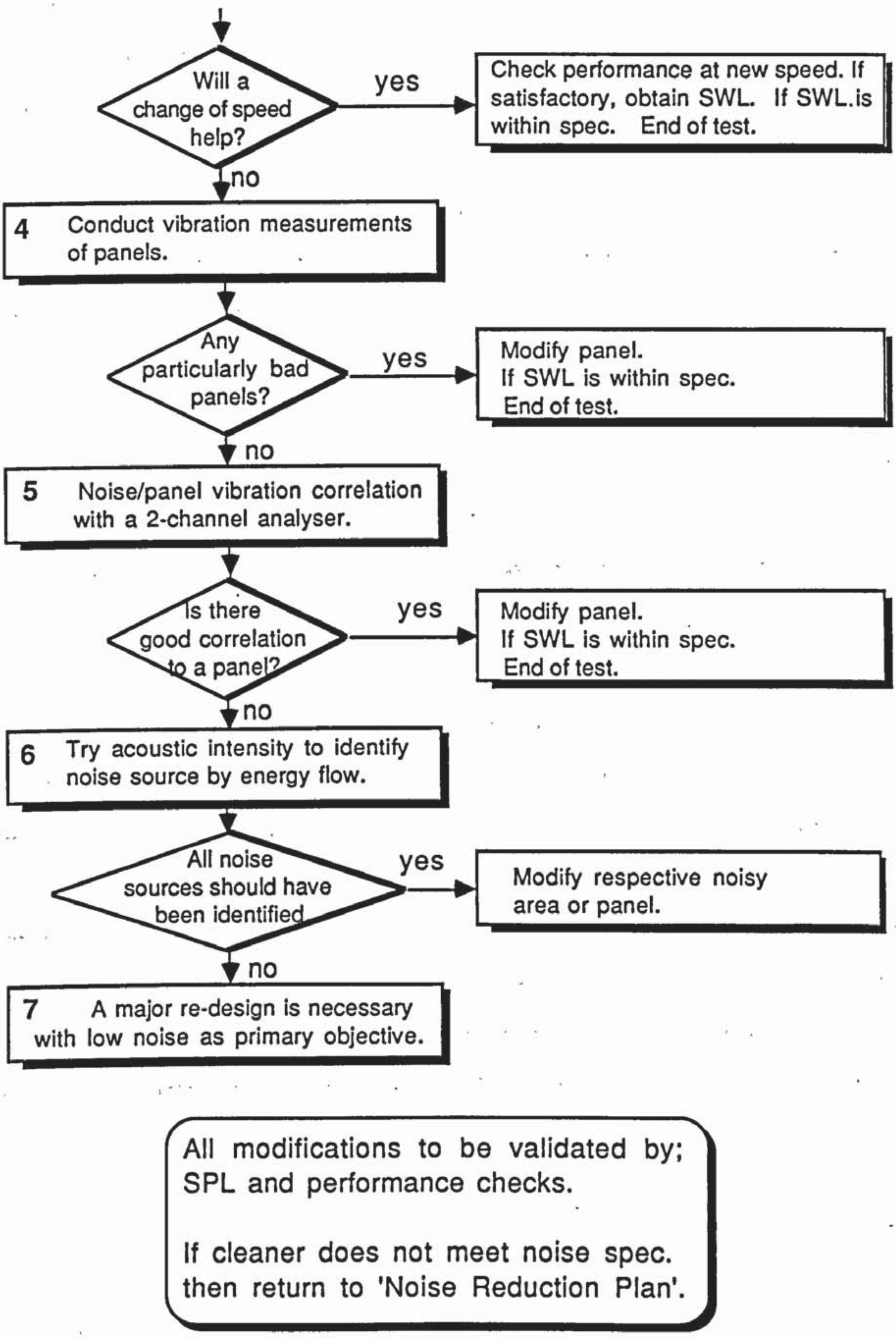


Figure 7.7 (con'd)

- maintain effective quality control procedures during production,
- ensure adequate number of Quality Inspectors,
- introduce preventative maintenance of machines to help identify machine and tool problems at an early stage and reduce the production of defective components,
- introduce record keeping of major tool changes to complement preventative maintenance.

7.8 Review

This chapter presented the elements of good acoustical design practices together with a discussion of the importance of manufacturing tolerances on the overall noise level of the cleaner. The role of the acoustics engineer, therefore, has to change from that of trouble-shooter to co-ordinator: this role has to extend through to customer liaison.

A review of Hoover plc's current design policy revealed that designers lacked the vital knowledge of the fundamentals of acoustics and noise control treatments, while there was little or no appreciation of the importance of production and quality control in relation to noise reduction. A handbook of design guidelines was, therefore, compiled to fill this gap in knowledge. The advent of a quiet or a low noise vacuum cleaner has to rely equally on Marketing, Design, Development and Production Departments, because the lack in co-ordination of these departments can nullify the noise reduction effort.

CHAPTER EIGHT: CONCLUSIONS AND RECOMMENDATIONS

Contents

8.1 Introduction

The research project which has been described was concerned with the reduction of noise from an upright vacuum cleaner. Factors that affect the generation of noise within the motor, the fan and the cleaner structure were studied. All of the noise reduction treatments were evaluated on a Turbopower hard-bag cleaner. The main conclusions from the experimental work are presented in section 8.2. The important points in designing for quietness were also evaluated and the conclusions resulting from this study are presented in section 8.3. Some topics for further research have been identified but could not be included in the time scale of the present project. These topics are suggested for further research in section 8.5. Finally, some concluding comments on the relevance of interdisciplinary "action research" in industry are presented in section 8.6.

8.2 Conclusions from the experimental work

The experimental work covered the following areas;

a) evaluation of the Perivale acoustic room,

b) application of noise source identification techniques,

c) noise reduction methodology,

d) the "Quiet Vacuum Cleaner".

The conclusions are as follows;

8.2.1 Acoustic room, Perivale

The Perivale acoustic room is too small and not adequately isolated from the rest of the building. As a result, its structural characteristics distort the frequency response measurements by attenuating some frequency components and amplifying others. Thus, in the past, the noise control effort became focused on frequency elements which were not the main source of noise. If and when a replacement test room is developed at Cambuslang it would be highly desirable for it to be built to a specification that would enable more reliable results. It would be particularly beneficial for sound measurements in narrow-band frequency to be directly compared with those from any other approved laboratory, such as the acoustic room at Hoover Inc. North Canton, USA.

8.2.2 Noise source identification techniques

A number of noise source identification techniques were used. Many of these techniques were found to be useful and provided information to pin-point the noise sources and, more significantly, to promote a better understanding of noise generating phenomenon. Vibration testing, however, was found to be ineffective in the present case because of the inherent damping in plastics and the change in the boundary conditions when lightweight components are assembled. Signature analysis and the coherence function were, so far as the author is aware, applied to a vacuum cleaner for the first time. The following techniques, many of which had not been used by Hoover engineers before, were employed in this study;

- near-field sound measurements,
- directivity of sound sources,
- signature analysis, (a paper on this technique was presented at the Inter-Noise 86 conference at MIT, Boston, USA),
- narrow-band vibration measurements of major components,
- coherence function measurements.

8.2.3 Noise reduction methods

1. The armature out-of balance was found to cause a large variation in the noise level of the cleaner, and hardly any spread in the noise level of the motor. The suction fan was found to be the cause of this problem and a new method of fan installation was

proposed. There should be no machining operation following the the balancing of the armature.

2. The transmission of vibration from the motor body to the chassis and then to the rest of the cleaner structure, was found to be important. The use of softer rubber mounts helped in two ways. Firstly, the overall noise level was reduced and, secondly, this reduction was proportional to the armature imbalance. This solution, therefore, reduced the variation in the cleaner noise levels and as a result, may limit the maximum noise of a "freak" cleaner. A noise ceiling of 88.0 dB(A) SWL is feasible through the use of softer mounts and the isolation of the hood.
3. The vent fan end plane imbalance was found to influence the noise level more than the suction fan end plane. Although, the value of imbalance at the suction fan end was greater than that at the vent fan end, the transmission path at vent fan end was more important.
4. The hood was found to be a significant secondary noise source on cleaners which had a high armature imbalance. For instance, noise for a particular cleaner reduced by 4 to 5 dB(A) by removing the hood. The isolation of the hood by rubber mounts was beneficial.
5. Elimination of the direct transmission path between the agitator and the chassis reduced the low frequency noise.
6. A reduction in the motor speed from 285 Hz (17100 rpm) to 270 Hz (16200 rpm) reduced the noise level without a significant penalty in the performance.
7. Although quality bearings did not prove beneficial, their use is recommended because when the cleaner noise level is lowered, the bearing noise will become more prominent. Quality bearings are made to closer tolerances and from better materials, therefore, their use will enhance the cleaner's refinement and reputation.

8. The air-flow efficiency of the cleaner was enhanced by reducing the air leakage. The air-flow efficiency of the cleaner rose from 14.5% to 17.6%. A further increase to about 19% is feasible by reducing the size of the bleed hole.
9. Preventing the exhaust air flowing over sharp edges on the bag housing covers resulted in noise reduction of 2.1 dB(A).
10. An adjustable furniture guard was beneficial in restricting the noise level of the cleaner especially at higher nozzle settings.
11. The upright of the Turbopower hard-bag cleaner was shown to be a "silencer", because there was no benefit offered by the add-on silencers.
12. The use of damping and sound deadening materials was successful. Therefore, space for these materials should be allowed early in the design stage.
13. The probability of component "rattle", for instance wheel-trims, should be avoided by interference fit. Cleaner noise was reduced by 1 dB(A) by this simple and cost-effective modification.
14. "Sound transparent" areas or "windows"; that is, areas around a cleaner where sound propagates without much absorption or attenuation, should be avoided. The bellows and the hood orifices were the main such areas. Sound insulation applied to cover the bellows reduced the overall noise level by 3.5 dB(A).

8.2.4 The "Quiet Vacuum Cleaner"

1. The sound power level of the "Quiet Cleaner" was 80.4 dB(A) compared with the average level of 87.5 dB(A) for a standard cleaner.
2. The near-field sound levels for the "Quiet Cleaner" were much lower than the standard cleaner.
3. There was virtually no penalty in the performance of the cleaner.

4. Perceived noisiness measured in "noys", a linear scale, was found to expand the rating even more than the "sone" scale. Therefore, the "noys" scale is proposed for noise-labelling.
5. The speech interference level for the "Quiet Cleaner" was far better at 68.8 dB, against 74.7 dB and 75.5 dB for the noisy and standard cleaners, respectively. Therefore, a conversation with normal voice is possible beyond a distance of 1 metre when operating the "Quiet Cleaner".
6. The subjective assessment by an audience at Hoover plc and Aston University supported the above findings.

8.3 The conclusions from the study of designing for quietness

1. Designers were not necessarily familiar with the fundamentals of acoustics and vibration. They were, therefore, unfamiliar with noise reducing treatments and so gave a lower priority to noise control.
2. The handbook of design guidelines was produced to fulfil one of the long term objectives of the project. It also filled a gap in the designers' knowledge.

8.3.1 The conclusions from the study of manufacturing problems

1. There was a lack of adequate Quality Control procedures during production,
2. There was inadequate maintenance of machines and tools,
3. Operators and foremen do not appear to appreciate the consequences of using defective components upon the attributes of the final product.

8.4 Recommendations:

1. The design process should include designing for lower noise as

follows;

a. Predict sound levels for the major noise sources, and study noise reduction at the source wherever possible,

b. Estimate the probable noise attenuation or absorption due to standard noise treatments such as, plastic sheet and sound absorbent materials, and use features such as directivity and proper sealing around any noise source.

c. Conduct comprehensive tests to verify the noise treatments against noise targets during development and pre-production,

2. A database of sound power levels for the motor, cleaner, noise treatments and materials should be compiled and frequently up-dated and used as input to the design procedure.
3. Obtain detailed information on sound "descriptors", through the Marketing Department and use this information in design to "tune" the sound of a product to requirements. Thus, the company will place the customer at the start and the end of the design process.
4. Instigate an active and prominent role for Quality Control,
5. Offer all key personnel a brief training in aspects of acoustics and vibrations and the importance of manufacturing variations to the overall noise level,
6. Implement preventative maintenance for all machines and tools.

These factors should be refreshed and up-dated as appropriate.

8.5 Suggestions for further work

The topics outlined below are suggested as meriting further investigation.

8.5.1 Motor and agitator on a separate sub-base

The isolation of the motor and agitator vibration resulted in a reduction in the overall noise level and also reduced the noise variation. Following this line of thought and using an analogy of a motor car, all the "running gear" should be assembled on a separate sub-base. All other components can then be mounted on the base by "tuned" rubber mounts. As a result, the rest of the cleaner structure will not be excited by high levels of vibration and therefore costly, add-on noise treatments may be avoided.

8.5.2 Jet of air to disturb the debris and therefore enhance the agitator action

It was mentioned in Chapter 3 that, for a similar air-flow in a duct of similar dimensions, the velocity profile of a jet of air extends further than suction by a factor of about 40. The jet can be created by simply collecting and re-routing the exhaust air. A thin oblong jet, or a series of small jets, can be positioned just behind the agitator to dislodge deep-rooted dust which can then be sucked up in the normal way.

8.5.3 Use of alternative materials

It was not possible to obtain the sound transmission loss data for a selection of common plastic materials. This data is, however, considered to be important because it may highlight materials that have better sound attenuating properties. For instance, rubber based compounds may be better than acrylics. The increased cost of a better material may be justified by reduced add-on noise treatments. Although, the mechanical properties of new materials will dictate their use, a compromise may result in the adoption of alternative features, such as "sheen" surface finish, extra flexibility and non-scratch durable surface. Considerable exciting openings are envisaged and indeed may be offered by new materials.

8.5.4 Market research using sound "descriptors"

The significance of the description of sound is reflected in the way a customer perceives or values a product within a particular market. For instance, the door of an expensive car is expected to shut with a "thud" or "clunk" and not a "clang". Hence, the terminology implies quality/cheap ideals. Consequently, this kind of knowledge of the market can be used in product design specifications to "tune" the sound output. A benefit to the company ensues as a consequence of refined products, the company being regarded as more modern by its customers.

8.5.5 Evaluation by Finite Element and Modal Analyses and their integration into the design process

Finite element, FEA, and modal analyses have been in use for some time. These were normally applied separately; that is, FEA is usually applied at the design stage to check stresses in the components, whilst modal analysis is conducted on a prototype or a production model to determine the resonances and the mode shapes experimentally. The resulting modifications are usually evaluated by trial and error. These methods are increasingly being replaced by structural dynamic modification packages which provide the effect on the system frequency response due to changes in mass, stiffness or damping. Moreover with the reduction in the cost of computers and software and the advances in techniques and programming the FE and modal analyses are now being integrated and introduced to desk-top machines. Such an integrated package is considered to be an essential tool for the acoustics designer. The package would save time and cost as shown in Figure 8.1. Therefore, further work in evaluating such a package is suggested.

8.5.6 To optimise the motor and hood mounting

Further work should be undertaken to optimise the motor and hood

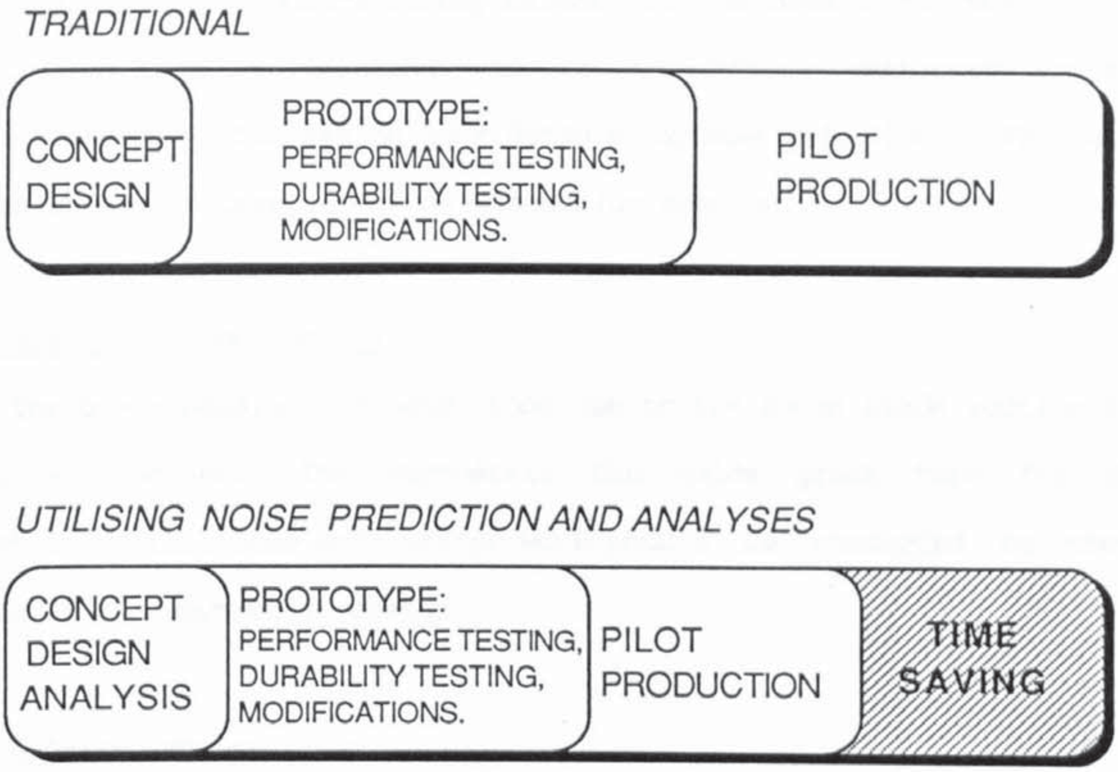


Figure 8.1 Potential time savings in the product development cycle by utilising noise prediction and analyses procedures.

mounting on the chassis by determining the mode shapes of the chassis. The sub-system model to optimise the motor mounts needs to be studied further and verified experimentally.

8.5.7 Post-assembly motor balance

The need for a post-assembly balance of the Turbopower motor, may be eliminated after implementing the recommendations mentioned in the previous section. Thus saving considerable expense and effort. This work requires extensive development on production samples.

8.5.8 Asymmetric suction fan

The blade passing frequency tone due to the seven blade suction fan was quite prominent. The asymmetric fan holds great hope for the elimination of this tone and further work should be conducted to check benefits of the asymmetric design.

8.6 Closing remarks

Perhaps the only knowledge of interest to pure research is that which was previously unknown to mankind. In applied research, however, the area in which the knowledge is sought and the purpose to which it will be put, are closely defined. Research in a manufacturing organisation is rarely aimed at creating new knowledge, more frequently it seeks to introduce existing knowledge to the activities within the organisation in an original way, in simple terms, to transfer the technology. That was the context of this research project.

Projects involving noise reduction are not new, nor are the noise source identification techniques, reduction treatments or the acoustical design practices. Nevertheless, these techniques and treatments have been successfully applied to reduce the noise level of an upright vacuum cleaner. In addition, good acoustical design practices have been proposed so that the next generation of products will benefit from this research at an early stage. Furthermore, the importance of market research for the initial information and frequent feedback to close the design-cycle and the importance for good manufacturing of the Quality Control procedures during production were recognised during this project.

It was found that a high noise level is a not a good feature of a product. Noise is quality related and the relation between quality and noise must be appreciated on the production line and employed as a test to discriminate between good and bad cleaners. Noise control should also be considered from another angle. As noise from most appliances cannot be totally attenuated, its characteristics should be "designed" so that the sound is pleasant to the user.

APPENDIX 1: Photographs

A1.1 The Hoover Turbopower vacuum cleaner

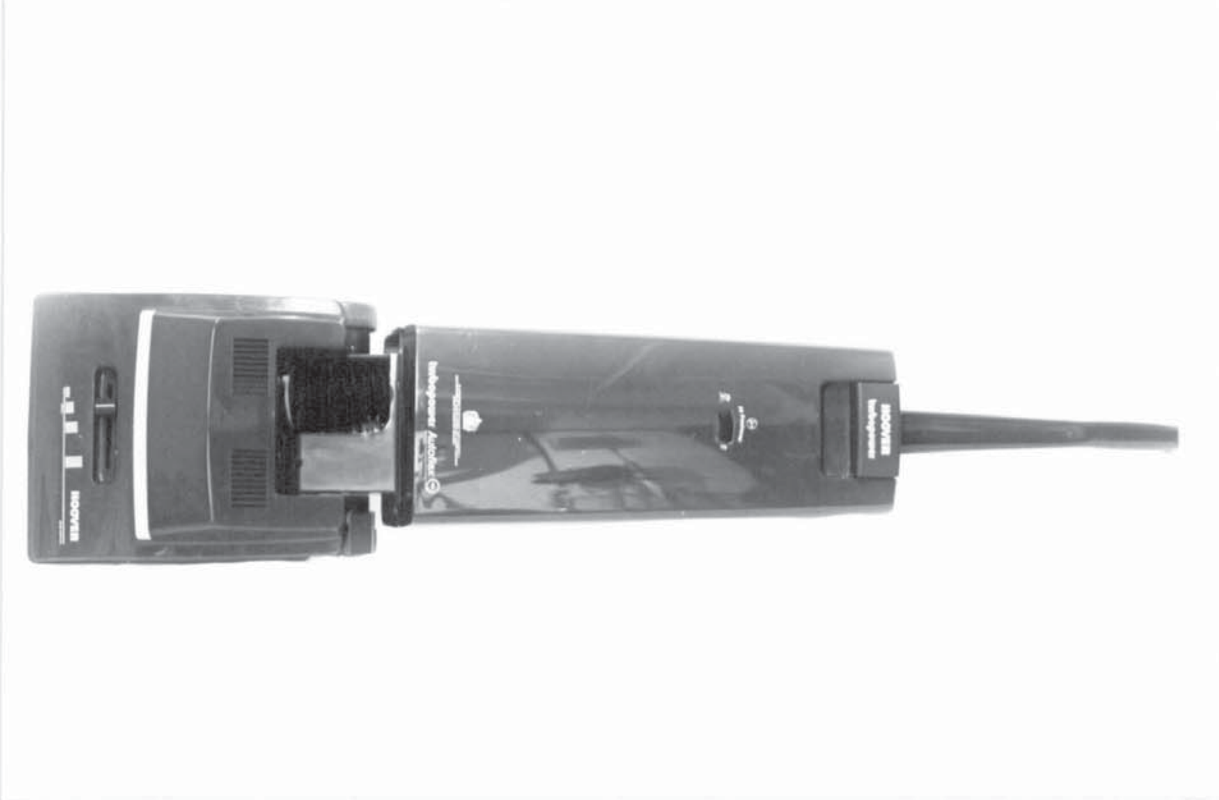
A1.2 The upright of the Turbopower cleaner reclined to show the bellows and the openings in the hood

A1.3 The EURO motor

A1.4 The armature assembly

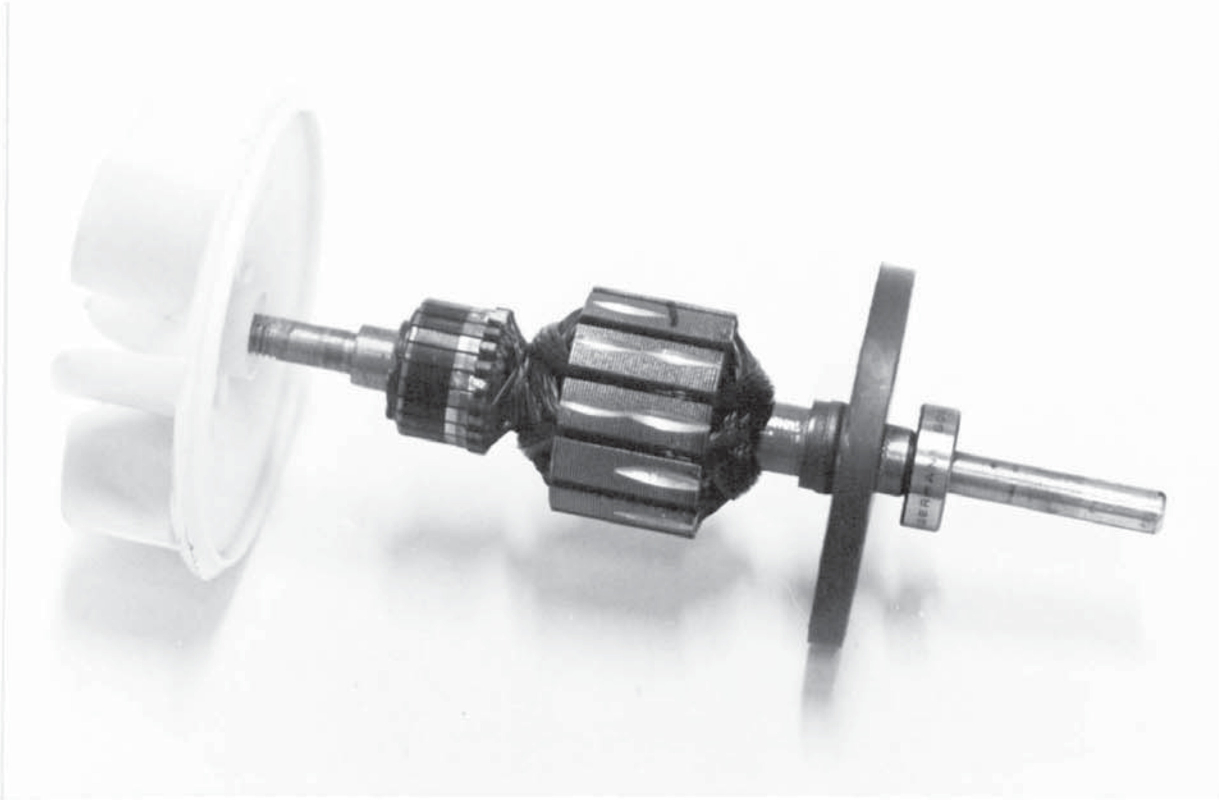
A1.5 Suction fan with a plastic cover on the blade tips (refer to section 5.8.8)

Photograph Al.1 The Hoover Turbopower vacuum cleaner

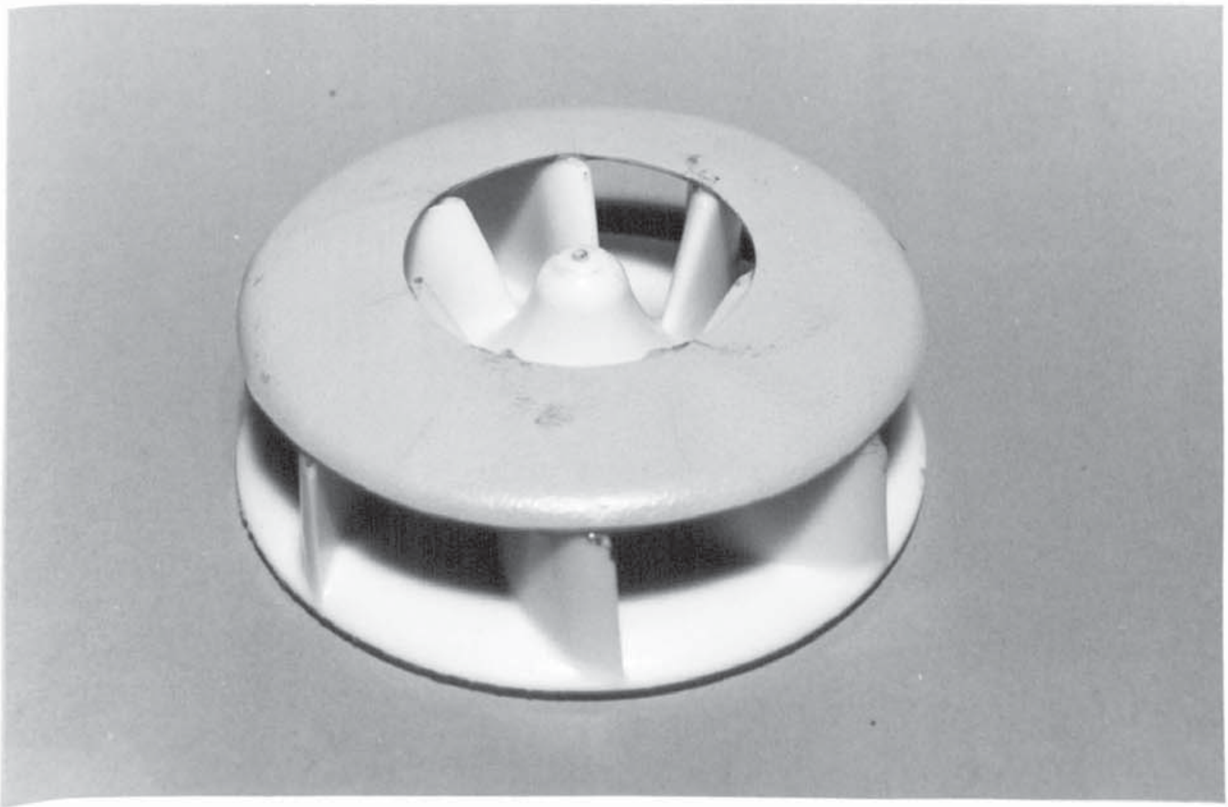


Photograph A1.2 The upright of the Turbopower cleaner reclined to show the bellows and the openings in the hood

Photograph A1.3 The EURO motor



Photograph A1.4 The armature assembly of the EURO motor



Photograph A1.5 Suction fan with a plastic cover on the blade tips (refer to section 5.8.8)

APPENDIX 2

IDENTIFYING NOISE SOURCES IN A VACUUM CLEANER

Paper presented at the "Inter-Noise 86" International Conference held at the Massachusetts Institute of Technology, Cambridge, Boston, USA, 21-23 July 1986.

APPENDIX 3.

RIGID BODY AND SUB-SYSTEM MODELS OF MOTOR ON A CHASSIS.

NOTATION:

a	distance from centre of gravity of motor to fan end mount
b	distance from centre of gravity of motor to belt end mount
c	distance from centre of gravity of motor to axis of agitator belt
F	agitator belt tension
I	moment of inertia of motor
K	stiffness of mount
m	mass of motor
M	mobility
ω	frequency in Hz
v	velocity
x	vertical displacement of motor at its C of G.
X	maximum value of x
θ	angular rotation of motor at its C of G.
Ω	maximum value of θ

Subscripts:

F	suction fan end mount
B	agitator belt end mount
c	chassis
i	imaginary part
m	motor
r	real part
s	isolator

RIGID BODY MODEL

Pitch Frequencies.

As a consequence of the agitator belt and its tension, the forces in-line with the belt will be different to the transverse plane. Therefore there are two cases for consideration:

Case 1; in-line with agitator belt
Case 2; transverse to belt.

Assuming small oscillations and no damping, for case 1 the forces acting on the motor are;

$$Mx = -K_F (x+a\theta) - K_B (x-b\theta) - F \qquad \text{....1}$$

$$I\theta = K_B (x-b\theta)b - K_F (x+a\theta)a + Fc \qquad \text{....2}$$

for small oscillations let $x = X \sin\omega t$ and $\theta = \Omega \sin\omega t$

subs. $(K_F + K_B - M\omega^2)\,x + (aK_F - bK_B)\,\theta = -F$

$(K_Bb^2 + K_Fa^2 - I\omega^2)\,\theta + (K_Fa - K_Bb)\,x = Fc$

In matrix form,

$$\begin{bmatrix} K_F + K_B - M\omega^2 & aK_F - bK_B \\ aK_F - bK_B & b^2K_B + a^2K_F - I\omega^2 \end{bmatrix} \begin{bmatrix} X \\ \Omega \end{bmatrix} = F \begin{bmatrix} -1 \\ c \end{bmatrix}$$

Natural frequencies are given by the solution to the square matrix. Therefore,

$$(K_F + K_B - M\omega^2)(\,a^2K_F + b^2K_B - I\omega^2\,) - (aK_F - bK_B)^2 = 0$$

This is a simultaneous equation and $\omega^2 = \dfrac{-B +/- \sqrt{B^2 - 4AC}}{2A}$

where
$A = MI$
$B = -[M(a^2K_F + b^2K_B) + I(K_F + K_B)]$
$C = (K_F + K_B)(a^2K_F + b^2K_B) - (aK_F - bK_B)^2$

For case 1: $K_F = 320.0$ N/mm
$K_B = 151.0$ N/mm
$I = 2.306$ E-3 kgm^2
and $M = 1.486$ kg, $F = 170$ oz $= 4.743$ kg.
$a = 95.3$ mm, $b = 76.2$ mm, $c = 30.3$ mm.

Solutions are: <u>ω = 1309.9 Hz and 491.7 Hz.</u>

For case 2, that is the plane transverse to agitator belt, there is no external force acting on the motor. Natural frequencies are given by the solution to the same square matrix albeit with different quantities, as follows;

$K_F = 251.14$ N/mm
$K_B = 193.65$ N/mm
$I = 2.453$ E-3 kgm^2

Solutions are: <u>ω = 1187.0 Hz and 527.7 Hz.</u>

<u>Rigid Body Bounce Frequencies.</u>

Natural frequency is given by $\omega_n = \sqrt{K/m'}$ where m' is the 'effective force' acting on the mount.

Bouncing on fan side mount; the effective force is obtained by taking moments about the motor attachment point at the other mount. For the in-line case , situation with and without the belt tension were considered.

With belt, $m'(a+b) = mb - Fc$

$m' = 0.1777$ kg

Bounce frequency is $\omega^2 = 320.0\text{ E3}/0.178$

$\underline{\omega = 1341 \text{ Hz.}}$

Without belt, $m'(a+b) = mb$

$m' = 0.66$ kg

Bounce frequency in-line: $\underline{\omega = 696.3 \text{ Hz}}$

transverse: $\underline{\omega = 616.9 \text{ Hz.}}$

Bouncing on belt side mount;

With belt, $m'(a+b) = ma + F(a+b+c)$

$m' = 6.407$ kg

Bounce frequency is $\omega^2 = 151.0\text{ E3}/6.407$

$\underline{\omega = 153.5 \text{ Hz.}}$

Without belt, $m'(a+b) = ma$

$m' = 0.826$ kg

Bounce frequency in-line: $\underline{\omega = 427.6 \text{ Hz}}$

transverse: $\underline{\omega = 484.3 \text{ Hz.}}$

The moment of inertia of the motor assembly was determined in the usual manner using a trifilar suspension. Details were as follows:

length of wires	= 1.631 m
radius of the pitch of wires	= 77 mm
base was a steel plate; thickness	= 3.23 mm
outside diameter	= 0.151 m
inside diameter	= 0.045 m
mass	= 0.426 kg.

Duration for base plate alone: 88.9 sec for 50 oscillations,
motor in-line to belt: 71.45 sec "
motor transvers to belt: 72.94 sec ".

VIBRATION ISOLATION BY SUB-SYSTEM MODEL

The effectiveness of an isolator can be defined as the ratio of the force on chassis with to without the isolator. This ratio is termed force transmissibility;

$$T_f = \frac{F_c^{isol}}{F_c^{rigid}}$$

Alternatively, the insertion loss IL can be used to define the isolator effectiveness;

$$IL = 10 \log_{10} \left[\frac{v_c^{rigid}}{v_c^{isol}}\right]^2$$

Without an isolator.

Without an isolator the following relations are true;

$$v_m^{rigid} = v_c^{rigid} \quad \text{and} \quad F_m^{rigid} = -F_c^{rigid}$$

By superposition, the net motion of the attachment point is the sum of the motion it would have due to internal forces of the motor (freely hung) and motion due to the resisting force (see also Muster and Plunkett in Beranek's 'Noise and Vibration Control' (1971));

$$v_m^{rigid} = v_m^{free} + M_m F_m^{rigid}$$

Sub for F_m^{rigid} and $F_c^{rigid} = \dfrac{v_c^{rigid}}{M_c}$

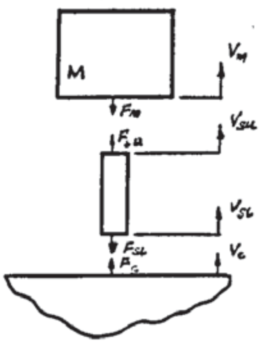
$$M_c F_c^{rigid} + M_m F_c^{rigid} = v_m^{free}$$

$$F_c^{rigid} = v_m^{free}\left[\frac{1}{M_c + M_m}\right]$$

or,

$$v_c^{rigid} = v_m^{free}\left[\frac{M_c}{M_c + M_m}\right]$$

With an isolator. The following properties hold;



$$F_c^{isol} = -F_{s\,lower} \quad \text{and} \quad F_m^{isol} = -F_{s\,upper}$$

$$F_s = F_{s\,upper} = -F_{s\,lower}$$

$$v_s = v_{s\,upper} - v_{s\,lower}$$

$$v_c^{isol} = v_{s\,lower} \quad \text{and} \quad v_m^{isol} = v_{s\,upper}$$

Using superposition as before;

$$v_m^{isol} = v_m^{free} + M_m F_m^{isol}$$

by subs,

$$v_s = v_m^{free} + M_m F_m^{isol} - v_c^{isol}$$

$$M_s F_s = v_m^{free} + M_m(-F_c^{isol}) - M_c F_c^{isol}$$

It follows, $$F_c^{isol} = v_m^{free}\left[\frac{1}{M_c + M_m + M_s}\right]$$

or, $$v_c^{isol} = v_m^{free}\left[\frac{M_c}{M_c + M_m + M_s}\right]$$

The insertion loss is given by;

$$IL = 10 \log\left[\frac{M_c + M_m + M_s}{M_c + M_m}\right]^2 = 20 \log\left[\frac{M_c + M_m + M_s}{M_c + M_m}\right]$$

Force transmissibility is given by;

$$T_f = \left[\frac{M_c + M_m}{M_c + M_m + M_s}\right]$$

COMPUTER PROGRAM FOR INSERTION LOSS AND FORCE TRANSMISSIBILITY

We need to determine the expression which is common to IL and T_f. All measured quantities are complex, therefore let,

M_m be replaced by $M_r + iM_i$
M_c " $I_r + iI_i$
M_s " $F_r + iF_i$

therefore, $$\left[\frac{M_c + M_m}{M_c + M_m + M_s}\right] = \left[\frac{F_r + M_r + i(F_i + M_i)}{(F_r + M_r + I_r) + i(F_i + M_i + I_i)}\right]$$

let,

$$a = F_r + M_r$$
$$b = F_i + M_i$$
$$c = F_r + M_r + I_r$$
$$d = F_i + M_i + I_i$$

the above expression reduces to; real part $= \dfrac{ac + bd}{c^2 + d^2}$

imaginary part $= \dfrac{bc - ad}{c^2 + d^2}$

SYSTEM SUB-STRUCTURE COMPUTER PROGRAM.

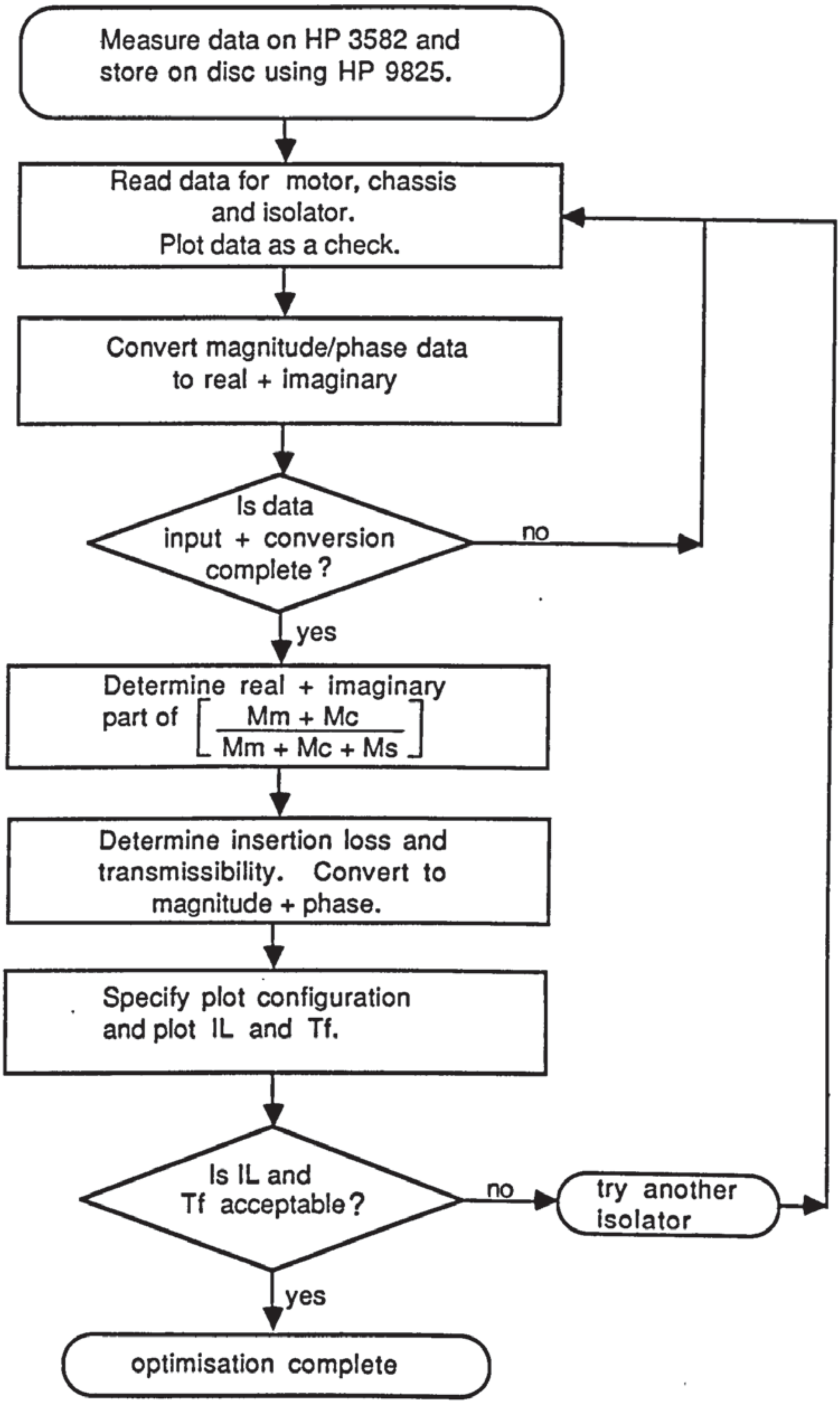


Figure A3.1 Flow chart for the computer program.

```
1      ! SYSMO;  FORCE TRANSMISSIBILITY ACROSS ISOLATOR
2      !    WRITTEN BY  HARDIAL S SAGOO     8 MARCH 1986.
3      !
4      !
5      DEG
10     ! RE-STORE"DREAD:F8"
12     DIM M(130),N(130),F(130),G(130),U(130),W(130),Y(130)
13     DIM A(130),B(130),C(130),D(130),T(130),V(130)
14     GOTO 680
19 Dread:  DIM R(130),S(130)
20     DIM X(500)
21 !   PRINT "FILE No OF MOTOR DATA"
22 !   INPUT A$
23     Filename$=A$
30     ASSIGN #1 TO " "&Filename$
31     PRINTER IS 16
40     FOR I=1 TO 273
50     READ #1;X(I)
70     IF I<>145 THEN 90
80     PRINT "-----------"
90     NEXT I
91     FOR I=18 TO 145
92     J=I-17
93     PRINT J,X(I),X(I+128)
94     NEXT I
100    PRINTER IS 16
110    PRINT PAGE
120    PRINT "    DO YOU WANT TO PLOT THE DATA  Y or N   "
130    INPUT A$
140    IF A$<>"N" THEN 160
150    GOTO 320
160    PLOTTER IS 13,"GRAPHICS"
170    GRAPHICS
180    LIMIT 0,184,0,140
190    LOCATE 20,120,20,80
200    SCALE 18,145,-50,50
210    FRAME
220    FOR I=18 TO 145
230    PLOT I,X(I)
240    NEXT I
250    PENUP
260    SCALE 1,128,-200,200
270    LINE TYPE 6
280    FOR I=146 TO 273
290    PLOT I-145,X(I)
300    NEXT I
310 !  PENUP
320 !  PRINTER IS 16
330 !  PRINT PAGE
340 !  PRINT "   DO YOU WANT A COPY OF THE GRAPH  Y or N    "
350 !  INPUT A$
```

```
360 ! IF A$<>"Y" THEN 380
370 ! DUMP GRAPHICS
380 ! STOP
390   ! TO CHANGE POLAR TO RECT CO-ORS
400   !
410   !
420   ! REAL=TF*COS(PHA)   PHA+X(146) TO X(273)
430   ! IMAG=TF*SIN(PHA)
440   FOR I=18 TO 145
441   K=I-17
450   J=I+128
451   X(I)=X(I)/20
452   X(I)=10^X(I)
460   IF X(J)>180 THEN X(J)=X(J)-360
470   IF X(J)<-180 THEN X(J)=360+X(J)
480   IF (X(J)>=0) AND (X(J)<=90) THEN 520
490   IF (X(J)>90) AND (X(J)<=180) THEN 550
500   IF (X(J)<0) AND (X(J)>=-90) THEN 580
510   IF (X(J)<-90) AND (X(J)>=-180) THEN 610
520   R(K)=X(I)*COS(X(J))
530   S(K)=X(I)*SIN(X(J))
540   GOTO 640
550   R(K)=-X(I)*COS(X(J))
560   S(K)=X(I)*SIN(X(J))
570   GOTO 640
580   R(K)=X(I)*COS(X(J))
590   S(K)=-X(I)*SIN(X(J))
600   GOTO 640
610   R(K)=-X(I)*COS(X(J))
620   S(K)=-X(I)*SIN(X(J))
630   GOTO 640
640   NEXT I
641   RETURN
650   !
660   ! END OF POLAR TO RECT
670   ! THIS GETS MOTOR DATA
680   PRINTER IS 16
690   PRINT "FILE No OF MOTOR DATA"
700   INPUT A$
710   GOSUB Dread  ! TO READ AND CHANGE POLAR TO RECT
711   PRINTER IS 16
720   FOR I=1 TO 128
730   M(I)=R(I)
740   N(I)=S(I)
741   IF I>15 THEN 760
742   PRINTER IS 16
743   IMAGE DDD,3X,MDD.DDE,3X,MDDD.DE,3X,MDDD.DE,3X,MDDD.DE
750   PRINT USING 743;I,X(I+17),X(I+145),M(I),N(I)
760   NEXT I
770   !
780   ! THIS GETS THE CHASSIS DATA
```

```
790   PRINTER IS 16
800   PRINT "FILE No OF CHASSIS DATA"
810   INPUT A$
820   GOSUB Dread
830   FOR I=1 TO 128
840   F(I)=R(I)
850   G(I)=S(I)
851   IF I>15 THEN 870
860   PRINT USING 743;I,X(I+17),X(I+145),F(I),G(I)
870   NEXT I
880   !
890   !
900   ! THIS GETS THE ISOLATOR DATA
910   PRINTER IS 16
920   PRINT "FILE No OF THE ISOLATOR DATA"
930   INPUT A$
940   GOSUB Dread
950   ! USE R(I) AND S(I) FOR ISOLATOR ARRAY
960   FOR I=1 TO 15
970   PRINT USING 743;I,X(I+17),X(I+145),R(I),S(I)
980   NEXT I
990   !
1000  !
1010  ! NOW TO WORK OUT THE SYSTEM MOBILITY
1020  !
1030  FOR I=1 TO 128
1040  A(I)=M(I)+F(I)
1050  B(I)=N(I)+G(I)
1060  C(I)=A(I)+R(I)
1070  D(I)=B(I)+S(I)
1080  Y(I)=C(I)*C(I)+D(I)*D(I)
1090  T(I)=(A(I)*C(I)+B(I)*D(I))/Y(I)
1100  V(I)=(B(I)*C(I)-A(I)*D(I))/Y(I)
1110  NEXT I
1120  ! SYSTEM RESULTS IN RECT CO-ORS  T, V
1130  !
1140  !
1150  !
1160  ! SPACE TO PLOT RECT CO-ORS
1170  !
1180  !
1190  !
1200  !
1210  ! TO PLOT POLAR CO-ORS
1220  FOR I=1 TO 128
1230  U(I)=SQR(T(I)*T(I)+V(I)*V(I))
1240  W(I)=ATN(V(I)/T(I))
1250  !
1260   IF (T(I)>=0) AND (V(I)>=0) THEN 1350
1270   IF (T(I)>=0) AND (V(I)<0) THEN 1300
1280   IF (T(I)<0) AND (V(I)>=0) THEN 1320
```

```
1290   IF (T(I)<0) AND (V(I)<0) THEN 1340
1300  W(I)=-W(I)
1310   GOTO 1350
1320  W(I)=180-W(I)
1330   GOTO 1350
1340  W(I)=180+W(I)
1350  NEXT I
1360  ! SO POLAR CO-ORS ARE U W
1370  !
1380  ! TO PLOT THE SYSTEM MOBILITY
1390  !
1400  PRINTER IS 16
1410  PRINT PAGE
1420  PRINT "    DO YOU WANT TO PLOT SYSTEM MOBILITY   Y  or  N "
1430  INPUT A$
1440  IF A$<>"N" THEN 1460
1450  STOP
1460  PLOTTER IS 13,"GRAPHICS"
1470  GRAPHICS
1480  LIMIT 0,184,0,140
1490  LOCATE 20,120,20,80
1500  SCALE 1,128,-10,10
1510  FRAME
1520  FOR I=1 TO 128
1521  U(I)=LGT(U(I))
1530  PLOT I,U(I)
1540  NEXT I
1550  PENUP
1560  SCALE 1,128,-200,200
1570  LINE TYPE 6
1580  FOR I=1 TO 128
1590  PLOT I,W(I)
1600  NEXT I
1610  PENUP
1620  PRINTER IS 16
1630  PRINT PAGE
1640  PRINT "  DO YOU WANT A COPY OF THE SYSTEM MOBILITY  Y or N"
1650  INPUT A$
1660  IF A$<>"Y" THEN 1680
1670  DUMP GRAPHICS
1680  STOP
1690  END
```

APPENDIX 4

NOISE CONTROL HANDBOOK FOR HOOVER DESIGN ENGINEERS

SUMMARY

Many engineers do not have the necessary background or the appreciation of the principles of acoustics to arrive at an optimum solution of a noise reduction problem. Yet, there is a vast amount of relevant information available in an increasing pool of literature. What is required is a well signposted and cross referenced noise control handbook which can be directly accessed to solve noise problems. This brief handbook should fill that need by providing the most relevant information with a collection of commonly used concepts, facts and guidelines.

To hold this handbook to a manageable length, a compromise has been made as to the range and depth of subjects covered. A full treatise of acoustic information would run into several volumes. The range of topics is, therefore, limited to the needs of Hoover design engineers and more inclined towards vacuum cleaner design. The subject material has been drawn from an extensive noise control literature and references are given where further information is required . Therefore, the handbook is designed to complement, rather than compete with, the available relevant literature.

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