DESIGN AND QUALIFICATION OF A SEMI-ANECHOIC CHAMBER AND INVESTIGATION INTO NOISE CHARACTERISTICS OF A VACUUM CLEANER

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Approval of Thesis

"DESIGN AND QUALIFICATION OF A SEMI-ANECHOIC CHAMBER AND INVESTIGATION INTO NOISE CHARACTERISTICS OF A VACUUM CLEANER"

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I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

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ABSTRACT

DESIGN AND QUALIFICATION OF A SEMI-ANECHOIC CHAMBER AND INVESTIGATION INTO NOISE CHARACTERISTICS OF A VACUUM CLEANER

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In this study a centrifugal fan is studied for noise characteristics and measurements in a semi- anechoic room. A semi-anechoic room is constructed inside Fluid Mechanics Laboratory of Mechanical Engineering Department has been qualified with respect to ISO 3745 standard. The fan characteristic is obtained as proposed in AMCA standards 210-75, by simply measuring the voltage and current of the motor during operation and calculating the power consumption of the assembly. Noise measurements are taken using two microphones attached to a multi-channel data acquisition and processing system in the semi anechoic room. Several different configurations of the vacuum cleaner with some parts removed or replaced systematically are considered during the noise measurements. Some of the results showed that the damping material placed inside the motor cover is proved to be very effective in noise reduction. Two different damping materials are examined for comparative evaluation.

Keywords: Vacuum cleaner, Centrifugal Fan, Noise, Semi- Anechoic Room

ÖZ

YARI YANSIMALI ODA YETERLİLİĞİ VE ELEKTRİK SÜPÜRGESİ GÜRÜLTÜ KARAKTERİSTİKLERİ ÜZERİNE UYGULAMASI

Kayhan, Cihan
Yüksek Lisans, Makina Mühendisliği Bölümü
Tez Yöneticisi: Prof. Dr. O. Cahit Eralp
Yardımcı Tez Yöneticisi Prof.Dr. Mehmet Çalışkan

Mayıs 2008, (94 sayfa)

Bu çalışmada bir fan karakteristiriği ve bu fanın gürültü ölçümü yarı yansımalı bir odada yapılmıştır. Ses ölçümlerinin yapıldığı yarı yansımalı oda ISO 3745 standardına gore tasarlanmış ve ODTÜ Makina Mühendisliği Akışkanlar Mekaniği Laboratuarı'nda inşa edilmiştir.. Fan karakteristği basit olarak AMCA 210-75 standardında öngörüldüğü üzere fan verimi, motor montajı ve motorun voltaj ve akımı ile motor parçasının güç harcamasını hesaplayarak belirlenmiştir. Gürültü ölçümleri iki mikrofon kullanılarak çok kanallı veri toplama ve işleme sistemi kullanılarak yapılmıştır. Elektrikli süpürge ses ölçümleri için bir çok değişik pozisyonda ve konfigürasyonda elektrik süpürgesi parçaları yer değiştirilerek kaşılaştırmalı olarak ses ölçümleri alınmıştır. Sonuçlara gore motor çevresindeki sönümleme malzemesinin gürültü yalıtımında etkili olduğu gözlemlenmiştir. İki farklı yutumlama malzemesi karşılaştırılmıştır.

Anahtar Kelimeler: Elektrik Süpürgesi, Santrifüj Fan, Gürültü, Yarı Yansımalı Oda

To My Family,

To My Wife,

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NOMENCLATURE

V	:Voltage	(V),
P	: Power	(W),
I	: Electrical Current	(A),
T	:Tempareture	(^oC),
t	:Time	(s),
$h_{dy,o}$	:Initial deflection at the manometer	(mm)
T_{amb}	:Ambient temperature	(^oC)
P_{amb}	:Ambient pressure	(Pa)
ρ_{alc}	:Density of alcohol	(g/mm^3)
ρ_{water}	:Density of water	(g/mm^3)
h_{dy}	:Δh reading of the inclined manometer	(mm Alcohol)
h_{tot}	:Δh reading of the U tube manometer	(mm Water)
R	:Gas constant of air	
P_{dyn}	:Dynamic pressure	(Pa)
V_{avg}	:Average air velocity	(m/s)
Q	:Volumetric flow rate	(m^3/s)
H_{fan}	:The Fan head that the fan produces	(m Air)
η	:Efficiency	
N	:Rotational speed	
f_{imb}	:Fundamental frequency due to rotary imbalance	
f_{bpf}	: fundamental blade passing frequency	

L_{pi} : corrected sound pressure level for the ith microphone position and frequency

L'_{pi} : the measured sound pressure level for the ith microphone position and frequency

L''_{pi} : background sound pressure level for the ith microphone position and frequency

K_{1i} : background noise correction for the ith microphone position and frequency

L_{pf} : surface band pressure level in a frequency band

$\overline{L_{pf}}$: Averagesurface band pressure level over the test hemisphere.

S_2 : $2 \cdot \pi \cdot r^2$ is the area of the test hemisphere (of radius r);

S_0 : 1 m^2

B : the barometric pressure during the measurements (Pa)

B_0 : the reference barometric pressure (Pa)

θ : the air temperature during measurements, in ^oC and manometer inclination

L_{Wj} : the sound power level in the j^{th} 1/3 octave band (in dB).

C_j : A-weighing values

L_{WA} : sound power level

CHAPTER 1

INTRODUCTION

1.1 General

Laboratories and engineering test cells often require a sound environment that is completely free of reverberation throughout the audible frequency spectrum from low frequency to high frequency. This environment allows measurement of radiated sound power and directivity.

Anechoic chamber is an enclosure especially designed with walls that absorb sound or radiation, creating an essentially free-field environment for testing. Anechoic chambers are used both for acoustic measurements. In this study a semi anechoic chamber is designed and constructed inside Fluid Mechanics Laboratory of Mechanical Engineering Department. This facility has been qualified with respect to ISO 3745 standard.

Noise is one of the comfort parameters for house appliances. In this study, small centrifugal fans that are utilized in vacuum cleaners are the focus of interest. Operational noise of the small size centrifugal fan is to be investigated. The centrifugal fan, widely used in home appliance poses a serious noise problem. Especially, the centrifugal fan used in a vacuum cleaner produces a high-level noise due to its high rotating speed.

Rotating machinery noise is dominated by tones at the *blade passing frequency (bpf)*, which is defined as the frequency of blades passing a stationary point, and higher harmonics [1]. Besides impeller blades, shaft rotation and rotating components of driving motor are also sources of noise.

In this study, acoustic noise of a vacuum cleaner fan is experimentally determined at its design operating points. Experiments are conducted in a semi-anechoic test room, in two steps. Experimental work has been performed in two main stages. In the first stage of experiments, only the vacuum cleaner noise is measured. In the second stage, an noise control study of a vacuum cleaner is performed.

1.2 Review of Literature

In the literature, there are experiments with estimated results concerning vibrational or acoustical relations between turbomachine performances. Results are usually reported on graphs of frequency spectra. Information concerning amplitudes of levels is rare.

The centrifugal fan has an impeller, a diffuser and a circular or rolute casing. As the size of an impeller becomes smaller, its rotating speed needs to be increased to meet required performance specification, and therefore, the aerodynamic force applied on the impeller blades becomes severer. This unsteady aerodynamic force may generate excessive noise to the environments. Among the various noise sources in the centrifugal fan of a vacuum cleaner, the flow interaction between the rotating impeller and stationary diffuser vane plays a major role in generating the strong tonal noise. In addition, the flow separated at the impeller blades and the inflow turbulence contribute to the broadband noise. Due to a small gap between the impeller and diffuser, the sound pressure levels at the blade passing frequency and its higher harmonic frequencies are dominant in the noise spectrum of the vacuum cleaner. As the control and the reduction of the noise from vacuum cleaners are essential requirements for the fan design, various noise reduction methods have been studied for the last three decades [1–4]. Earlier works mostly focused on identifying the dominant noise generation mechanism for the centrifugal fan and suppressing the generated noise [1,2]. Neise summarized the efforts in reducing the blade passage tone by changing the geometry of the impeller and the cut-off [1,2]. Sugimura and Watanabe discussed the resonance phenomena at the impeller passage and the noise reduction method for the centrifugal fan [3]. Lauchle and Brungart changed the regular pitched impeller blade to the uneven pitched impeller configuration in order to reduce the tonal noise [4].

On the other hand Wan Ho Jeon developed an aeroacoustical methodology for a centrifugal fan utilized in a vacuum cleaner to predict the sound pressure level numerically so that it can be implemented approximately during the design stage of the centrifugal fan. They used a two-dimensional vortex method to analyze the unsteady flow field of the centrifugal fan. Vortex method has been widely used in calculating the unsteady flow field of the turbomachines[5,6]. The impeller and diffuser vanes were modelled as point vortices and the surrounding casing was modelled as source panels. The aeroacoustic pressure was calculated by Ffowcs–Williams and Hawkings (FWH) equation. In that equation, only dipole term was considered because the other terms like monopole and quadrupole were not largely affect on the sound generation compared to the dipole term [7,8]. The acoustics generated by the moving impeller blades and flow interactions in the stationary diffuser calculated separately

1.3 Aim of the Thesis

Noise generated from centrifugal fans, which are widely used in fluid machines, is becoming more serious problems as the required power and the rotating speed of impeller increase. Although some limited researches on the noise source identification and analyses have been performed since early 1960's, they are still not enough to establish a systematic methodology that can be actually implemented for the design of centrifugal fans.

In the scope of this study several noise measurements are performed in the semi anechoic room constructed in fluid mechanics laboratory to develop a methodology of noise control for a vacuum cleaner.

The experiments are conducted in a semi-anechoic test room. The design and qualification criteria for the room are described in Chapter 2. Experimental work has been performed in two main phases. In the first group of experiments, characteristics of the centrifugal fan of a vacuum cleaner, used in the experiments, are evaluated. These are described in Chapter 3.
Chapter 4 describes the set-up and the experimental procedure for the noise experiments in detail.

Chapter 5 describes the case studies considering the noise measurements for vacuum cleaner and its components.

The discussed in Chapter 6.

CHAPTER 2

THE SEMI ANECHOIC ROOM

2.1 General

Anechoic literally means without echoes. Anechoic refers to the absence of audio reflections. The closest thing to this situation in nature is the great outdoors, but even here there are reflections from the ground, various objects, etc. It is almost impossible to create a truly anechoic environment, as there is no such thing as a perfect sound absorber. At high frequencies, it is possible to create near-anechoic conditions, but the lower the frequency, the harder this is because the absorption is wavelength (or frequency) dependent.

Laboratories and engineering test cells often require a sound environment that is completely free of reverberation throughout the audible frequency spectrum from low frequency to high frequency. This environment allows measurement of radiated sound power and directivity.

Anechoic chamber is an enclosure especially designed with walls that absorb sound or radiation, creating an essentially free-field environment for testing. Anechoic chambers used for acoustic measurements are considered.

Acoustic anechoic chambers have surfaces that absorb more than 99% of incident sound energy over the frequency range of interest, thereby approximating a free field. Room surfaces are covered with sound-absorbing wedges. Anechoic chambers are used in precision-grade measurements.

The Murray Hill anechoic chamber, built in 1940, is the world's oldest wedge-based anechoic chamber. The interior room measures approximately 30 feet high by 28 feet wide by 32 feet deep. The exterior cement and brick walls are about 3 feet thick to keep outside noise from entering the chamber. [4]

Semi – anechoic room includes one perfectly reflecting surface which is usually the floor. Absorption coefficient for that single surface must not exceed 0.06. Semi – anechoic room is defined as the room in which a free field over a reflecting plane is obtained. It is a fact that an isolated surface, perfectly reflecting for sound applications, is a symmetry plane.

The construction details of the anechoic room and the verification results are given in Appendix A.

2.2 General Theory Of Acoustics

Sound:

Sound is a sensation caused by fluctuations in equilibrium pressure in air.. When sound pressure fluctuations reach the listener they are detected by a delicate mechanism in the inner ear and perceived as sound by the brain. Sound has three principal characteristics: frequency, amplitude and time pattern. These characteristics can be seen in figure 2.1.

Sound Level Measurements:

In field of research and product development, various sound level measurements are required for detailed analysis of noise. Special equipments and test rooms employed to determine local sources of sound, directivity patterns, acoustic efficiency, power output etc. All these tests are very helpful and information enables the manufacturer to control the noisiness of his product; also to maintain a check on the uniformity of their performance.

In order to achieve the object in view, it is necessary that the sound level measurements are made in controlled environment. It is known that sound pressure level measured on a relative position of a sound source depends as well as distance from the source and direction, but also on environmental acoustical conditions. Thus, the test chamber that is to be used should not interrupt the experiment.

Free Field:

Free field is a sound field in a homogeneous, isotropic medium, free of boundaries. The most obvious method for making free-field measurements is; out-of-doors at a great distance from all reflecting surfaces. However, there are practical difficulties because of interruptions that may be caused due to rain, wind, noise (either natural or man made). Apart from these, there are environmental difficulties in arranging measurements at an adequate distance from the earth, so that the sound reflections are negligible.

The simplest kind of ideal environment visualized that there is no other object, except the source under test, the measuring equipment, air and non-reflecting surfaces. In order to approximate free-field approach reflections from the walls shouldn't affect the measured values. The rooms designed to meet the above requirements are called as anechoic chambers.

Anechoic chambers:

Anechoic chambers are used where conditions approximating free-field is required. The sound absorption in such rooms is between 99 percent and 100 percent, or the sound pressured reflection is from 10 percent to zero [4]

However test on anechoic chambers are great problem since large (heavy) products like cars, buses, motor cycles, tool machinery, washing machines, refrigerators, and all other equipment intended for staying on a floor or on the ground should during the performance tests.

Semi-anechoic chambers:

Semi-anechoic chambers are like anechoic chambers. Only difference is semi-anechoic chambers have a reflecting and stable ground surface. It is a simulation of half-free field. They are usually used for large (heavy) products like cars, buses, motor cycles, tool machinery, washing machines, refrigerators, etc…

Instantaneous Sound Pressure: P(t)

Sound wave is small amplitude pressure wave. It creates pressure differences in the medium that it moves in.

Therefore, instantaneous sound pressure is a value at a particular instant in time of the fluctuating pressure that is superimposed on the atmospheric static pressure due to the presence of a sound wave, and existing at a given point in space, in a stated frequency band. It is expressed in pressure units of Pa (Pascal).

There is also logarithmic measures of energy, intensity and power of a sound wave relative to a reference noise source.

Sound Pressure Level: Lp

Sound Pressure Level Lp is a logarithmic measure of the energy of a particular noise relative to a reference noise source. It is almost always expressed in decibels compared to a reference source of 20 µPa (micro Pascal) sound pressure.

$$L_p = 20 \cdot \log\left(\frac{P_1}{P_0}\right) \text{dB} \tag{2.1}$$

$$L_p = 10 \cdot \log\left(\frac{P_1^{\,2}}{P_0^{\,2}}\right) dB \qquad (2.2.)$$

The reference sound pressure is

$$P_0 = 2 \cdot 10^{-5} Pa$$

= 20 μPa (microPascal), (Pa = Pascal = N / m²; N = newton)

It can be useful to express sound pressure in this way when dealing with hearing, as the perceived loudness of a sound correlates roughly logarithmically to its sound pressure.

By using the data above one can calculate the sound pressure (Pa) of the given sound pressure level (dB). The calculations and P versus L_p are shown in figure 2.2.

$$P_0 := 20 \cdot 10^{-5}$$

$$i := 1..120$$

$$Lp_i := i$$

$$P_i := P_0 \cdot 10^{\frac{Lp_i}{20}}$$

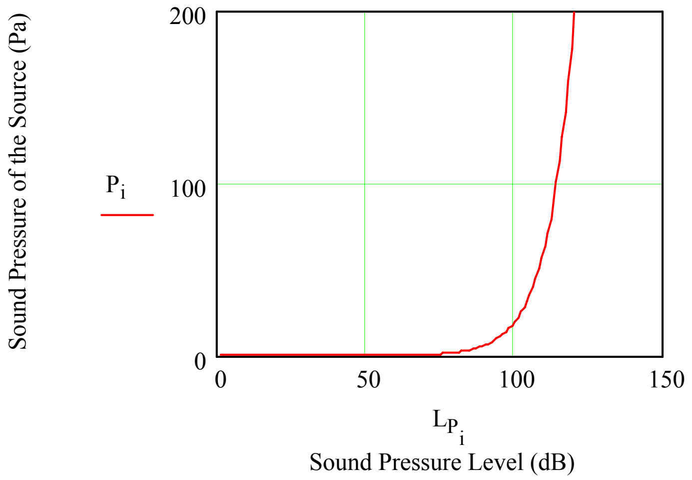


Figure 2.1. A Sample Solution of Sound Pressure versus Sound Pressure Level

Sound Intensity Level: L_I

Sound intensity level or acoustic intensity level is a logarithmic measure of the sound intensity in comparison to the reference level of 0 dB (deciBell).

The measure of a ratio of two sound intensities is

$$L_I = 10 \cdot \log\left(\frac{I}{I_0}\right) \quad dB \tag{2.3}$$

where I and I_0 are the intensities.

The sound intensity level is given the letter "L_I" and is measured in "dBW". dB is dimensionless.

If I_0 is the standard reference sound intensity, where

$$I_0 = 10^{-12} \quad \frac{W}{m^2} \tag{2.4}$$

Sound Power Level: L_w

Sound power level or acoustic power level is a logarithmic measure of the sound power in comparison to the reference level of 0 dB (decibels).

The measure of a ratio of two sound powers is

$$L_W = 10 \cdot \log\left(\frac{W}{W_0}\right) \quad dB \tag{2.5}$$

where W and W_0 are the powers.

The sound power level is given the letter "L_w" measured in "dB".

If W_0 is the standard reference sound power, where

$W_0 = 10^{-12} W$

2.3. The Inverse Square Law

The sound intensity from a point source of sound will obey the inverse square law if there are no reflections or reverberation. A plot of this intensity drop shows that it drops off rapidly. A schematic diagram is given İn Figure 2.2.

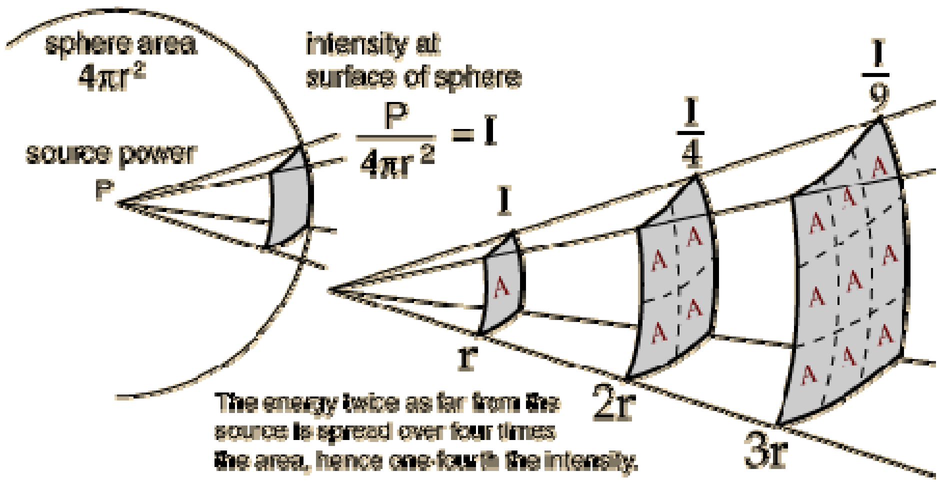


Figure 2.2: Sound Intensity Decrease of a Ideal Point Sound Source

2.4 Theory of Experiment

The performance of an anechoic or semi-anechoic room is assessed by comparing the spatial decrease of sound pressure emitted from a test source with the decrease of sound pressure with distance from source, according to the inverse square law that occur in a true anechoic or semi anechoic field.

From the sound pressure levels measured at the positions specified in A-4-1, the estimation of sound pressure levels based on the inverse square law shall be determined for each direction of measurement from the following equation:

$$L_p(r) = 20\log\left(\frac{a}{r - r_0}\right) \text{dB}$$

(2.6)

Where;

$$a = \frac{\left(\sum_{i=1}^{N} r_i\right)^2 - N \cdot \sum_{i=1}^{N} r_i^2}{\left(\sum_{i=1}^{N} r_i \cdot \sum_{i=1}^{N} q_i\right) - N \cdot \sum_{i=1}^{N} r_i \cdot q_i}$$

(2.7)

and r_0 is the collinear offset of acoustic center along the axis of the microphone traverse. It is a measurement of the separation between the acoustic center of the source and the measurement hemisphere.

I is given by the following formula:

$$r_0 = -\left[\frac{\sum_{i=1}^{N} r_i \cdot \sum_{i=1}^{N} r_i \cdot q_i - \sum_{i=1}^{N} q_i \cdot \sum_{i=1}^{N} r_i^2}{\left(\sum_{i=1}^{N} r_i \cdot \sum_{i=1}^{N} q_i\right) - N \cdot \sum_{i=1}^{N} r_i \cdot q_i}\right]$$

(2.8)

Where, $L_{pi's}$ are the functions of frequency

- L_{pi} is the sound pressure level at the i^{th} measurement position, in decibels;
- r_i is the distance of the i^{th} measurement position from the centre of the measurement sphere or hemisphere;
- N is the number of measurement positions along each microphone traverse.

Using the estimation of sound pressure levels based on the inverse square law, deviations of the sound pressure levels at all measurement positions from the inverse square law are determined by the following equation:

$$\Delta L_{pi} = L_{pi} - L_p(r_i) \quad \text{dB}$$

ΔL_{pi} is the deviation from the inverse square law;

L_{pi} is the sound pressure level at the i^{th} measurement position, in dB

$L_p(r_i)$ is the sound pressure level at distance r_i estimated by the inverse square law; in dB

2.4.1. Qualification procedures for Semi-Anechoic Rooms

The deviations from the measured sound pressure levels from those estimated using the inverse square law, obtained according to ISO 3745 A.4.3.2, shall not exceed the values given in ISO 3745 Table A.2. This table is shown below as Table 2.1.

Table 2.1 Maximum Allowable Deviation of Measured Sound Pressure Levels from Theoretical Levels using Inverse Square Law

Type of test room	1/3 octave band center frequency in Hz	Maximum Allowable Deviations in dB
Anechoic Room	<630	± 1.8
	800 to 5000	± 1.0
	>6300	± 1.5
Semi-Anechoic Room	<630	± 2.5
	800 to 5000	± 2.0
	>6300	± 3.0

.

Deviations in Table 2.1 determine the frequency range over which measurements may be made according to ISO 3745. If the test room qualified over a specified frequency range, then measurements are reported to be "in conformance" with the international standard ISO 3745

Between the frequencies 63 Hz- 4000 Hz; all the sound pressure levels of 11 points P1 to P11 are known.

The estimation of sound pressure levels based on the inverse square law will be calculated.

2.7 Theory of Directivity of a Sound Source

Assume a point source, suspended in mid-air and radiates sound waves uniformly in all directions. The sound will spread out spherically and the intensity at any location will depend on the surface area of the sphere at that location.

The sound intensity, I, in Watt/m2, at any distance r meters from the source is given by:

$$I = \frac{W}{4 \cdot \pi \cdot r^2} \tag{2.9}$$

This equation is a type of Inverse Square Law which state that the sound intensity (outward) is inversely proportional to the square of the distance from the point source.

Sound intensity level in terms of decibels is;

$$L_I = 10 \cdot \log\left(\frac{I}{I_0}\right) \tag{2.10}$$

Where it is known that that L_p = L approximately in air

When the sound intensity and pressure levels at two distances, r_1 and r_2 from the sound source are compared:

$$L_{I1} - L_{I2} = 10 \cdot \log\left(\frac{I_1}{I_2}\right) \tag{2.11}$$

$$L_{I1} - L_{I2} = 10 \cdot \log\left(\frac{{r_2}^2}{{r_1}^2}\right) \tag{2.13}$$

$$L_{I1} - L_{I2} = 20 \cdot \log\left(\frac{r_2}{r_1}\right) \tag{2.14}$$

If $r_2 = 2\, r_1$

Then;

$$L_{I1} - L_{I2} = 20 \cdot \log(2) \tag{2.15}$$

$$L_{I1} - L_{I2} = 6 \tag{2.16}$$

Nonetheless, it is absolutely necessary to estimate at least the directional characteristics of the noise source in question. This requirement is met by an estimation of the directivity factor, Q, directly or by an approximate calculation using the directivity index. (DI)

The directivity factor, Q, is defined as the ratio of the intensity (W/m^2) at some distance and angle from the source to the intensity at the same distance, if the total power from the source were radiated uniformly in all directions.

$$Q = \frac{I_\theta}{I} \tag{2.17}$$

Where I_θ is the sound intensity at distance r and angle θ from the source

I is the Average sound intensity over a spherical surface at the distance r, due to a uniformly radiating source

And the directivity index (DI) is defined as:

$$DI = 10 \cdot \log\left(\frac{I_\theta}{I}\right) \tag{2.18}$$

To take account of any directional effect of the sound source, a directivity factor Q is defined such that the sound intensity is given by:

$$I = Q \cdot \left(\frac{W}{4 \cdot \pi \cdot r^2}\right) \tag{2.19}$$

The known equations are:

$$L_p = 10 \cdot \log\left(\frac{I}{I_0}\right) \tag{2.20}$$

$$L_W = 10 \cdot \log\left(\frac{W}{W_0}\right) \tag{2.21}$$

$$I = Q \cdot \left(\frac{W}{4 \cdot \pi \cdot r^2}\right) \tag{2.22}$$

Relation between sound intensity (I) & directivity factor (Q) is defined as Eqn 2.15.

By taking the logarithm of equations described above:

$$\log(I) = \log(Q) + \log(W) - \log(4 \cdot \pi) - 2 \cdot \log(r) \quad (2.23)$$

$$\log(I) = \log(I_0) + \frac{L_p}{10} \quad (2.24)$$

$$\log(I) = -12 + \frac{L_p}{10} \quad (2.25)$$

By taking logarithm of equation 2.22 :

$$\log(W) = \log(W_0) + \frac{L_w}{10} \quad (2.26)$$

$$\log(W) = -12 + \frac{L_w}{10} \quad (2.27)$$

By combining them

$$L_p = 10 \cdot \log(Q) + L_w - 10 \cdot \log(4 \cdot \pi) - 20 \log(r) \quad (2.28)$$

$$L_p = 10 \cdot \log(Q) + L_w - 11 - 20 \log(r) \quad (2.29)$$

In order to determine sound pressure level of a point on experiment one has to know directivity index or directivity factor.

In Instruction Manual for Sound Source 4224 directivity index for all sound frequencies in 1/3 bands when source is at position 45 degrees are given.

2.6. Experimental Setup and Results For Room Qualification

2.6.1 Test Apparatus

Dual Channel Dynamic Signal Analyzer

The analyzer is an instrument which allows the capture of up to two channels of time varying signal and a number of analysis modes. Its maximum single channel analysis usable frequency range is 104 kHz, which is reduced to 52 kHz for dual channel operation. Other types of analysis include full, 1/3 and 1/12 octave analysis, correlation analysis, and a range of triggering modes. It also features a simple time domain data capture (2 MB buffer) capability, GPIB interface for control and data transfer, floppy disc data storage and GPIB plotter output.

Sound Level Calibrator Type

The sound calibrator is a tool for adjustment of the sensitivity of the measurement chain at a certain frequency prior to a measurement. This is a source for pure harmonic tone 1000 Hz, 93.8 dB. Microphone was calibrated with that defined tone.

Sound Source Type 4224:

The Sound Source Type 4224 is a loudspeaker with built-in power amplifier and noise generator. The type 4224 is specially designed for building acoustics measurements. When driven continuously, the type 4224 can typically deliver up to 115 dB sound power level in the frequency range 100 Hz to 4 kHz or up to 118 dB sound power level.

To produce bands of noise, this signal can fed to an external 1/3 or 1/1 octave filter, before amplification and reproduction by the loudspeaker.

Sound Level Meter Type 2239:

Integrating Sound Level Meter Type 2239 is a rugged, easy-to-use. It provides with A-weighted RMS and C-weighted Peak levels in parallel. Measurements can be manually controlled or, the measurement time can be pre-set (from 10 seconds to 8 hours). Type 2239 can store up to 40 measurements and, if the time is pre-set, storage is automatic. Indeed, it's a one-button-operation - just switch on and it measures, stops and stores automatically. Uses include;

- · Noise surveys
- · Spot checks
- · Occupational noise exposure
- · Measuring machinery noise

2.6.2 SET-UP

Preparing Of Microphone Traverse (ISO 3745 A.3.3)

First of all microphone traverses must have been defined. They have been formed along five straight paths away from the geometric center of the measurement hemisphere in different directions. Strings are used for realizing the paths. Lines from the geometric center of the measurement hemisphere to the room corners have been selected as the key microphone paths are the (where the corners refer to the intersection between the two walls and the ceiling, or two walls and the floor). These key paths to the corners are the four of the five required traverses. The fifth one has been selected as the line from the geometric center of the measurement hemisphere to the top of the hemisphere which can be fixed to center of ceiling. The set up is shown in Figure 2.3..

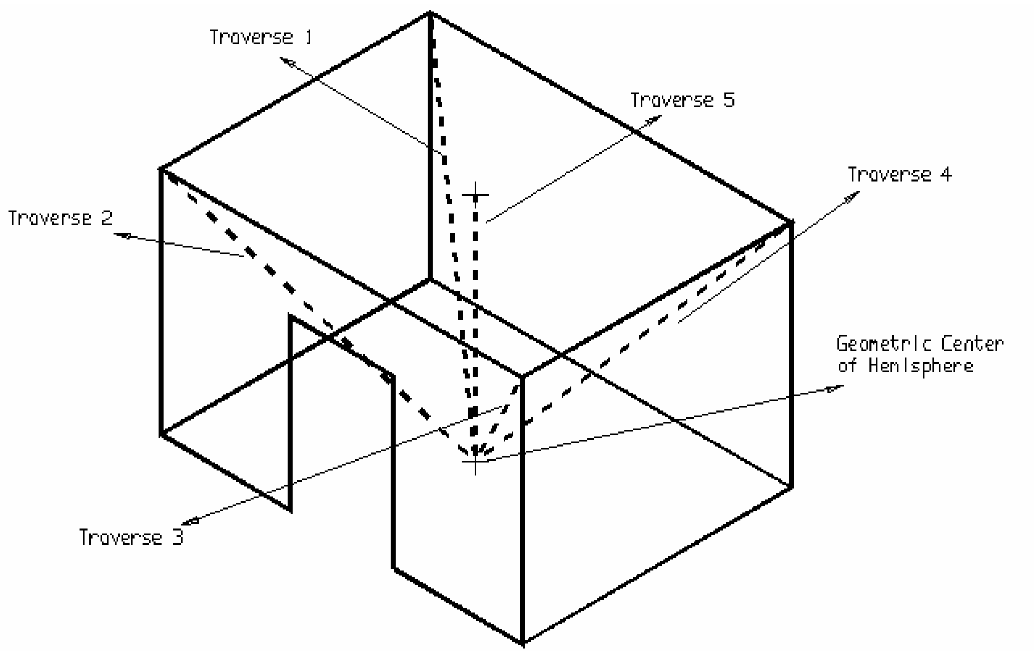
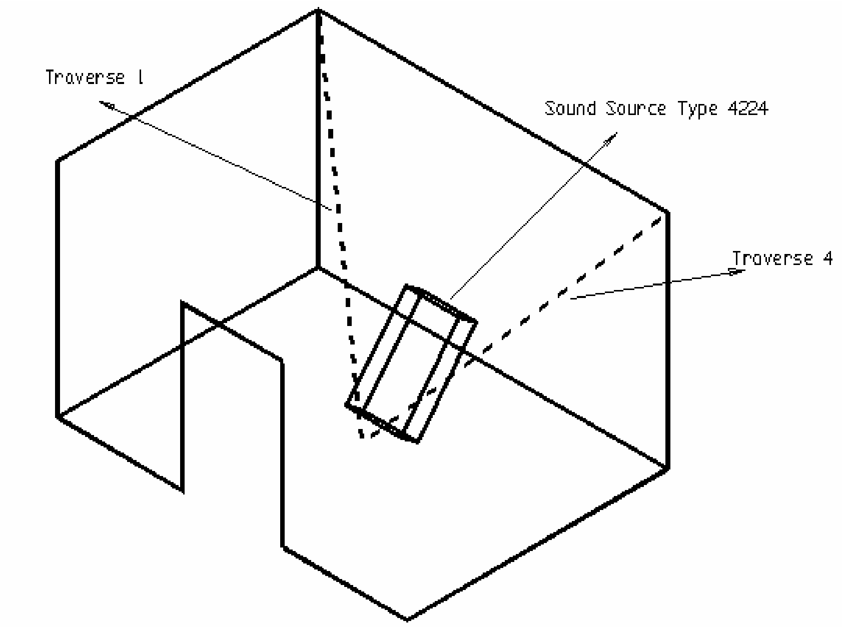


Figure 2.3. Microphone Traverses

Locating the Sound Source

The sound source Type 4224 have been placed to the position as shown in figure



2.4.

Figure 2.4. A Schematic View of Test Environment

As it is seen in figure B-2 only two of the traverses used in test and the microphone is not omni directional (the requirements of ISO 3745 could not been satisfied successfully.)

Calibration of Microphone and Getting the Data

The microphone has been calibrated with sound level Calibrator Type 4230. The calibration was made in acoustic room and with silent environment. 4230 is a source for pure harmonic tone 1000 Hz, 93.8 dB. Microphone was calibrated with that defined tone.

After the calibration data acquisition is started. First of background pressure is measured. Then it is started to measure the sound pressure levels. Only two of the traverses (traverse 1 and traverse 4) are used. Totally 11 data (sound pressure level) have been taken from 2 traverses. It is used the one third octave band between the frequencies 63 Hz to 4000 Hz. The pressure levels measured in 19 frequency levels. The positions of microphone (the positions where the pressure levels were measured) and frequency levels are as follows.

On Traverse 1: r_1=1 m (P1), r_2=1,2 m (P2), r_3=1,4 m (P3), r_4=1,6 m (P4), r_5=1,8 m (P5), r_6=1,9 m (P6),

On Traverse 2: r_7=1,8 m (P7), r_8=1,6 m (P8), r_9=1,4 m (P9), r_{10}=1,2 m (P10), r_{11}=1 m (P11)

Center Frequencies of interest for One-Third-Bands ranging from 63 Hz to 4000 kHz

63-80-100-125-160-200-250-315-400-500-630-800-1000-1250-1600-2000-2500-3150-4000 Hz

2.6.3. Sample Calculation

Sample calculations for traverse 1 at frequency of 63 Hz are as follows;

Path (Travers) 1- frequency band 63Hz

The measured background pressure (B) for 63 Hz = 55.96 dB,

Points where the measurements have been done and the measured sound pressure levels are as follows;

r_1=1 m (P1), r_2=1,2 m (P2), r_3=1,4 m (P3), r_4=1,6 m (P4), r_5=1,8 m (P5), r_6=1,9 m (P6)

P1: 78.82 dB (Sound pressure level not pressure) r_1=1.0 m (P1)

P2: 74.89 dB r_2=1.2 m (P2)

P3: 75.35 dB r_3=1.4 m (P3)

P4: 68.86 dB r_4=1.6 m (P4)

P5: 67.48 dB r_5=1.8 m (P5)

P6: 65.57 dB r_6=1.9 m (P6)

The differences of sound pressure levels from background noise levels are shown in table 2-2

Table 2-2: The differences of measured sound pressure level from background pressure levels (results are in dB)

$$B := 55.96$$

$$\begin{aligned}
B\Delta 1 &:= P_1 - B & B\Delta 1 &= 22.87\\
B\Delta 2 &:= P_2 - B & B\Delta 2 &= 18.93\\
B\Delta 3 &:= P_3 - B & B\Delta 3 &= 19.39\\
B\Delta 4 &:= P_4 - B & B\Delta 4 &= 12.9\\
B\Delta 5 &:= P_5 - B & B\Delta 5 &= 11.52\\
B\Delta 6 &:= P_6 - B & B\Delta 6 &= 9.61
\end{aligned}$$

The calculations made according to written procedure in ISO 3745 are shown in tables 2-3, 2-4, 2-5

Table 2.3: Calculations of q Values According to ISO 3745

$q_1 := 10^{-0.05 \cdot P_1}$	$q_1 = 1.144 \times 10^{-4}$
$q_2 := 10^{-0.05 \cdot P_2}$	$q_2 = 1.801 \times 10^{-4}$
$q_3 := 10^{-0.05 \cdot P_3}$	$q_3 = 1.708 \times 10^{-4}$
$q_4 := 10^{-0.05 \cdot P_4}$	$q_4 = 3.606 \times 10^{-4}$
$q_5 := 10^{-0.05 \cdot P_5}$	$q_5 = 4.227 \times 10^{-4}$
$q_6 := 10^{-0.05 \cdot P_6}$	$q_6 = 5.266 \times 10^{-4}$

Table 2-4: Calculation of r_0 According to ISO 3745)

$$\Sigma r := r_1 + r_2 + r_3 + r_4 + r_5 + r_6$$

$$\Sigma r = 8.9$$

$$\Sigma rr := r_1^2 + r_2^2 + r_3^2 + r_4^2 + r_5^2 + r_6^2$$

$$\Sigma rr = 13.81$$

$$\Sigma q := q_1 + q_2 + q_3 + q_4 + q_5 + q_6$$

$$\Sigma q = 1.775 \times 10^{-3}$$

$$\Sigma rq := r_1 \cdot q_1 + r_2 \cdot q_2 + r_3 \cdot q_3 + r_4 \cdot q_4 + r_5 \cdot q_5 + r_6 \cdot q_6$$

$$\Sigma rq = 2.908 \times 10^{-3}$$

$$r_0 := -\frac{\Sigma r \cdot \Sigma rq - \Sigma rr \cdot \Sigma q}{\Sigma r \cdot \Sigma q - 6 \cdot \Sigma rq}$$

$$r_0 = 0.828 \quad \text{m}$$

Table 2-5: Calculations of L_p Values According to ISO 3745 (results are in dB)

$$L_{p1} := 20\log\left(\frac{a}{r_1 - r_0}\right) \qquad L_{p1} = 82.208$$

$$L_{p2} := 20\log\left(\frac{a}{r_2 - r_0}\right) \qquad L_{p2} = 75.5$$

$$L_{p3} := 20\log\left(\frac{a}{r_3 - r_0}\right) \qquad L_{p3} = 71.76$$

$$L_{p4} := 20\log\left(\frac{a}{r_4 - r_0}\right) \qquad L_{p4} = 69.154$$

$$L_{p5} := 20\log\left(\frac{a}{r_5 - r_0}\right) \qquad L_{p5} = 67.153$$

$$L_{p6} := 20\log\left(\frac{a}{r_6 - r_0}\right) \qquad L_{p6} = 66.302$$

The deviations for each measured sound pressure levels can be calculated by subtracting theoretical sound pressure level (L_p) from measured one (P). These values must satisfy the values of allowable deviations that are given in ISO 3745. the calculated deviations are shown in table 2-6

Table 2-6: Calculations of Deviations in dB

$$\Delta 1 := P_1 - L_{p1} \qquad \Delta 1 = -3.378$$
$$\Delta 2 := P_2 - L_{p2} \qquad \Delta 2 = -0.61$$
$$\Delta 3 := P_3 - L_{p3} \qquad \Delta 3 = 3.59$$
$$\Delta 4 := P_4 - L_{p4} \qquad \Delta 4 = -0.294$$
$$\Delta 5 := P_5 - L_{p5} \qquad \Delta 5 = 0.327$$
$$\Delta 6 := P_6 - L_{p6} \qquad \Delta 6 = -0.732$$

2.7 Results

2.7.1 Estimation of Sound Pressure Levels Based on Inverse Square Law for Path-1:

The performance of an anechoic or semi-anechoic room is assessed by comparing the spatial decrease of sound pressure emitted from a test source with the decrease of sound pressure with distance from the source, according to the inverse square law (Theory of inverse square law is given on Part 2.2) that would occur in a true anechoic or hemi-anechoic field.

From the sound pressure levels measured at the paths-4 and path-4, the estimation of sound pressure levels based on the inverse square law shall be determined for "each direction of measurement" from the equation that has stated on ISO 3745.

Firstly for path-1, estimated sound pressure levels based on inverse square law is calculated. In order to obtain estimated sound pressure levels based on inverse square law, firstly q values for each point at each frequency must be calculated. Then for each frequency band and for path-1, r_0 and a values are calculated. Finally the estimated sound pressure levels based on inverse square law for path-1 and for each frequency band, is obtained as show on the sample calculations part. The calculated q values at each point at each frequency band can be seen in Table A1 (App. A). Also r_0 and a Values for path-1 on each frequency band ca n be seen in Table A2 (App. b). And also the calculated L_p values are shown in Table A3

Then the difference (deviation) between sound pressure levels and estimated sound pressure levels based on inverse square law at each frequency and point on path-1 is calculated. These calculated deviations are shown in Table A4

Finally for each frequency band, maximum deviation from the inverse square law is obtained.

According to Table ISO 3475 A.2; maximum deviations and allowable deviations are compared. Also the graph of the comparison of deviations can be seen in Figure 2.5.

Table 2.7: The Comparison between Calculated Deviations and Allowable Deviations

1/3 octave bands Frequency	Calculated ΔL_{pi} max	Allowable ΔL_{pi} max
63	3,59	2,5
80	2,04	2,5
100	1,63	2,5
125	1,18	2,5
160	5,09	2,5
200	1,40	2,5
250	1,29	2,5
315	1,97	2,5
400	1,13	2,5
500	0,82	2,5
630	0,83	2,5
800	1,23	2
1000	0,98	2
1250	0,86	2
1600	0,49	2
2000	1,00	2
2500	0,67	2
3150	0,27	2
4000	0,66	2

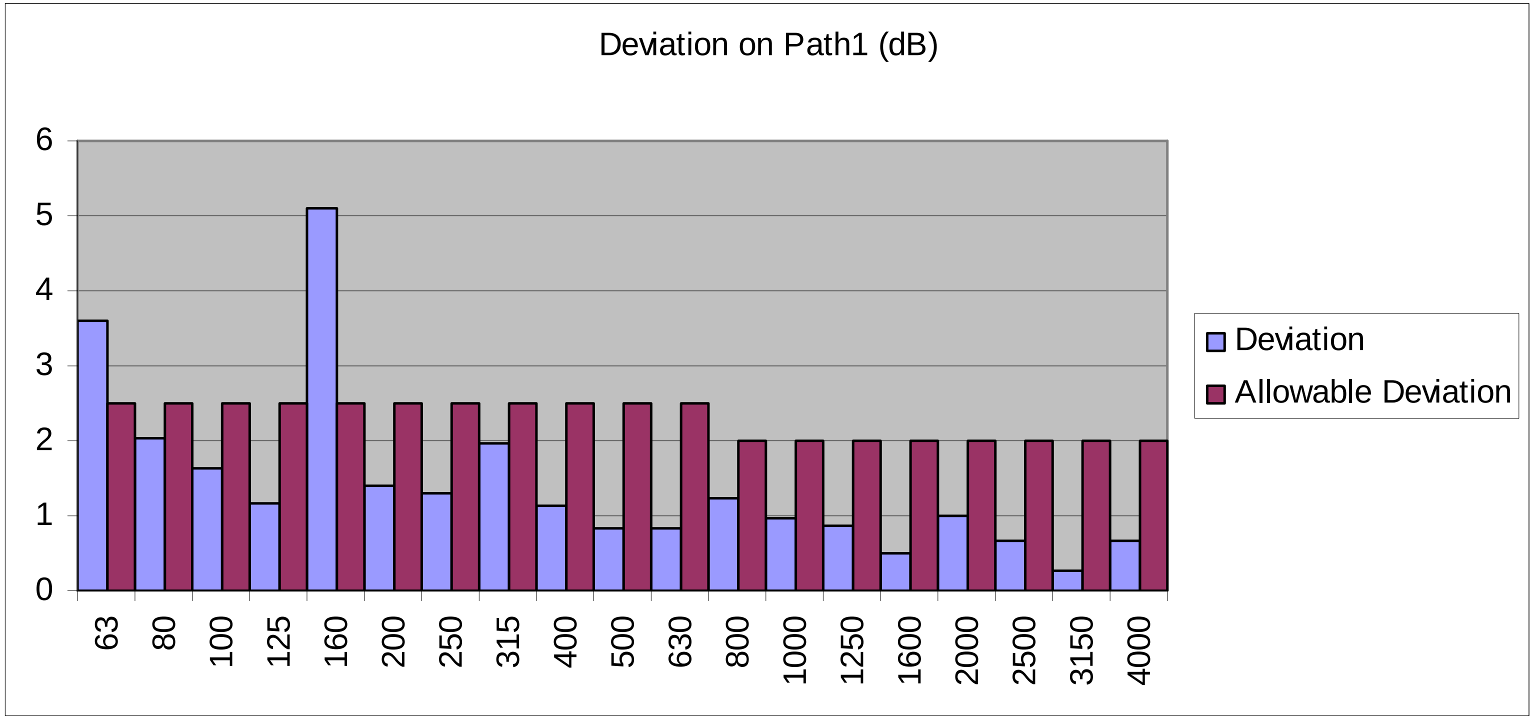


Figure 2.5. The Comparison between Calculated Deviations and Allowable Deviations

2.7.2. Estimation of Sound Pressure Levels Based on Inverse Square Law for Path 2:

The performance of an anechoic or semi-anechoic room is assessed by comparing the spatial decrease of sound pressure level emitted from a test source with the decrease of sound pressure level with distance from the source, according to the inverse square law that would occur in a true anechoic or semi-anechoic field.

From the sound pressure levels measured at the paths 1 and 4, the estimation of sound pressure levels based on the inverse square law shall be determined for “each direction of measurement” from the equation that has stated on ISO 3745.

Firstly for path 4, estimated sound pressure levels based on inverse square law is calculated. In order to obtain estimated sound pressure levels based on inverse square law, firstly q values for each point at each frequency must be calculated. Then for each frequency band and for path 4, r_0 and a values are calculated. Finally the estimated sound pressure levels based on inverse square law for path 4 and for each frequency band, is obtained as show on the sample calculations part. The calculated q values at each point at each frequency band can be seen in Table B5 (App. B). Also r_0 and a Values for path 4 on each frequency band ca n be seen in Table B6 (App. B). And also the calculated L_p values are shown in Table B7

Then the difference (deviation) between sound pressure levels and estimated sound pressure levels based on inverse square law at each frequency and point on path 4 is calculated. These calculated deviations are shown in Table B8 (App B)

Finally for each frequency band, maximum deviation from the inverse square law is obtained

Maximum deviations from inverse square law and conformance with table ISO 3475 are given in Figure A.2. Also the graph of the comparison of deviations can be seen in figure 2.6.

Table 2-8: The comparison of calculated deviations with allowable deviations for path 4

1/3 octave bands Frequency	Calculated ΔL_{pi} max	Allowable ΔL_{pi} max
63	3,70	2,5
80	2,97	2,5
100	2,67	2,5
125	0,98	2,5
160	2,25	2,5
200	0,67	2,5
250	1,25	2,5
315	2,11	2,5
400	1,01	2,5
500	1,10	2,5
630	0,68	2,5
800	0,65	2
1000	1,04	2
1250	2,06	2
1600	0,28	2
2000	0,53	2
2500	0,82	2
3150	0,37	2
4000	1,15	2

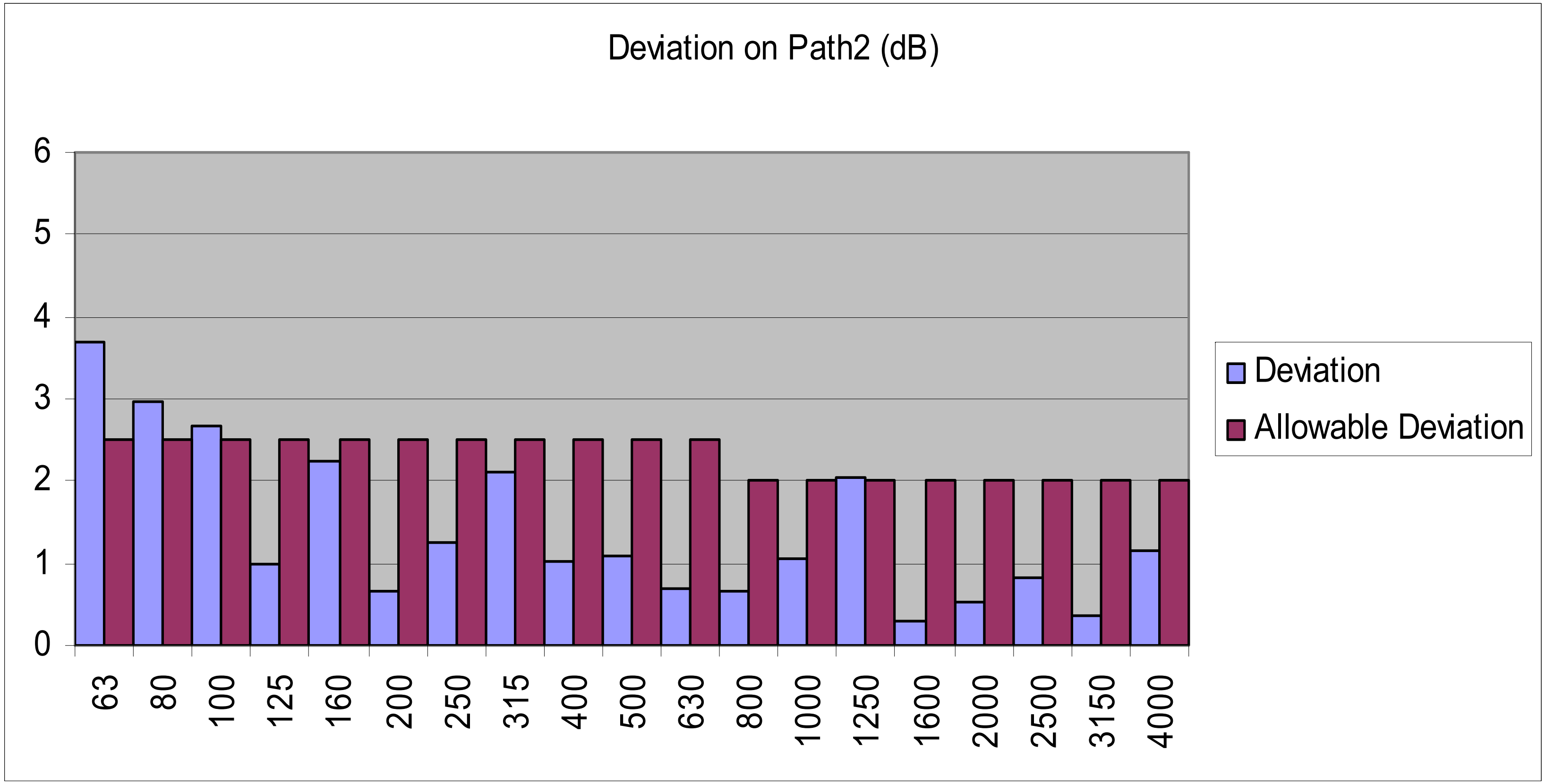


Figure 2.6. The Comparison between Calculated Deviations and Allowable Deviations (for Path-4).

As seen on the figure 2.5."According the calculations for path 1; semi-anechoic chamber in Fluid Mechanics Laboratory in Mechanical Engineering Department is in conformance with ISO 3745 between 63-4000 Hz; except the frequencies 63Hz and 160 Hz.

As seen on the graph 2.6."According the calculations for path 4; semi-anechoic chamber in Fluid Mechanics Laboratory in Mechanical Engineering Department is in conformance with ISO 3745 between 63-4000 Hz; except the frequencies 63Hz, 80 Hz, and 100 Hz

According to results obtained from path 1 and path 4; semi-anechoic chamber is in commence with ISO 3745 on the 1/3 frequency band between 160 Hz- 4000 Hz.

Also it was understood that near field radius of the chamber is less then 1m; since proper results were obtained with r_{min}: 1 m for each path.

CHAPTER 3

FLUID MECHANICS OF VACUUM CLEANER

3.1. Ggeneral

The vacuum cleaners work with the help of pressure difference created by a radial fan which causes a flow due to suction from the carpet (or cleaning surface) to the exhaust part of the machine.

There are many resistances to the flow of air during the processes which are in flow order from upstream to downstream:

- Cleaning surfaces itself, which is , most of the time, a carpet; a carpet during cleaning process acts as a reversed filter which filters the clean room air and leaves its dust, dirt and other particles in to the flowing air so that the carpet is cleaned up from unwanted particles and dirt. During this process there are head losses due to filtering effect of the carpet.
- The flow tubes from cleaning head to the dust bag in the machine which are not very effective in the head losses of flowing air.
- The dust bag itself is one of the major head loss points in the machine especially when its filled with dust and dirt, due to its filtering properties; the head loss occur when the air is passing through the bag surface, which is generally made up of paper.
- The first real filter is before the radial fan, which filters particles that might escape from the dust bag to vacuum tank and reach to the radial fan.
- The flow reaches to the radial flow fan and electric motor, which is the most important part of a vacuum cleaner, where there is a filter around the motor fan assembly, is also a reason head loss of.
- After the radial fan motor another filter comes this generates head loss as it filters the exhaust air and absorb some of the sound.
- The last head loss it due to the HEPA filter at the exhaust of the machine.

Since there are a number of head loss sources during the air flow the main need is not a high volumetric flow but a high head air flow, which brings the specific speed to a low value and as a result the radial flow fan is used to be for air flow in a vacuum cleaner.

The parts of a vacuum cleaner which functions described above are given in Appendix B.

3.2. Vacuum Cleaner Fan Characteristics

3.2.1. Determining Fan Characteristics Experimental Setup

To obtain the fan characteristics, one of the recommended setups in the AMCA 210-75 Standard was selected and designed according to the motor-fan assembly of the vacuum cleaner. The technical drawings of the fan characteristics set-up, the flow straightener and the pitot tubes are given in Figure 3.1, 3.2, 3.3. The final setup constructed in fluid mechanics laboratory is shown in Figure 3.5.

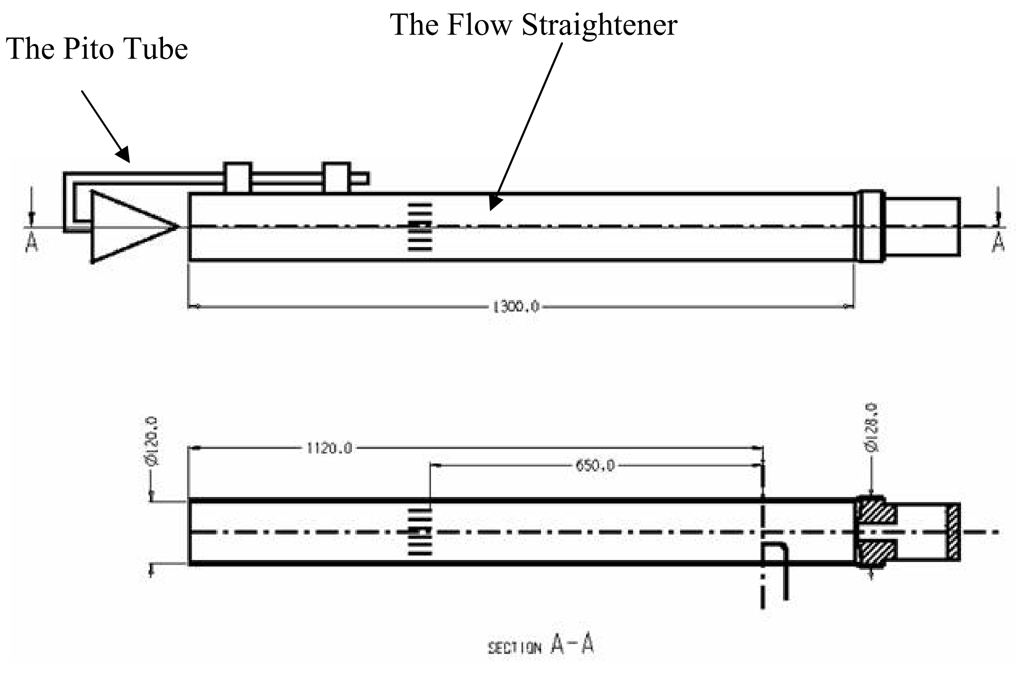


Figure 3.1. The Fan Characteristics Set-Up

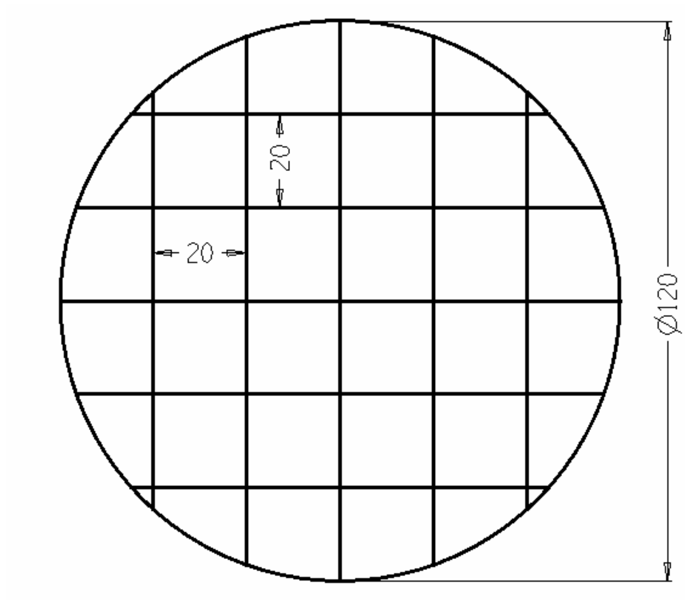


Figure 3.2. The Straightener

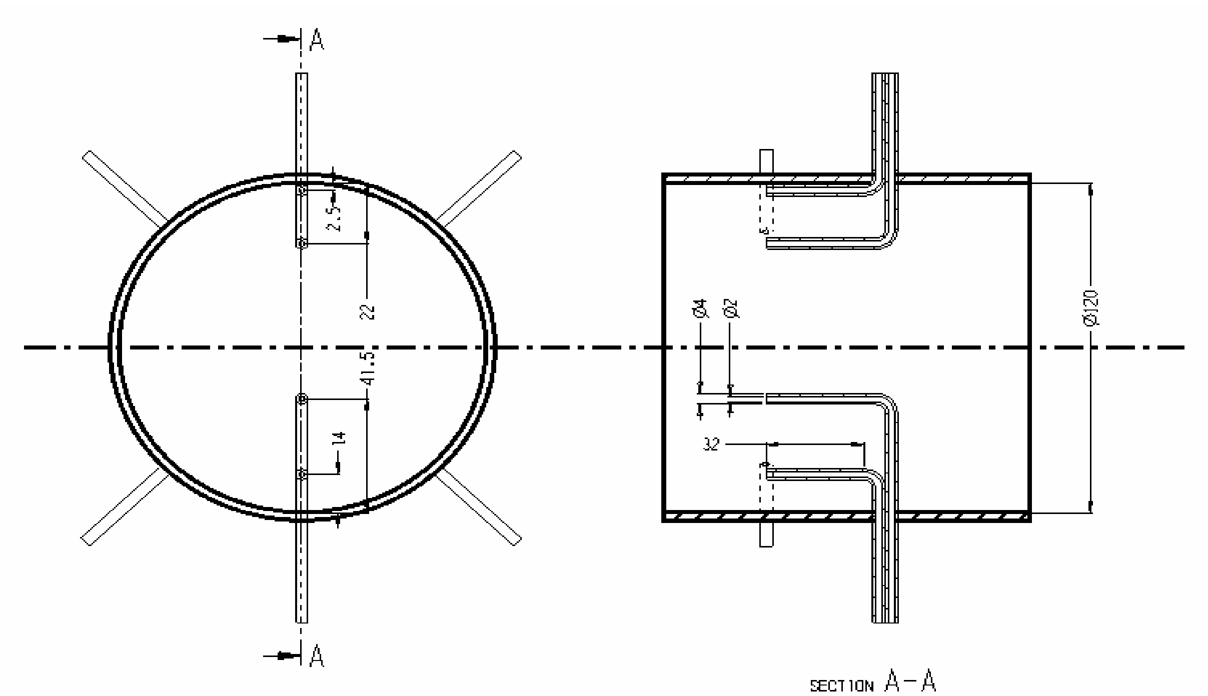


Figure 3.3. The Pitot Tube Locations

Figure 3.4. The Experimental Setup constructed in Fluid Mechanics Laboratory 35

Throttling device is two plates symmetrically to each other at the inlet of the setup. This enables to make fine adjustments since the fan is most sensitive when the inlet almost fully throttled. Adjusting plate slightly resulted in a change in the operating point.

To take the average of measurements of the 4 pitot tubes, the outlets are combined with joints to a one tube. Similarly 3 static pressure taps are attached to a single tube. In the end we had two tubes merge to give the stagnation pressure and the static pressure.
The flow rate is proportional to the dynamic pressure which is the pressure difference between the stagnation and static pressures.

$$P_v = \frac{1}{2}\rho V^2 . \qquad (3.1.)$$

Dynamic pressure is proportional to air density ρ which is around 1 kg/m^3. So extremely low differences are need to be measured. To do so we used an inclined manometer with 2° inclination, described in Figure 3.5.. Ethyl alcohol is used as manometer fluid.

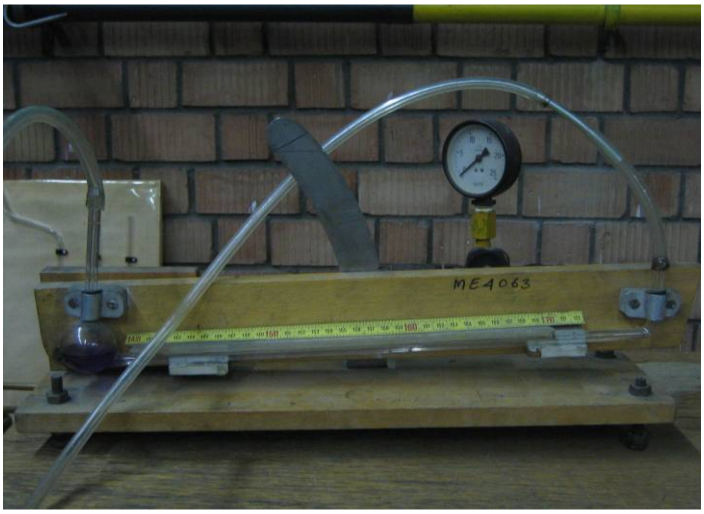


Figure 3.5. The Inclined Manometer

The angular velocity of the fan motor assembly is measured with a laser tachometer. To provide a constant rotational speed and to correct the motor speed to the desired one, after for every throttle point, the voltage source was adjusted.

The tests were performed for five different angular speed of fan respectively, 10000rpm, 14000rpm, 18900rpm, 25000rpm and 30000rpm.as described in the related standard. The results are given in Appendix B.

3.2.2 Fan Characteristic Test Results

The difference of the total pressure taken from the pitot tubes and static pressure from the static pressure taps, that is, the dynamic pressure, h_{dy} is evalueated.

$$P_{dyn} = \frac{1}{2}\rho_{air} V_{avg}^2 = \rho_{alc} \cdot g \cdot (h_{dy} - h_{dy,o}) \cdot \sin\theta \tag{3.2}$$

$$V_{avg} = \sqrt{2 \cdot \frac{\rho_{alc}}{\rho_{air}} \cdot g \cdot (h_{dy} - h_{dy,o}) \cdot \sin\theta} \tag{3.3}$$

Then the Volumetric flow rate is,

$$Q = \text{Area} \cdot V_{avg} = (r^2 \cdot \pi) \cdot \sqrt{2 \cdot \frac{\rho_{alc}}{\rho_{air}} \cdot g \cdot (h_{dy} - h_{dy,o}) \cdot \sin\theta} \tag{3.4}$$

The h_{tot} is the gage pressure and the Head that the fan produces is directly proportional to h_{tot} through the following formula,

$$H_{fan} = \frac{\rho_{water}}{\rho_{air}} \cdot h_{tot} \qquad (3.5.)$$

To find the efficiency at an specified point,

$$\eta = \frac{P_{out}}{P_{inp}} = \frac{\rho_{air} \cdot g \cdot Q \cdot H_{fan}}{V \cdot A} \tag{3.6}$$

Note that the above efficiency expression does not indicate the fan efficiency but the efficiency of the fan-motor assembly.

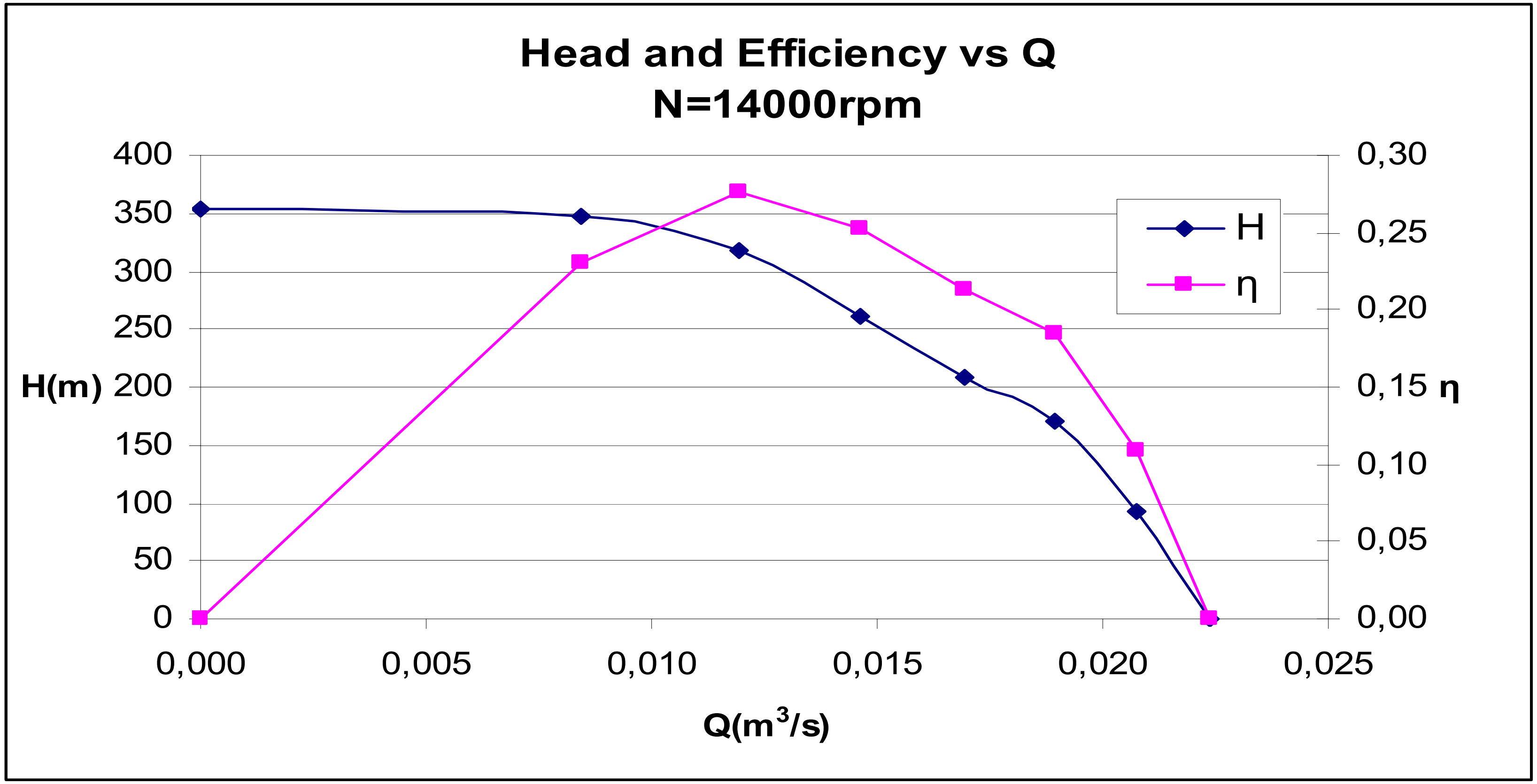
Employing these three, Q, H_{fan} and η can be found and they are shown at the below table for the case N=14000rpm.

Table 3-1 Calculated Q, H_{fan} and η

h_{dy}(cm)	Δh_{tot}(cm)	Current(A)	Voltage(V)	Q(m3/s)	H(m)	η
2,3	0	2,8	68	0,0224	0,0	0,00
2,2	9,6	2,7	66	0,0207	93,2	0,11
2,1	17,5	2,7	65	0,0189	169,8	0,18
2	21,4	2,6	64	0,0169	207,7	0,21
1,9	26,8	2,5	61	0,0147	260,1	0,25
1,8	32,8	2,4	58	0,0120	318,3	0,28
1,7	35,9	2,3	56	0,0085	348,4	0,23
1,6	36,4	1,8	47	0,0000	353,3	0,00

The fan characteristic curves can be plotted as follows: The test results and related figures are given in a detailed manner in Appendix B.

As seen on Figure 3-6 the best efficiency point is approximately 0.012 m^3 / s.


Head and Efficiency vs Q
N=14000rpm
H(m)
η
H
η
Q(m3/s)
400
350
300
250
200
150
100
50
0
0,30
0,25
0,20
0,15
0,10
0,05
0,00
0,000
0,005
0,010
0,015
0,020
0,025

CHAPTER 4

ACOUSTICAL CHARACTERISTICS OF A VACUUM CLEANER

4.1 Experimental Procedure

4.1.1. Experimental Set-Up

Since the measurements will be taken in a semi anechoic room, the measurement surface will be a semi sphere. ISO 3745 Standards recommends that the radius of the test hemi sphere should be either 1m or twice the largest source dimension. For the case of interest, the vacuum cleaner is the noise source and radius of hemi sphere is taken as 1 m.

To have the microphone positions associated with equal partial areas of the test hemisphere, ISO 3745 Standards recommends the maximum 20 microphone positions. To facilitate the measurements it was determined 8 positions according to the microphone positions shown in Figure 4.1. The microphone positions are taken according to the availability and accuracy of the positions. Theses positions are chosen from the main 20 position defined in Figure 4.1. and standards.

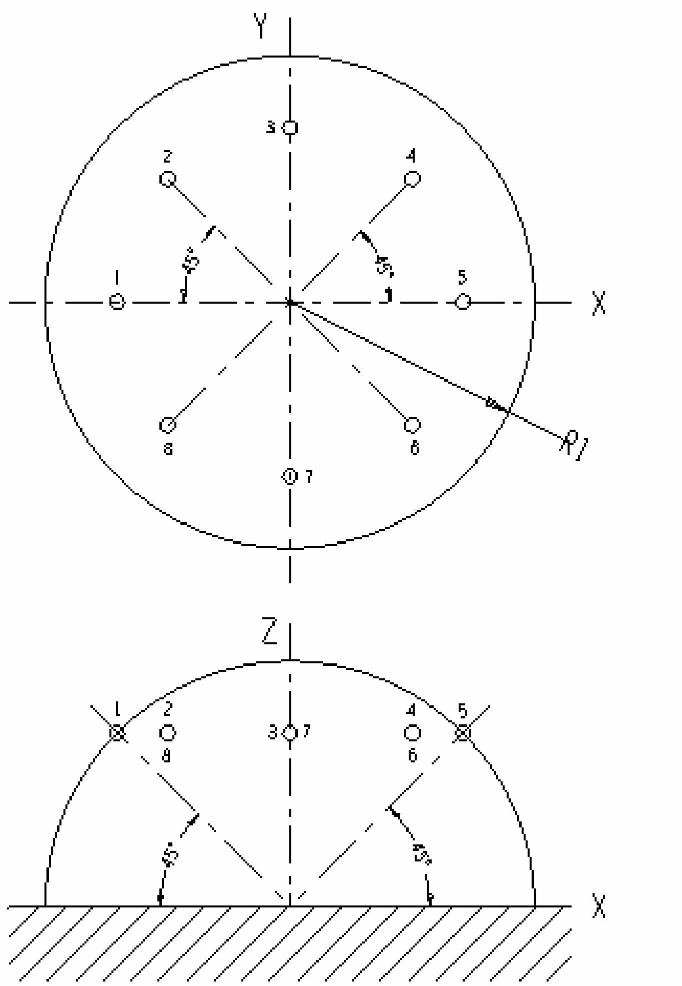


Figure 4.1. The Microphone Positions Described in Standards

There are two microphones in the semi anechoic room. During the measurements, they are placed opposite to each other, and the vacuum cleaner is rotated 45° at each measurement period to simulate the standard. The noise of a specific situation of the vacuum cleaner is measured in four measurement periods. The configuration photos of the 4 measurement periods are shown in the figure.

Angle 0° *Angle 45°*

Figure 4.2.a. The Experimental Setup and the Positions of Microphones

Angle 90°

Angle 135°

Figure 4.2.b. The Experimental Setup and the Positions of Microphones

4.1.2 Experimental Procedure

Regardless of whose noise is measured in the anechoic room, the noise measurement procedure is the same. First put the vacuum cleaner into the set-up in "Angle 0°" case. Stop the machine and take the background noise data for 30 seconds. Then start the vacuum cleaner and take the noise data for 30 seconds. Repeat the procedure for the other three cases (Angle 45º, 90º and 135º) except the background noise measurement part. In the middle of the experiment, say after angle 45º, measure the temperature in the anechoic room.

4.2. Noise Measurement Results

When the sound spectra is drawn as sound pressure level versus frequency, it can be seen that at the frequencies f_{imb} (fundamental frequency due to rotary unbalance), *bpf* (fundamental blade passing frequency) and their harmonics, that is, the integer multiples of these frequencies, are dominant in the form of peaks. Utilizing those one can find the angular velocity of the fan. Unfortunately, the peaks are not clear in 1/3 octave band, so another set of data consisting of narrow band spectral analysis must be taken for the frequencies increasing 50 Hz at every step up to 20000 Hz. Repeat this for the states of vacuum cleaner that were investigated in the anechoic room.

The noise data was always taken at 8 locations that were described before. The data can be processed according to ISO 3745 Standard as described below.

First background noise correlation should be performed on sound pressure levels.

For the i^{th} microphone position,

L_{pi} : corrected sound pressure level $\quad$ L'_{pi} : the measured sound pressure level

L''_{pi} : background sound pressure level $\quad$ K_{1i} : background noise correction

Then $\Delta L_i = L'_{pi} - L''_{pi}$ and $L_{pi} = L'_{pi} - K_{1i}$ is applied. K_{1i} is calculated at every frequency according to standards as below.

$$\text{if } \Delta L_i > 20\text{dB} \rightarrow \text{No correction}$$

$$\text{if } 20\text{dB} > \Delta L_i > 10\text{dB} \rightarrow K_{1i} = -10 \cdot \log(1 - 10^{-0.1\Delta L_i})\text{dB}$$

$$\text{if } 10\text{dB} > \Delta L_i \rightarrow \text{the maximum correction to be applied shall be 0.5 dB.}$$

For the last case 0.5 dB correction is to be applied.

The noise measurement results of the vacuum cleaner are given in Appendix X.

The data taken from channel 1 and channel 2 was corrected with silent channel 1 and channel 2 respectively

Since the microphone positions are associated with equal partial areas of the test hemisphere, the following equation is used to obtain the surface sound pressure level, L_{pf}, at every frequency:

$$L_{pf} = 10 \cdot \log(\frac{1}{N}\left[\sum_{i=1}^{N} 10^{0.1 \cdot L_{pi}}\right] \tag{4.1}$$

where N is the number of microphone positions.

To obtain the surface sound pressure level over the test hemi sphere,

$$\overline{L_{pf}} = 10 \cdot \log \sum 10^{0.1 \cdot L_{pf}} \text{ dB} \tag{4.2}$$

L_{pf} and $\overline{L_{pf}}$ are calculated according to the equations given above.

To obtain the surface sound power levels, L_{Wj}, at every frequency the following equation is used.

$$L_{Wj} = L_{pf} + 10 \cdot \log\left(\frac{S_2}{S_0}\right)\text{dB} + C_1 + C_2 \tag{4.3}$$

where

$$C_1 = -10 \cdot \log\left[\frac{B}{B_0} \cdot \sqrt{\frac{313.15}{273.15+\theta}}\right] dB \tag{4.4}$$

$$C_2 = -15 \cdot \log\left[\frac{B}{B_0} \cdot \left(\frac{296.15}{273.15+\theta}\right)\right] dB \tag{4.5}$$

where

$S_2 = 2 \cdot \pi \cdot r^2$ is the area of the test hemisphere (of radius r);

$S_0 = 1\,m^2$;

B is the measured barometric pressure during the measurements, in pascals;

B_0 is the reference barometric pressure, $1.01325 \times 10^5\,Pa$

θ is the air temperature during measurements, in ^oC.

Finally, A-weighted sound power level is calculated according to the following equation.

$$L_{WA} = 10 \cdot \log \sum_{j=j_{min}}^{j_{max}} 10^{0.1 \cdot (L_{Wj}+C_j)} dB \tag{4.6}$$

Where L_{Wj} is the sound power level in the j^{th} 1/3 octave band (in dB). C_j is A-weighted values

j_{min}, j_{max} are the values of j corresponding to the lowest and highest frequency bands of measurement.

Employing the above definitions, the noise measurement results are presented in Table 4.1 and Figure 4.3, and Figure 4.4.

To find the angular velocity of the fan, one should plot L_{pf} versus frequency of the noise data taken for the frequency increasing 50 Hz successively. Employing the similar procedure given for 1/3 octave band,

Blade Passing Frequency Comparison and Sample Calculation

The three peaks are clear on the graph and as it is seen they are the multiples of 5400 Hz. So most probably the blade passing frequency is 5400 Hz. Noting that on the fan there are 9 blades,

$$f_{bpf} = \frac{N \times Z}{60} \Rightarrow 5400 = \frac{N \times 9}{60} \Rightarrow N = 36000\text{rpm} \tag{4.7}$$

So the fan is rotating at 36000 rpm for the case Vacuum Cleaner with all parts together.

For the volumetric flow rate, the dynamic pressure was measured at the hose. The dimensions and the measurements are summarized below.

T = 19.5 ºC and P_{atm} = 655 mm Hg.

$h_{dyn} = 11\text{cm of water}$ and ρ_{water} = 998 kg/m^3

$D_{hose} = 3.6\text{cm}$

Assuming that the air is an ideal gas at this pressure,

$$\rho_{air} = \frac{P_{amb}}{R\,T_{amb}} = \frac{0.655 \times 13600 \times 9.81}{287.1 \times (19.5 + 273)} = 1.041\,\text{kg}/\text{m}^3 \tag{4.8.}$$

Dynamic pressure h_{dy} is related to the air velocity V_{avg} with the following equation.

$$P_{dyn} = \frac{1}{2}\rho_{air} V_{avg}^2 = \rho_{water} \cdot g \cdot h_{dyn} \tag{4.9}$$

$$V_{avg} = \sqrt{2 \cdot \frac{\rho_{water}}{\rho_{air}} \cdot g \cdot h_{dyn}} \tag{4.10}$$

Then the Volumetric flow rate is,

$$Q = \text{Area} \cdot V_{avg} = (r^2 \cdot \pi) \cdot \sqrt{2 \cdot \frac{\rho_{water}}{\rho_{air}} \cdot g \cdot h_{dyn}}$$

$$Q = \left(0.018^2 \cdot \pi\right) \times \sqrt{2 \cdot \frac{998}{1.041} \cdot 9.81 \cdot 0.11} = 0.0463 \ \text{m}^3/\text{s}$$

Finally the following table is obtained;

Table 4.1. Vacuum Cleaner Noise Results

$\overline{L_{pf}}$ (dB)	L_{WA}(dB)	Q(m^3/s)	N(rpm)
77.08	85.66	0.0463	36000

Table 4-2 Noise Measurements and Results

T(°C)=	22	C_1=	0,514
P(mmHg)=	655	C_2=	0,942

!/3 Octave Band Frequency (Hz)	L_{pf}	Lw(dB)	C_j	L_{WAj}(dB)	L_{WA}
20	48,3	57,7	-50,5	7,2	85,7
25	39,7	49,1	-44,7	4,4	
31,5	33,8	43,3	-39,4	3,9	
40	32,2	41,6	-34,6	7,0	
50	28,3	37,7	-30,2	7,5	
63	34,8	44,3	-26,2	18,1	
80	34,2	43,7	-22,5	21,2	
100	49,4	58,9	-19,1	39,8	
125	42,9	52,4	-16,1	36,3	
160	49,4	58,9	-13,4	45,5	
200	51,5	61,0	-10,9	50,1	
250	54,3	63,8	-8,6	55,2	
315	55,1	64,6	-6,6	58,0	
400	57,8	67,2	-4,8	62,4	
500	59,8	69,3	-3,2	66,1	
630	64,9	74,3	-1,9	72,4	
800	64,5	74,0	-0,8	73,2	
1000	62,2	71,6	0,0	71,6	
1250	62,0	71,5	0,6	72,1	
1600	63,4	72,8	1,0	73,8	
2000	64,5	73,9	1,2	75,1	
2500	65,0	74,5	1,3	75,8	
3150	65,4	74,8	1,2	76,0	
4000	64,5	73,9	1,0	74,9	
5000	64,5	73,9	0,5	74,4	
6300	64,4	73,8	-0,1	73,7	
8000	66,1	75,5	-1,1	74,4	
10000	67,0	76,5	-2,5	74,0	
12500	66,2	75,6	-4,3	71,3	
16000	66,0	75,4	-6,6	68,8	
20000	63,3	72,7	-9,3	63,4	

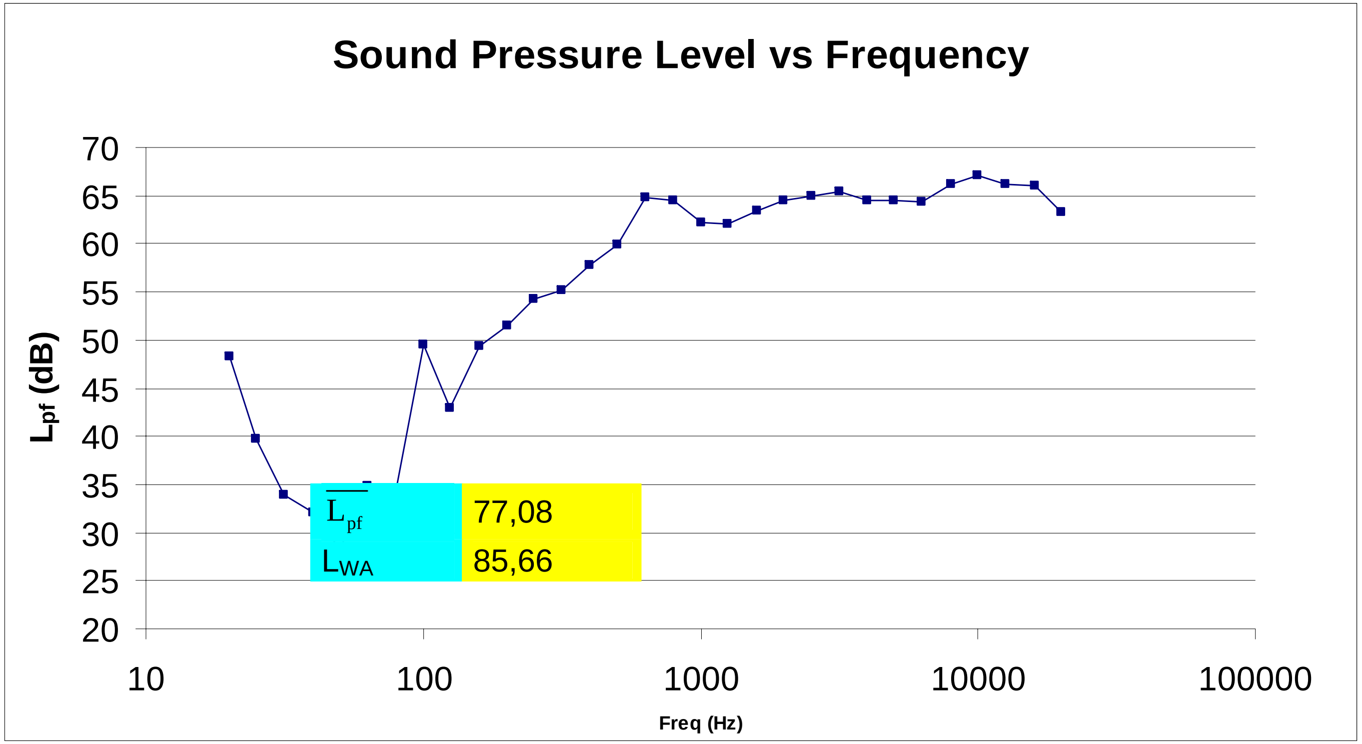


Figure 4-3 Sound Pressure Level Distribution of the Vacuum Cleaner

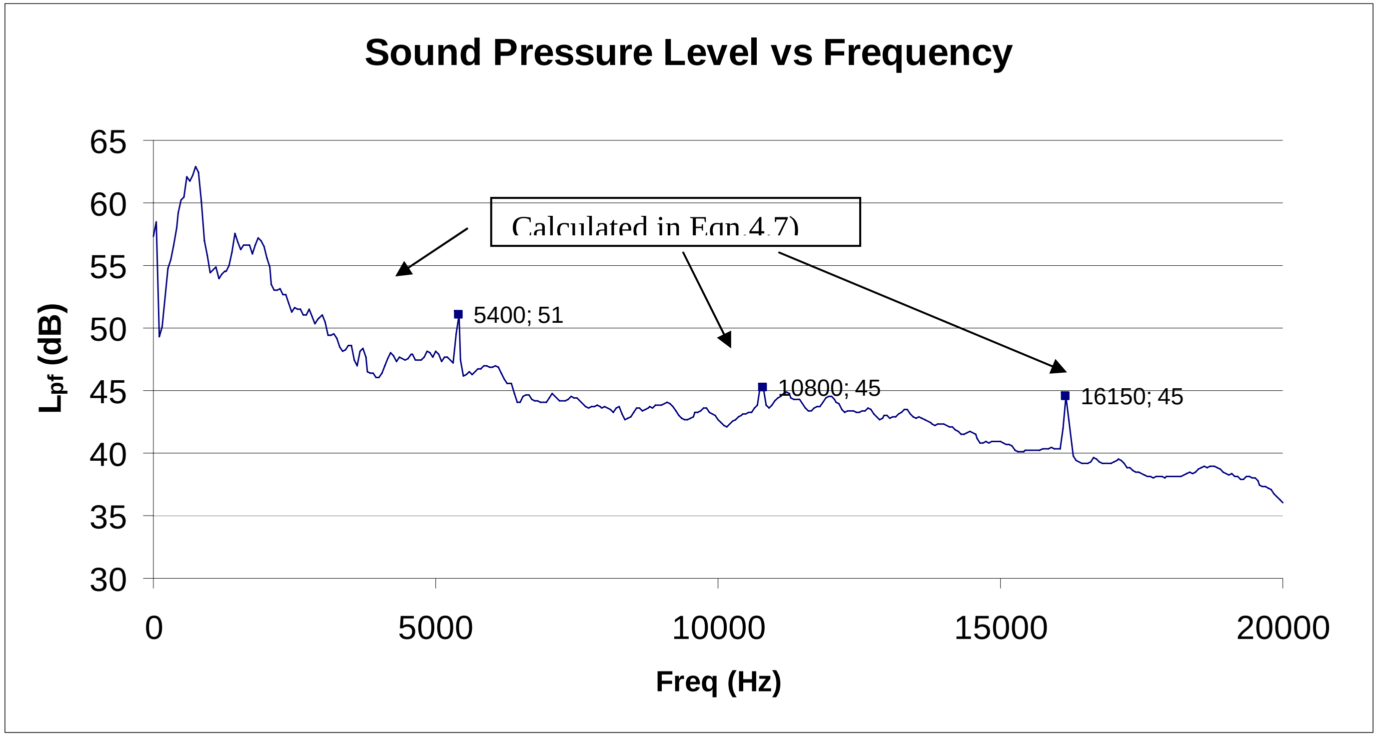


Figure 4.4. Sound Pressure Level Distribution including Blade Passing Frequency

CHAPTER 5

CASE STUDIES

5.1. Definition of the Cases

Sample case studies are performed, to investigate the effect of fan in a vacuum cleaner.

5.1.1. Case 1

Comparison of Sound Power Levels of the prototype vacuum cleaner and the final commercial product.

Figure 5.1. Prototype and Commercial Products in the Semi,-anechoic chamber

In this case the noise measurement of two different vacuum cleaner are performed and these results are compared.

5.1.2. Case 2

Comparison of the effects of the Back Cover, Hepa Filter and the Sponge on the vacuum cleaner noise generation.

Cover+Hepa--Sponge

Cover--Hepa+Sponge

Cover--Hepa--Sponge

--Cover—Hepa--Sponge

Figure 5.2. Different Configurations for Noise Measurement

In case 2 a signature analysis for vacuum cleaner component is perform***ed.***

5.1.3. Case 3

Comparison of the effects of the inside coating of the motor casing.

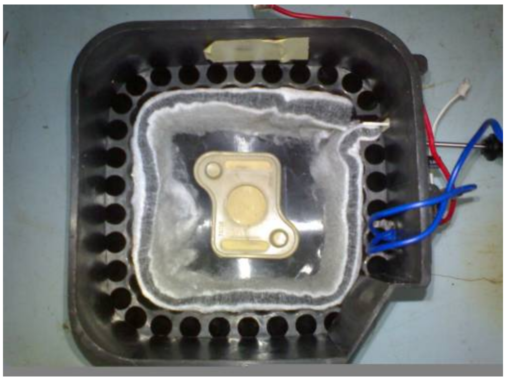


Figure 5-3.a Motor Casing with the original coating

Figure 5.3.b Motor Casing without coating on the left; Motor Casing coated with the special black sponge on the right

5.1.4. Case 4

Comparison of the effects of the fullness of the dust bag

To observe the effect of the dust bag on overall noise generation, and to simulate the operating condition of a vacuum cleaner the dust bag was filled with a material (white, linen threads) in different amounts (28g, 51.7g and 104.7 g)

After noise measurements, it is also crucial to find the operating point of the fan. To specify the operating point, it is enough to specify the rotational velocity of the fan and the volumetric flow rate.

When the sound pressure level versus frequency graph is observed, it can be seen that at the frequencies f_{imb} (fundamental frequency due to rotary imbalance), f_{bpf} (fundamental blade passing frequency) and their harmonics, that is, the integer multiples of these frequencies, the plot makes peaks. Unfortunately the peaks are not clear in 1/3 octave band, so another set of data must be taken for steps of frequencies increasing 50 Hz at every up to 20000 Hz..

Volumetric flow rate can be measured after the noise measurement experiments. For the all parts of the vacuum cleaner that are investigated in the anechoic room, measure the volumetric flow rate with a pitot tube. Here it is necessary not to disturb the operating point of the fan while inserting the pitot tube.

5.2. Noise Measurement Results

5.2.1. Case 1

Comparison of Sound Power Levels of the prototype vacuum cleaner and the final commercial product.

Table 5-1.a Ambient Conditions

	Prototype	Commercial Product	
P_{amb}	655	655	(mmHg)
T	20,5	22	(°C,for the noise measurement at the anechoic room)
T	19,5	19,5	(°C,for the flow rate measurement with pitot tube)
h_{dyn}	8,1	11	(cm of water)
D_{hose}	3,6	3,6	(cm)
ρ_{water}	998	998	(kg/m^3)

1/3 octave band frequency spectrum of the prototype and the commercial product is given in Appendix B.1. The narrow band spectrum with 50 Hz frequency resolution is given in Appendix B.2.

Table 5-1.b. Noise Measurement Comparison

	$\overline{L_{pf}}$ (dB)	L_{WA}(dB)	Q(m^3/s)	N(rpm)
Prototype	76.76	85.41	0.040	36000
Commercial Product	77.08	85.66	0.046	36000

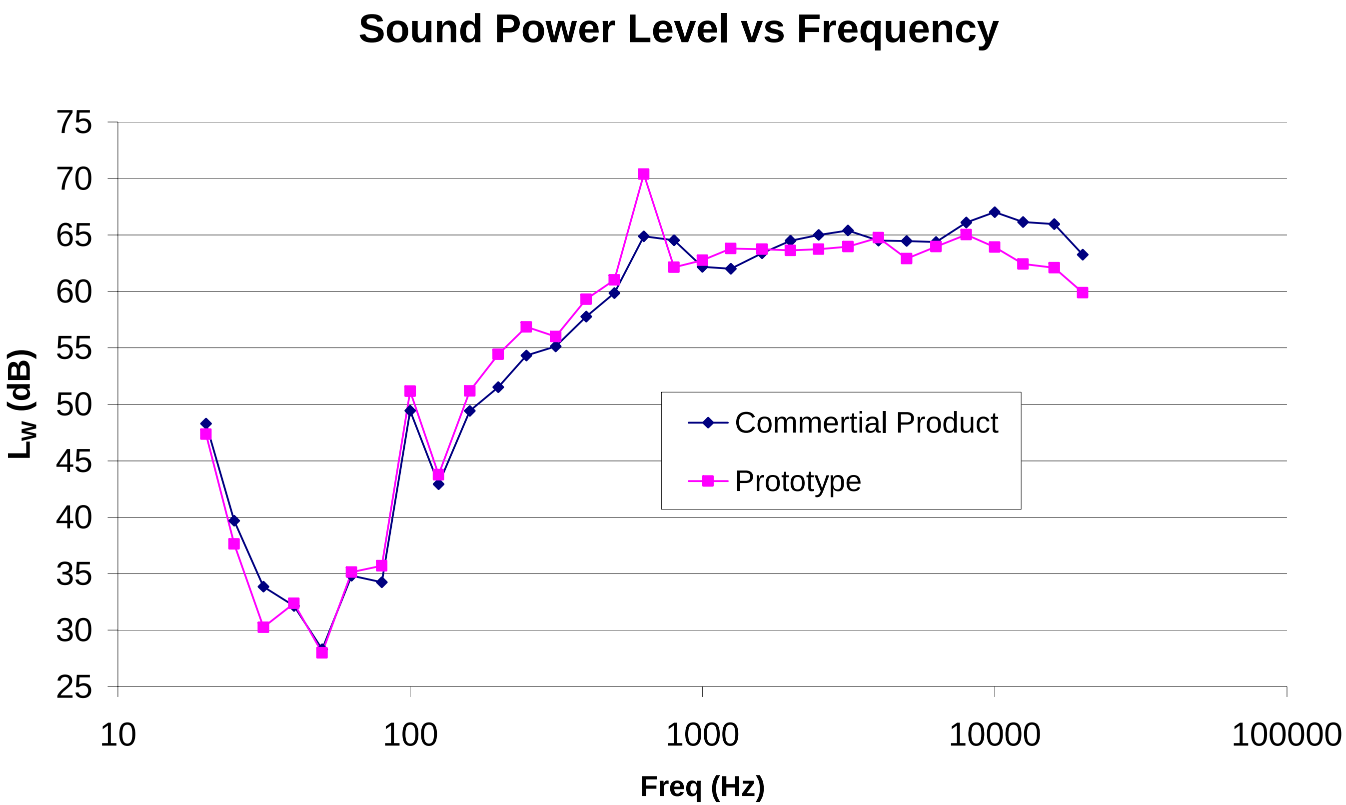


Figure 5.4. The Sound Power Level Spectra of Prototype and Commercial Product

5.2.2. Case 2

Comparison of the effects of the Back Cover, Hepa Filter and the Sponge on the vacuum cleaner noise production.

Table 5-2.a Ambient Conditions

	Commercial Product(c+h+s)	c+h-s	c-h+s	c-h-s	-c-h-s	
P_{amb}	655	655	655	655	655	(mmHg)
T	22	22	22	22	22	(°C,for the noise measurement at the anechoic room)
T	19,5	19,5	19,5	19,5	19,5	(°C,for the flow rate measurement with pitot tube)
h_{dyn}	11	11,3	12,9	13,1	13,2	(cm of water)
D_{hose}	3,6	3,6	3,6	3,6	3,6	(cm)
ρ_{water}	998	998	998	998	998	(kg/m^3)

Here “c” stands for the cover, “h” stands for the heap filter and “s” stands for the black sponge. “+” and “–” represents whether that part is present or not.
1/3 octave band frequency spectra of the five cases is given in Appendix B.3. The narrow band spectra with 50 Hz frequency resolution is given in Appendix B.4, 5, 6, 7 and 8.

Table 5-2.b. Noise Measurement Comparison

	$\overline{L_{pf}}$ (dB)	L_{WA}(dB)	Q(m^3/s)	N(rpm)
c+h+s	77.08	85.66	0.046	36000
c+h-s	76.95	85.40	0.047	36222
c-h+s	77.95	86.16	0.050	36333
c-h-s	78.32	86.61	0.051	36333
-c-h-s	79.00	87.22	0.0507	36333

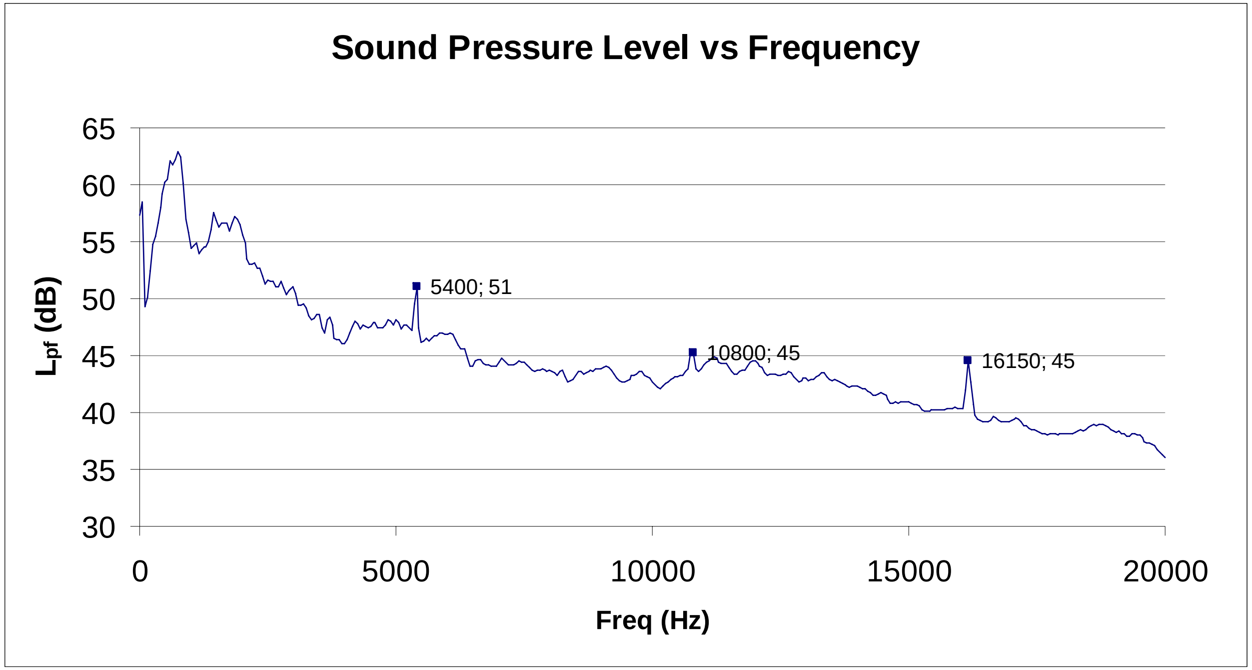


Figure 5.5. FFT of Case-2

5.2.3. Case 3

Comparison of the effects of the inside coating of the motor casing.

Table 5-3.a Ambient Conditions

	with original coating	without coating	coated with the special sponge	
P_{amb}	655	655	655	(mmHg)
T	22	22.5	21.5	(°C,for the noise measurement at the anechoic room)
T	19,5	19,5	19,5	(°C,for the flow rate measurement with pitot tube)
h_{dyn}	11	11	11	(cm of water)
D_{hose}	3,6	3,6	3,6	(cm)
ρ_{water}	998	998	998	(kg/m^3)

1/3 octave band frequency spectra of the five cases is given in Appendix B.9. The narrow band spectrum with 50 Hz frequency resolution is given in Appendix B.4 (since the same situation). The coating of the motor casing does not affect the operating point of the fan since it is not a pressure loss source. The air pumped by the fan passes through the rotor and leaves the rotor through the two holes at the back of the motor. Consequently h_{dyn} and the N are the same.

Table 5-3.b. Noise Measurement Comparison

	$\overline{L_{pf}}$ (dB)	L_{WA}(dB)	Q(m^3/s)	N(rpm)
with original coating	77.08	85.66	0.046	36000
Without coating	77.75	86.88	0.046	36000
coated with the special sponge	76.14	85.21	0.046	36000

5.2.4. Case 4

Comparison of the effects of the dust bag

Table 5-4.a Ambient Conditions

	Empty	28gr	51.7gr	104.7gr	
P_{amb}	655	655	655	655	(mmHg)
T	22	23	22	22	(°C,for the noise measurement at the anechoic room)
T	19.5	19.5	19.5	19.5	(°C,for the flow rate measurement with pitot tube)
h_{dyn}	11	11	11.1	11.1	(cm of water)
D_{hose}	3,6	3,6	3,6	3,6	(cm)
ρ_{water}	998	998	998	998	(kg/m^3)

1/3 octave band frequency spectra of the five cases is given in Appendix B.10. The narrow band spectrum with 50 Hz frequency resolution is given in Appendix B.4. The narrow band spectrum with 50 Hz frequency resolution is given in Appendix B.11.

Table 5-4.b. Noise Measurement Comparison

	$\overline{L_{pf}}$ (dB)	L_{WA}(dB)	Q(m^3/s)	N(rpm)
empty	77.08	85.66	0.046	36000
28gr	76.30	85.40	0.046	36000
51.7gr	76.38	85.61	0.047	36000
104.7gr	76.25	85.49	0.047	36000

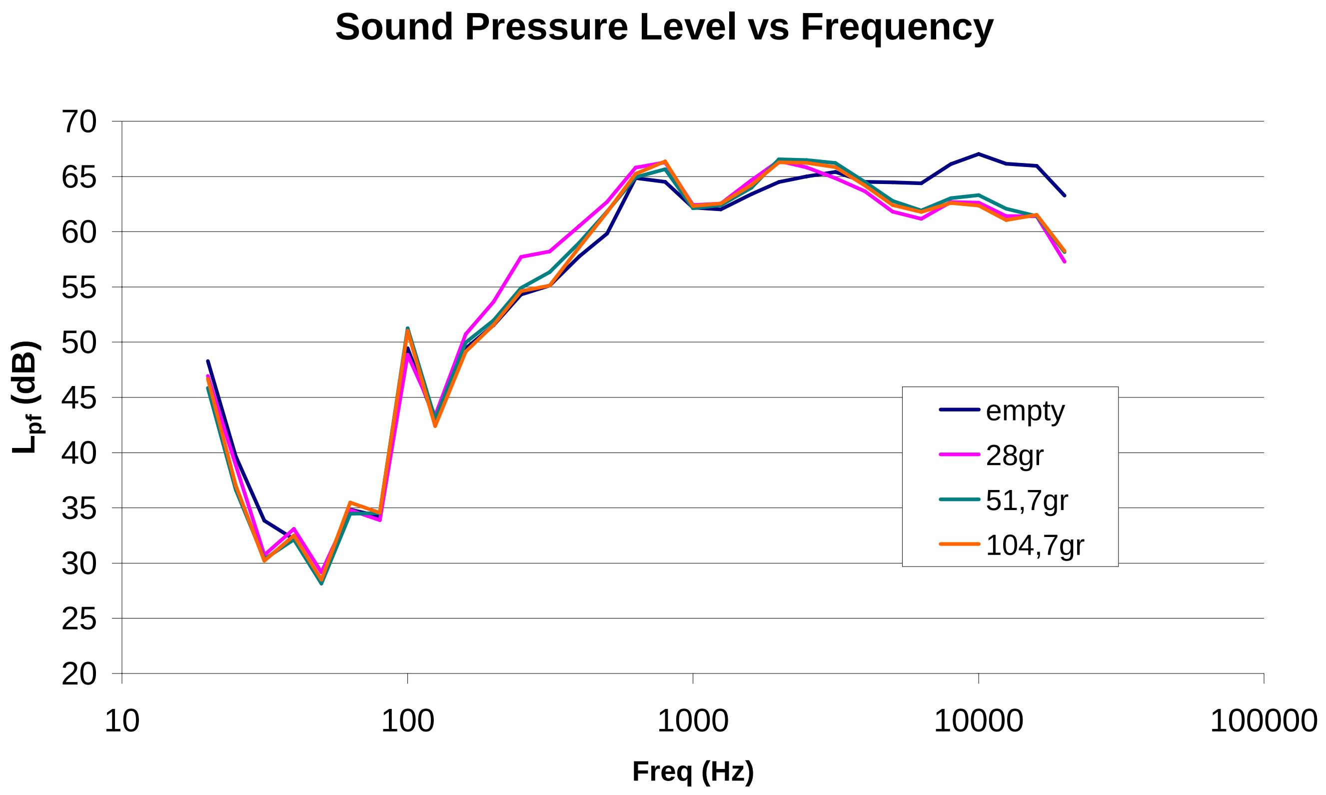


Figure 5.6. 1/3 Octave Band Sound Pressure Level Distributions

CHAPTER 6

DISCUSSION AND CONCLUSIONS

In the scope of this study a methodology is developed for determining the acoustical characteristics of a vacuum cleaner. This methodology is applicable for similar household appliances.

6.1. The Semi Anechoic Room

In this study a semi-anechoic room is designed to get reliable acoustical and noise data. The room is designed according to the international standard of ISO 3745. A qualification procedure for the room is applied and it is concluded that the acoustical test results handled in the semi anechoic room gives sufficient and acceptable test results in a defined range.

Background sound pressure level and measured sound pressure levels for each path is compared for each path. If the maximum deviation on each path for each frequency band is higher than the allowable values that has been stated on ISO 3745 table A.2; then at that frequency band semi-anechoic chamber doesn’t fulfill the hemi-anechoic conditions stated by ISO 3745.

For path 1; 1/3 octave frequency bands of 63, 160 Hz, for path 4; 1/3 octave frequency bands of 63, 80, 100 Hz doesn’t satisfy the requirement. However, the remaining frequencies are in qualified according to the standard.

According to results obtained from path 1 and path 4; semi-anechoic chamber is qualified with ISO 3745 on the 1/3 frequency band between 160 Hz- 4000 Hz.

Cut-off frequency of the chamber is 160 Hz. This implies below this frequency band the room doesn’t comply with standards hemi-anechoic properties.

Also it was understood that near field radius of the chamber is less then 1m; since proper results were obtained with r_{min} is equal to 1 m for each path.

6.2. Vacuum Cleaner

The noise characteristics of the vacuum cleaner is measured by a set of experiments

Firstly, in order to determine the operating point and steady state point of the vacuum cleaner the flow characteristics of the fan is investigated. Then, the noise are handled considering the operating point of the fan.

In this study, it can be seen that rotating machinery noise is dominated by tones at the *blade passing frequency (bpf)*, which is defined as the frequency of blades passing a stationary point, and higher harmonics. Besides impeller blades, shaft rotation and rotating components of driving motor are also considered as sources of noise.

According to the noise data, the noise generated by a vacuum cleaner is fan based. However, taking precautions it is possible to decrease the noise of a vacuum cleaner. Operating the vacuum cleaner at the fan's operating point is one of them. Choosing appropriate components such as heap filter etc. which are head loss sources for vacuum cleaner could be helpful. On the other side, application involving internal coating with absorptive materials helps to reduce the noise of vacuum cleaner.

Although some precautions could be taken, the dominant noise source of a vacuum cleaner is its casing excited at blade passing frequency. The structural resonance frequencies of the casing of the vacuum cleaner should be kept as far as possible to the blade passing frequency of the fan.

6.3. Recommendation for Future Work

Dimensions of the test room are sufficient for near field measurements but for far field measurements, a larger test room may be used. Smaller sound sources may be detected for far field noise levels, in the present test room.

Two microphones are not enough for proper measurements, because one should change the microphone positions before each measurement. Four or more microphones will give more accurate results.

In future studies, complex time histories can be processed with any common mathematical software in order to calculate coherency of two separate measurements.

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APPENDIX A

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Figure A1. Semi Anechoic Room Consturction Stages

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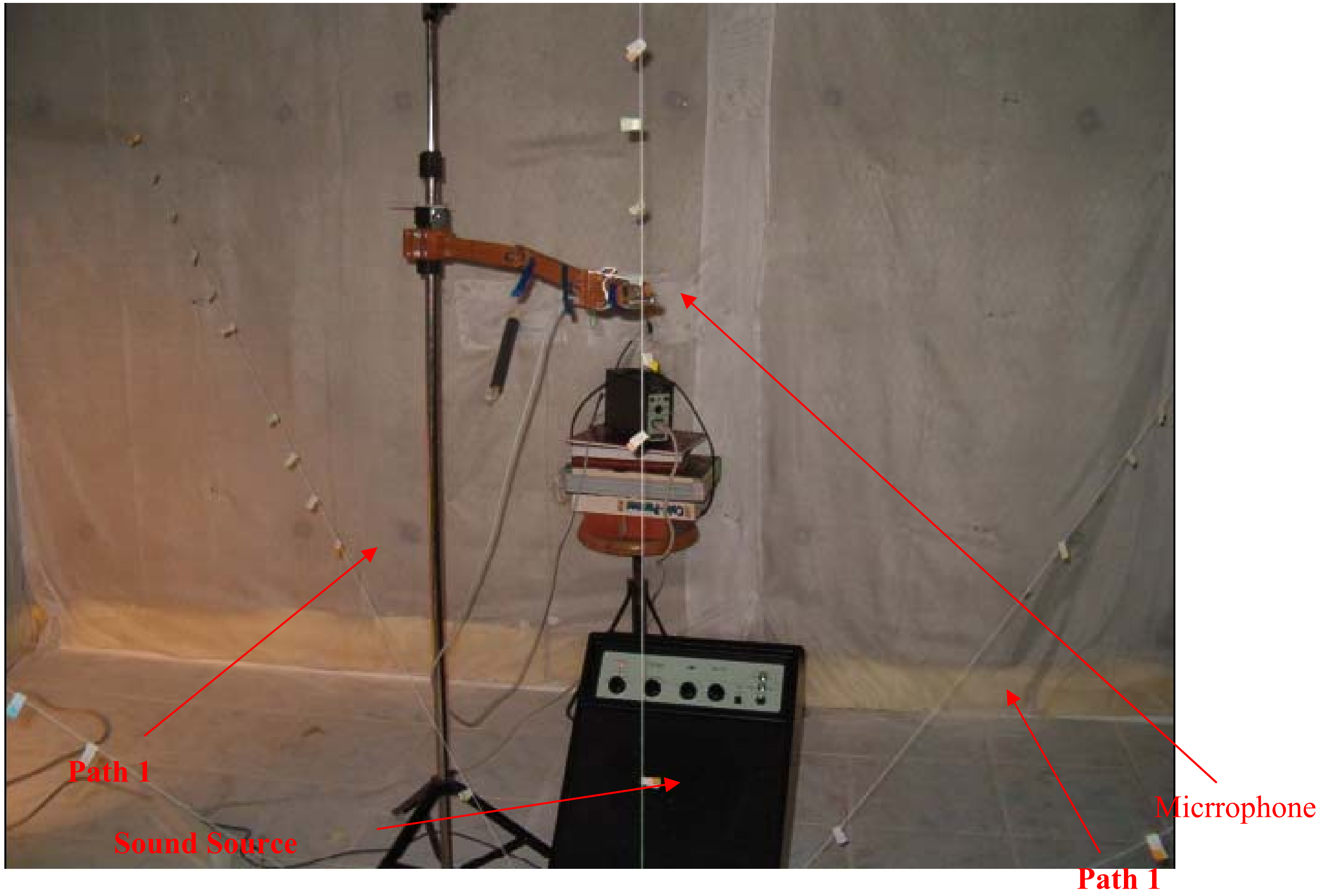

Figure A.2. Qualification Set-up for the Room

TAble A-1 Qualicification of Semi-Anechois Room

Frequency	P1	P2	P3	P4	P5	P6	Q1	q2	q3	q4	q5	q6
63	78,82	74,89	75,35	68,86	67,48	65,57	0,000115	0,00018	0,000171	0,000361	0,000423	0,000527
80	83,99	78,22	79,32	76,87	75,45	73,36	6,32E-05	0,000123	0,000108	0,000143	0,000169	0,000215
100	94	88,06	85,66	83	79,13	79,25	2E-05	3,95E-05	5,21E-05	7,08E-05	0,000111	0,000109
125	99	93,1	92	89,98	86,87	86,65	1,12E-05	2,21E-05	2,51E-05	3,17E-05	4,53E-05	4,65E-05
160	98,13	95,81	91,57	90,96	86,06	84,07	1,24E-05	1,62E-05	2,64E-05	2,83E-05	4,98E-05	6,26E-05
200	91,99	88,16	86,71	85,93	84,35	85,26	2,51E-05	3,91E-05	4,62E-05	5,05E-05	6,06E-05	5,46E-05
250	91,69	91,96	90,3	90,3	91,56	89,46	2,6E-05	2,52E-05	3,05E-05	3,05E-05	2,64E-05	3,37E-05
315	93,49	90,82	88,93	88,2	89,87	90,58	2,12E-05	2,88E-05	3,58E-05	3,89E-05	3,21E-05	2,96E-05
400	95,47	93,9	93,82	90,64	90,52	91,1	1,68E-05	2,02E-05	2,04E-05	2,94E-05	2,98E-05	2,79E-05
500	98,91	95,72	95,72	93,65	92,3	93,25	1,13E-05	1,64E-05	1,64E-05	2,08E-05	2,43E-05	2,18E-05
630	99,97	97,32	96	94,77	94,92	93,31	1E-05	1,36E-05	1,58E-05	1,83E-05	1,79E-05	2,16E-05
800	96,97	94,9	93,99	92,27	93,8	92,55	1,42E-05	1,8E-05	2E-05	2,44E-05	2,04E-05	2,36E-05
1000	90,1	87,81	86	86,68	84,77	85,1	3,13E-05	4,07E-05	5,01E-05	4,63E-05	5,77E-05	5,56E-05
1250	91,62	88	86,78	85,16	83,92	83,96	2,62E-05	3,98E-05	4,58E-05	5,52E-05	6,37E-05	6,34E-05
1600	87,66	85,56	84,9	83,78	81,75	81,78	4,14E-05	5,27E-05	5,69E-05	6,47E-05	8,18E-05	8,15E-05
2000	93,79	91	89,14	87,78	87,3	87,6	2,04E-05	2,82E-05	3,49E-05	4,08E-05	4,32E-05	4,17E-05
2500	91,42	88,89	86,51	85,57	84,72	84,41	2,69E-05	3,59E-05	4,73E-05	5,27E-05	5,81E-05	6,02E-05
3150	83,21	81,25	79,8	78,26	77,51	77	6,91E-05	8,66E-05	0,000102	0,000122	0,000133	0,000141
4000	83,39	80,23	79,36	77,85	77,33	76,23	6,77E-05	9,74E-05	0,000108	0,000128	0,000136	0,000154

Table A-1 Continuing

Frequency	Σq	Σr*q	A	r_0	Lp(1)	Lp(2)	Lp(3)	Lp(4)	Lp(5)	Lp(6)
63	0,00	0,00	2214,44	0,83	82,20	75,50	71,76	69,15	67,15	66,30
80	0,00	0,00	7125,90	0,51	83,22	80,26	78,05	76,29	74,83	74,19
100	0,00	0,00	9580,45	0,84	95,63	88,54	84,69	82,03	80,00	79,13
125	0,00	0,00	25726,21	0,70	98,75	94,28	91,34	89,15	87,40	86,65
160	0,00	0,00	18798,87	0,87	103,22	95,12	91,00	88,22	86,12	85,23
200	0,00	0,00	29319,70	0,13	90,59	88,79	87,29	86,02	84,91	84,40
250	0,00	0,00	165511,60	-3,27	91,76	91,36	90,98	90,62	90,27	90,10
315	0,00	0,00	107318,30	-1,85	91,52	90,93	90,38	89,86	89,37	89,14
400	0,00	0,00	68278,93	-0,16	95,40	94,01	92,82	91,77	90,84	90,41
500	0,00	0,00	79108,07	0,02	98,15	96,54	95,18	94,00	92,96	92,49
630	0,00	0,00	89640,27	0,03	99,31	97,68	96,31	95,13	94,09	93,61
800	0,00	0,00	110244,60	-0,73	96,08	95,13	94,28	93,50	92,78	92,45
1000	0,00	0,00	37926,57	-0,30	89,32	88,07	86,98	86,01	85,14	84,74
1250	0,00	0,00	24105,72	0,30	90,76	88,57	86,83	85,37	84,13	83,57
1600	0,00	0,00	21964,95	0,10	87,71	85,98	84,53	83,29	82,21	81,71
2000	0,00	0,00	40492,48	0,07	92,79	91,10	89,68	88,46	87,39	86,91
2500	0,00	0,00	27020,41	0,22	90,77	88,79	87,18	85,82	84,65	84,12
3150	0,00	0,00	12459,07	0,12	83,06	81,27	79,79	78,53	77,42	76,92
4000	0,00	0,00	11469,51	0,16	82,73	80,87	79,34	78,04	76,91	76,39

Table A-1 Continuing

frequency	P1	P2	P3	P4	P5	P6	Lp(1)	Lp(2)	Lp(3)	Lp(4)	Lp(5)	Lp(6)
63	78,82	74,89	75,35	68,86	67,48	65,57	82,20	75,50	71,76	69,15	67,15	66,30
80	83,99	78,22	79,32	76,87	75,45	73,36	83,22	80,26	78,05	76,29	74,83	74,19
100	94,00	88,06	85,66	83,00	79,13	79,25	95,63	88,54	84,69	82,03	80,00	79,13
125	99,00	93,10	92,00	89,98	86,87	86,65	98,75	94,28	91,34	89,15	87,40	86,65
160	98,13	95,81	91,57	90,96	86,06	84,07	103,22	95,12	91,00	88,22	86,12	85,23
200	91,99	88,16	86,71	85,93	84,35	85,26	90,59	88,79	87,29	86,02	84,91	84,40
250	91,69	91,96	90,30	90,30	91,56	89,46	91,76	91,36	90,98	90,62	90,27	90,10
315	93,49	90,82	88,93	88,20	89,87	90,58	91,52	90,93	90,38	89,86	89,37	89,14
400	95,47	93,90	93,82	90,64	90,52	91,10	95,40	94,01	92,82	91,77	90,84	90,41
500	98,91	95,72	95,72	93,65	92,30	93,25	98,15	96,54	95,18	94,00	92,96	92,49
630	99,97	97,32	96,00	94,77	94,92	93,31	99,31	97,68	96,31	95,13	94,09	93,61
800	96,97	94,90	93,99	92,27	93,80	92,55	96,08	95,13	94,28	93,50	92,78	92,45
1000	90,10	87,81	86,00	86,68	84,77	85,10	89,32	88,07	86,98	86,01	85,14	84,74
1250	91,62	88,00	86,78	85,16	83,92	83,96	90,76	88,57	86,83	85,37	84,13	83,57
1600	87,66	85,56	84,90	83,78	81,75	81,78	87,71	85,98	84,53	83,29	82,21	81,71
2000	93,79	91,00	89,14	87,78	87,30	87,60	92,79	91,10	89,68	88,46	87,39	86,91
2500	91,42	88,89	86,51	85,57	84,72	84,41	90,77	88,79	87,18	85,82	84,65	84,12
3150	83,21	81,25	79,80	78,26	77,51	77,00	83,06	81,27	79,79	78,53	77,42	76,92
4000	83,39	80,23	79,36	77,85	77,33	76,23	82,73	80,87	79,34	78,04	76,91	76,39

Table A-1 Continuing

frequency	Δ1	Δ2	Δ3	Δ4	Δ5	Δ6
63	-3,38	-0,61	3,59	-0,29	0,33	-0,73
80	0,77	-2,04	1,27	0,58	0,62	-0,83
100	-1,63	-0,48	0,97	0,97	-0,87	0,12
125	0,25	-1,18	0,66	0,83	-0,53	0,00
160	-5,09	0,69	0,57	2,74	-0,06	-1,16
200	1,40	-0,63	-0,58	-0,09	-0,56	0,86
250	-0,07	0,60	-0,68	-0,32	1,29	-0,64
315	1,97	-0,11	-1,45	-1,66	0,50	1,44
400	0,07	-0,11	1,00	-1,13	-0,32	0,69
500	0,76	-0,82	0,54	-0,35	-0,66	0,76
630	0,66	-0,36	-0,31	-0,36	0,83	-0,30
800	0,89	-0,23	-0,29	-1,23	1,02	0,10
1000	0,78	-0,26	-0,98	0,67	-0,37	0,36
1250	0,86	-0,57	-0,05	-0,21	-0,21	0,39
1600	-0,05	-0,42	0,37	0,49	-0,46	0,07
2000	1,00	-0,10	-0,54	-0,68	-0,09	0,69
2500	0,65	0,10	-0,67	-0,25	0,07	0,29
3150	0,15	-0,02	0,01	-0,27	0,09	0,08
4000	0,66	-0,64	0,02	-0,19	0,42	-0,16

Deviations from the inverse square law (Path 1)

Table A-2 Continuing

frequency	P7	P8	P9	P10	P11	q7	q8	q9	q10	q11
63	66,67	72,61	72,2	78,12	79	0,000464	0,000234	0,000245	0,000124	0,000112
80	71	74,85	76,44	81,82	84,98	0,000282	0,000181	0,000151	8,11E-05	5,64E-05
100	78,96	81,76	84	88,77	89,3	0,000113	8,17E-05	6,31E-05	3,64E-05	3,43E-05
125	87,47	89,62	92,45	93,92	97	4,23E-05	3,3E-05	2,39E-05	2,01E-05	1,41E-05
160	85,93	89,87	93,2	94	97,64	5,05E-05	3,21E-05	2,19E-05	2E-05	1,31E-05
200	85,2	85,95	87,85	89,86	93,67	5,5E-05	5,04E-05	4,05E-05	3,21E-05	2,07E-05
250	91,41	92,99	92,25	90,36	91,99	2,69E-05	2,24E-05	2,44E-05	3,03E-05	2,51E-05
315	90,92	89	89,27	91,48	95,11	2,84E-05	3,55E-05	3,44E-05	2,67E-05	1,76E-05
400	89,7	91,4	93,38	94,76	95,19	3,27E-05	2,69E-05	2,14E-05	1,83E-05	1,74E-05
500	91,56	91,2	94,7	95,11	97,11	2,64E-05	2,75E-05	1,84E-05	1,76E-05	1,39E-05
630	93,55	95,64	96,24	96,93	99,22	2,1E-05	1,65E-05	1,54E-05	1,42E-05	1,09E-05
800	91,3	92,64	93,13	95,43	97,87	2,72E-05	2,33E-05	2,21E-05	1,69E-05	1,28E-05
1000	84,81	86,74	87	87,38	90,78	5,75E-05	4,6E-05	4,47E-05	4,28E-05	2,89E-05
1250	86,6	83,63	86,3	88,36	90,05	4,68E-05	6,58E-05	4,84E-05	3,82E-05	3,14E-05
1600	83,9	83,64	83,83	84,57	84,67	6,38E-05	6,58E-05	6,43E-05	5,91E-05	5,84E-05
2000	85,82	87,26	88,39	91,46	94,14	5,12E-05	4,34E-05	3,81E-05	2,67E-05	1,96E-05
2500	84,83	87,43	88,78	90,96	92,77	5,73E-05	4,25E-05	3,64E-05	2,83E-05	2,3E-05
3150	76,28	77,69	78,57	80,72	82,98	0,000153	0,00013	0,000118	9,2E-05	7,1E-05
4000	76,04	79	78,91	81,24	82,5	0,000158	0,000112	0,000113	8,67E-05	7,5E-05

Table A-2 Continuing

frequency	Σq	$\Sigma r*q$	A	r_0	Lp(7)	Lp(8)	Lp(9)	Lp(10)	Lp(11)
63	0,00	0,00	2458,37	0,82	67,99	69,97	72,54	76,21	82,70
80	0,00	0,00	3631,22	0,85	71,69	73,75	76,47	80,44	87,95
100	0,00	0,00	9895,57	0,75	79,49	81,33	83,66	86,85	91,97
125	0,00	0,00	28868,33	0,63	87,84	89,47	91,47	94,08	97,83
160	0,00	0,00	23001,32	0,77	86,95	88,82	91,21	94,51	99,89
200	0,00	0,00	23060,31	0,48	84,87	86,30	88,01	90,15	93,00
250	0,00	0,00	-449160,00	13,01	92,06	91,91	91,75	91,61	91,46
315	0,00	0,00	65393,76	-0,46	89,21	90,02	90,90	91,89	93,00
400	0,00	0,00	50882,26	0,21	90,11	91,28	92,63	94,23	96,20
500	0,00	0,00	57247,18	0,21	91,13	92,30	93,65	95,25	97,21
630	0,00	0,00	89174,35	0,01	93,93	94,96	96,12	97,47	99,06
800	0,00	0,00	56646,73	0,24	91,21	92,40	93,78	95,43	97,46
1000	0,00	0,00	33106,37	-0,06	85,03	86,02	87,14	88,42	89,93
1250	0,00	0,00	34298,48	-0,18	84,76	85,69	86,72	87,89	89,25
1600	0,00	0,00	114240,40	-5,72	83,64	83,87	84,11	84,36	84,61
2000	0,00	0,00	25097,19	0,50	85,73	87,18	88,92	91,11	94,04
2500	0,00	0,00	24121,89	0,50	85,34	86,78	88,52	90,69	93,59
3150	0,00	0,00	9831,41	0,29	76,27	77,50	78,94	80,67	82,82
4000	0,00	0,00	10468,51	0,26	76,64	77,85	79,25	80,92	83,00

Table A-2 Continuing

frequency	P7	P8	P9	P10	P11	Lp(7)	Lp(8)	Lp(9)	Lp(10)	Lp(11)
63	66,67	72,61	72,2	78,12	79	67,987	69,96926	72,54197	76,21357	82,69967
80	71	74,85	76,44	81,82	84,98	71,68943	73,75414	76,46794	80,43648	87,9542
100	78,96	81,76	84	88,77	89,3	79,48905	81,32541	83,65705	86,85397	91,96695
125	87,47	89,62	92,45	93,92	97	87,84036	89,46775	91,472	94,082	97,83063
160	85,93	89,87	93,2	94	97,64	86,954	88,82334	91,20854	94,50757	99,89217
200	85,2	85,95	87,85	89,86	93,67	84,86858	86,29976	88,01421	90,15241	92,99511
250	91,41	92,99	92,25	90,36	91,99	92,05946	91,90579	91,7548	91,60639	91,46047
315	90,92	89	89,27	91,48	95,11	89,21181	90,01501	90,90012	91,88577	92,99776
400	89,7	91,4	93,38	94,76	95,19	90,11336	91,28244	92,63372	94,23465	96,19887
500	91,56	91,2	94,7	95,11	97,11	91,13055	92,29869	93,6487	95,24786	97,20941
630	93,55	95,64	96,24	96,93	99,22	93,93112	94,95815	96,12311	97,46887	99,06207
800	91,3	92,64	93,13	95,43	97,87	91,20541	92,39777	93,78026	95,42523	97,45626
1000	84,81	86,74	87	87,38	90,78	85,0286	86,01919	87,13747	88,42129	89,92835
1250	86,6	83,63	86,3	88,36	90,05	84,76206	85,68582	86,71965	87,89337	89,25085
1600	83,9	83,64	83,83	84,57	84,67	83,63702	83,87129	84,11206	84,35969	84,61459
2000	85,82	87,26	88,39	91,46	94,14	85,72563	87,17882	88,92498	91,11282	94,04431
2500	84,83	87,43	88,78	90,96	92,77	85,33727	86,78247	88,51707	90,68677	93,58565
3150	76,28	77,69	78,57	80,72	82,98	76,26927	77,50285	78,94108	80,66567	82,81969
4000	76,04	79	78,91	81,24	82,5	76,64104	77,84843	79,25117	80,92491	83,00007

Table A-2 Continuing

frequency	Δ7	Δ8	Δ9	Δ10	Δ11
63	-1,32	2,64	-0,34	1,91	-3,70
80	-0,69	1,10	-0,03	1,38	-2,97
100	-0,53	0,43	0,34	1,92	-2,67
125	-0,37	0,15	0,98	-0,16	-0,83
160	-1,02	1,05	1,99	-0,51	-2,25
200	0,33	-0,35	-0,16	-0,29	0,67
250	-0,65	1,08	0,50	-1,25	0,53
315	1,71	-1,02	-1,63	-0,41	2,11
400	-0,41	0,12	0,75	0,53	-1,01
500	0,43	-1,10	1,05	-0,14	-0,10
630	-0,38	0,68	0,12	-0,54	0,16
800	0,09	0,24	-0,65	0,00	0,41
1000	-0,22	0,72	-0,14	-1,04	0,85
1250	1,84	-2,06	-0,42	0,47	0,80
1600	0,26	-0,23	-0,28	0,21	0,06
2000	0,09	0,08	-0,53	0,35	0,10
2500	-0,51	0,65	0,26	0,27	-0,82
3150	0,01	0,19	-0,37	0,05	0,16
4000	-0,60	1,15	-0,34	0,32	-0,50

Table A-2 Continuing

frequency	BΔ1	BΔ2	BΔ3	BΔ4	BΔ5	BΔ6	BΔ7	BΔ8	BΔ9	BΔ10	BΔ11
63	24,72	20,79	21,25	14,76	13,38	11,47	12,57	18,51	18,1	24,02	24,9
80	22,48	16,71	17,81	15,36	13,94	11,85	9,49	13,34	14,93	20,31	23,47
100	42,09	36,15	33,75	31,09	27,22	27,34	27,05	29,85	32,09	36,86	37,39
125	44,77	38,87	37,77	35,75	32,64	32,42	33,24	35,39	38,22	39,69	42,77
160	39,82	37,5	33,26	32,65	27,75	25,76	27,62	31,56	34,89	35,69	39,33
200	41,16	37,33	35,88	35,1	33,52	34,43	34,37	35,12	37,02	39,03	42,84
250	37,32	37,59	35,93	35,93	37,19	35,09	37,04	38,62	37,88	35,99	37,62
315	33,34	30,67	28,78	28,05	29,72	30,43	30,77	28,85	29,12	31,33	34,96
400	43,89	42,32	42,24	39,06	38,94	39,52	38,12	39,82	41,8	43,18	43,61
500	44,41	41,22	41,22	39,15	37,8	38,75	37,06	36,7	40,2	40,61	42,61
630	40,97	38,32	37	35,77	35,92	34,31	34,55	36,64	37,24	37,93	40,22
800	44,6	42,53	41,62	39,9	41,43	40,18	38,93	40,27	40,76	43,06	45,5
1000	35,47	33,18	31,37	32,05	30,14	30,47	30,18	32,11	32,37	32,75	36,15
1250	31,55	27,93	26,71	25,09	23,85	23,89	26,53	23,56	26,23	28,29	29,98
1600	34,17	32,07	31,41	30,29	28,26	28,29	30,41	30,15	30,34	31,08	31,18
2000	39,46	36,67	34,81	33,45	32,97	33,27	31,49	32,93	34,06	37,13	39,81
2500	31,1	28,57	26,19	25,25	24,4	24,09	24,51	27,11	28,46	30,64	32,45
3150	28,11	26,15	24,7	23,16	22,41	21,9	21,18	22,59	23,47	25,62	27,88
4000	28,53	25,37	24,5	22,99	22,47	21,37	21,18	24,14	24,05	26,38	27,64

APPENDIX B

VACUUM CLEANER NOISE DISTRIBUTIONS

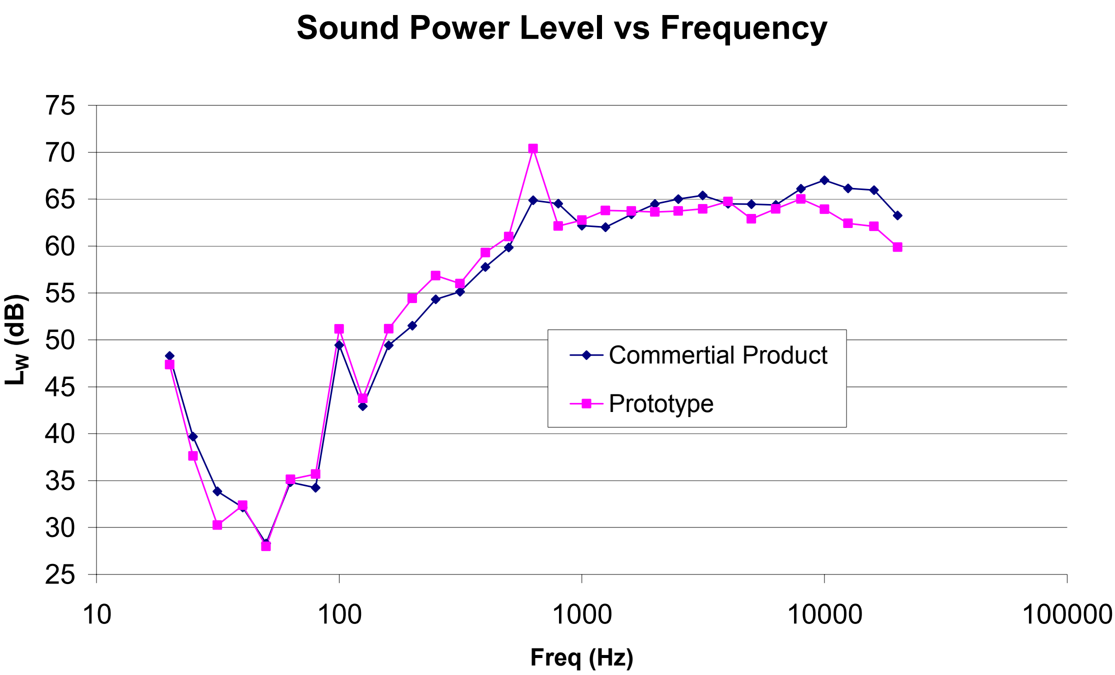

Figure B.1 Prototype and Commercial Product

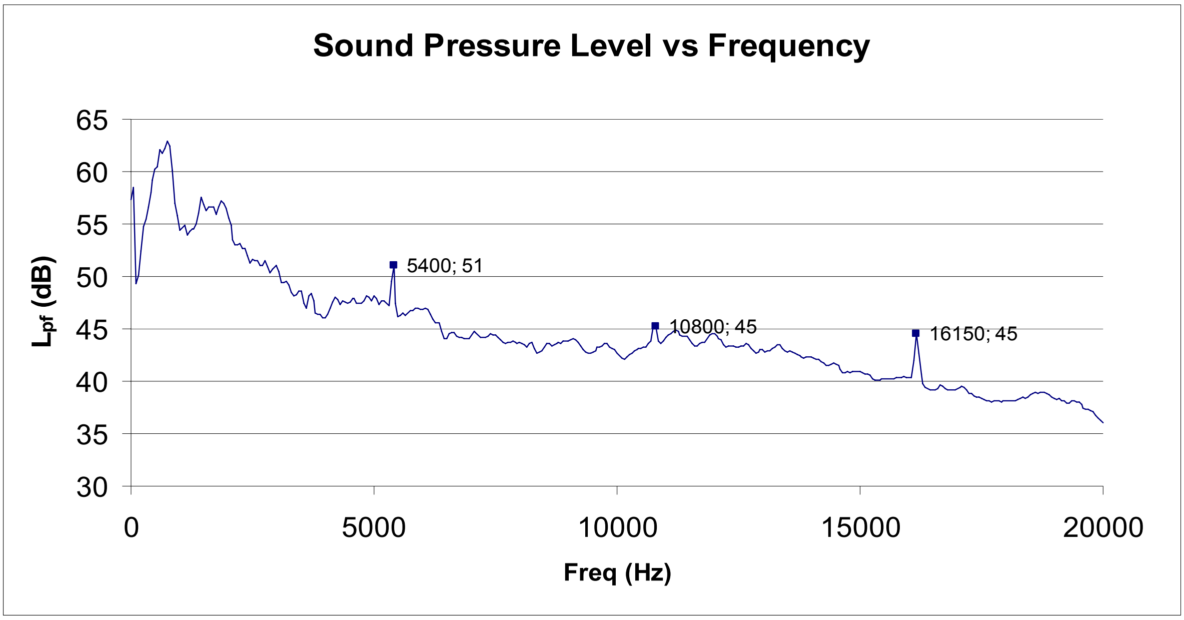

Figure B.2 Spectrum distribution for frequencies increasing 50 Hz of the Commercial Product

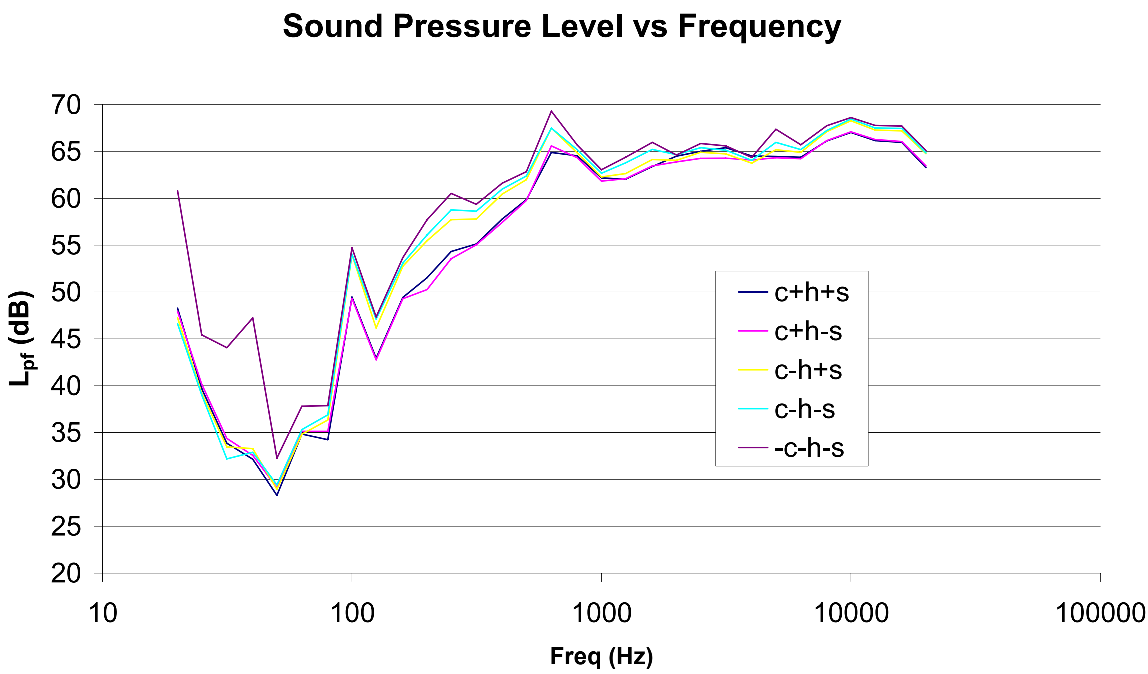

Figure B.3 1/3 octave band sound pressure level distributions

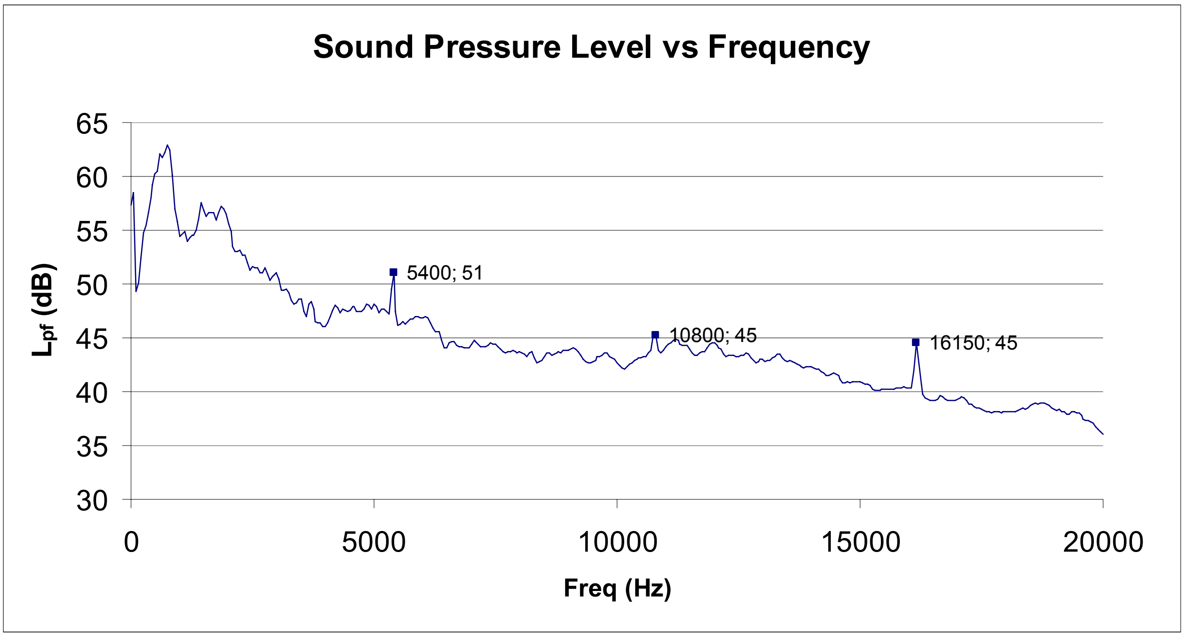

Figure B.4 Spectrum distribution for frequencies increasing 50 Hz of the c+h+s

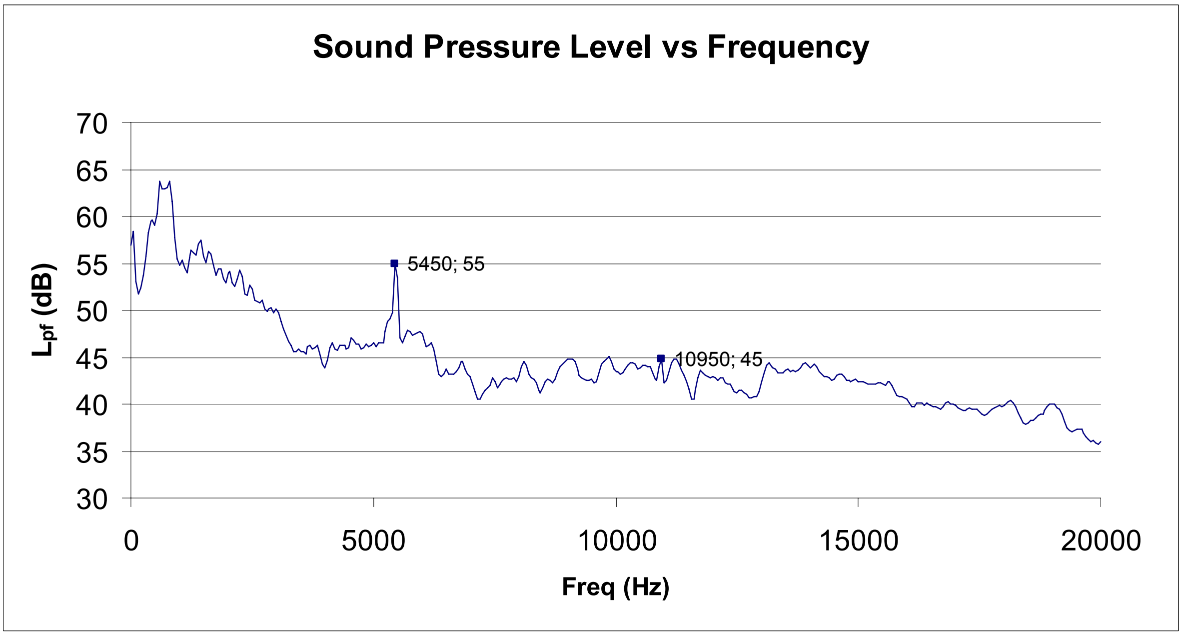

Figure B.5 Spectrum distribution for frequencies increasing 50 Hz of the c+h-s

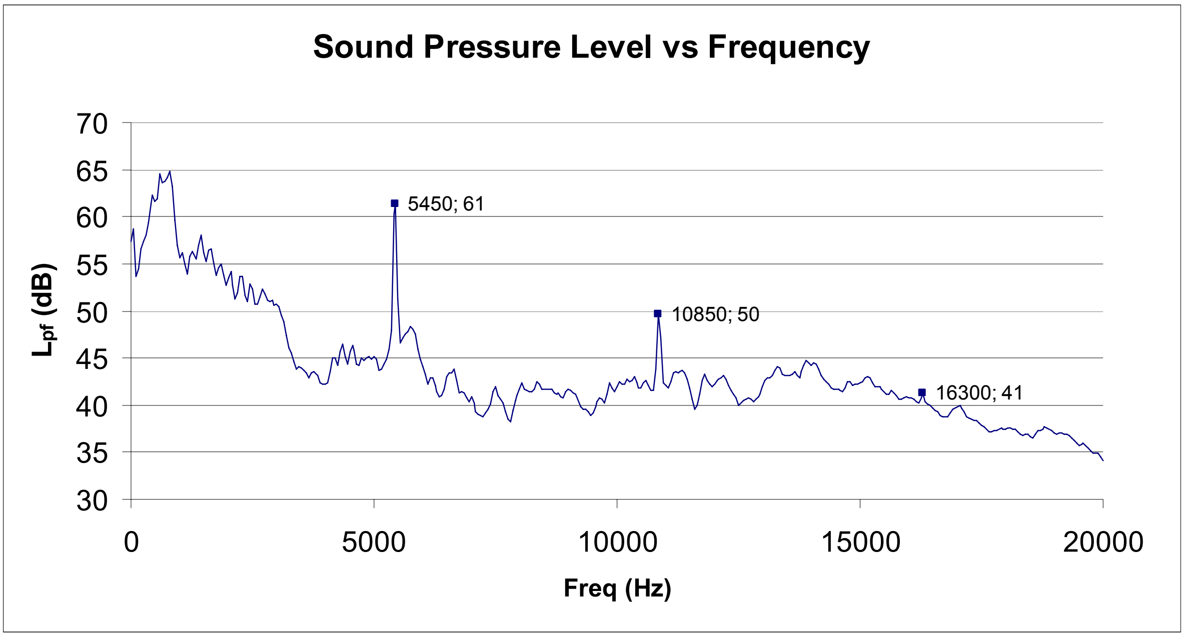

Figure B.6 Spectrum distribution for frequencies increasing 50 Hz of the c-h+s

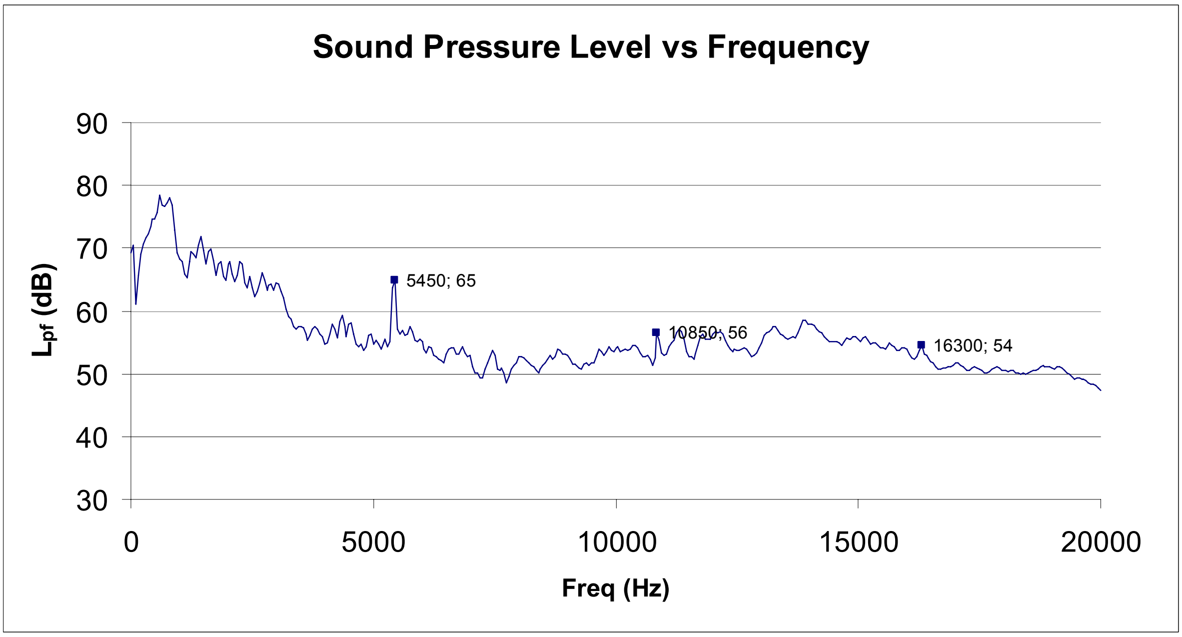

Figure B.7 Spectrum distribution for frequencies increasing 50 Hz of the c-h-s

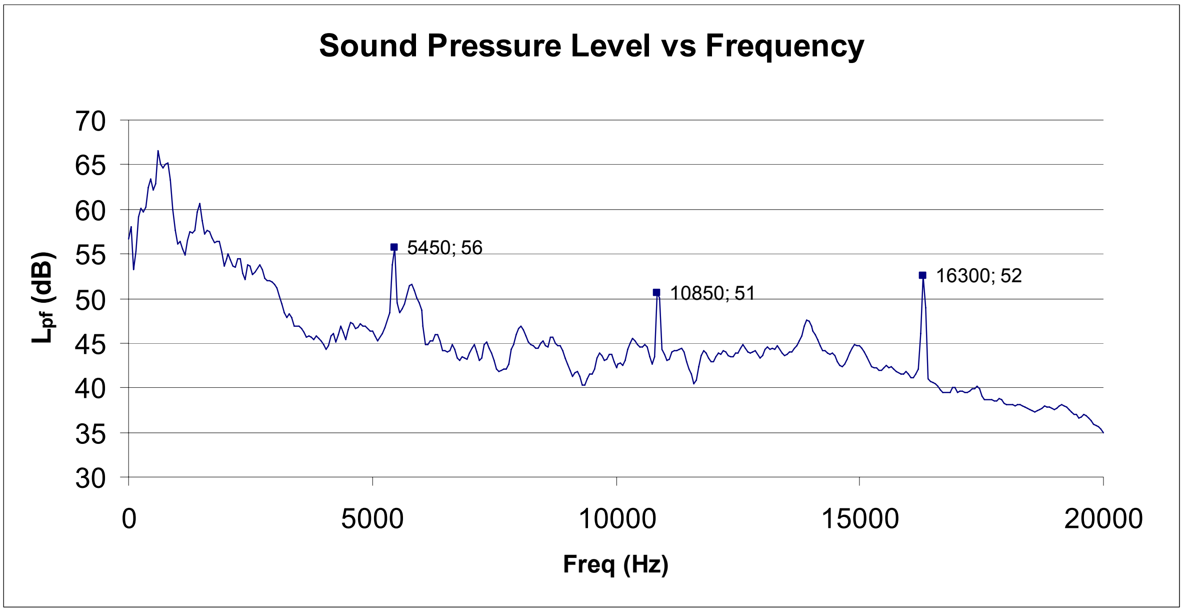

Figure B.8 Spectrum distribution for frequencies increasing 50 Hz of the –c-h-s

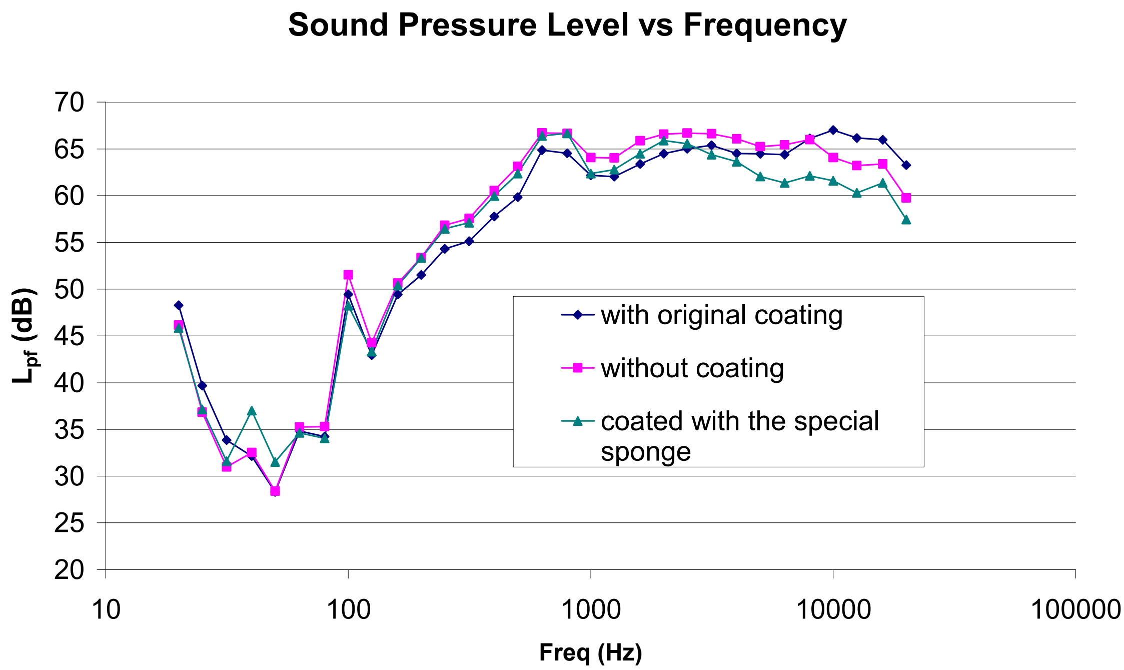

Figure B.9 1/3 octave band sound pressure level distributions

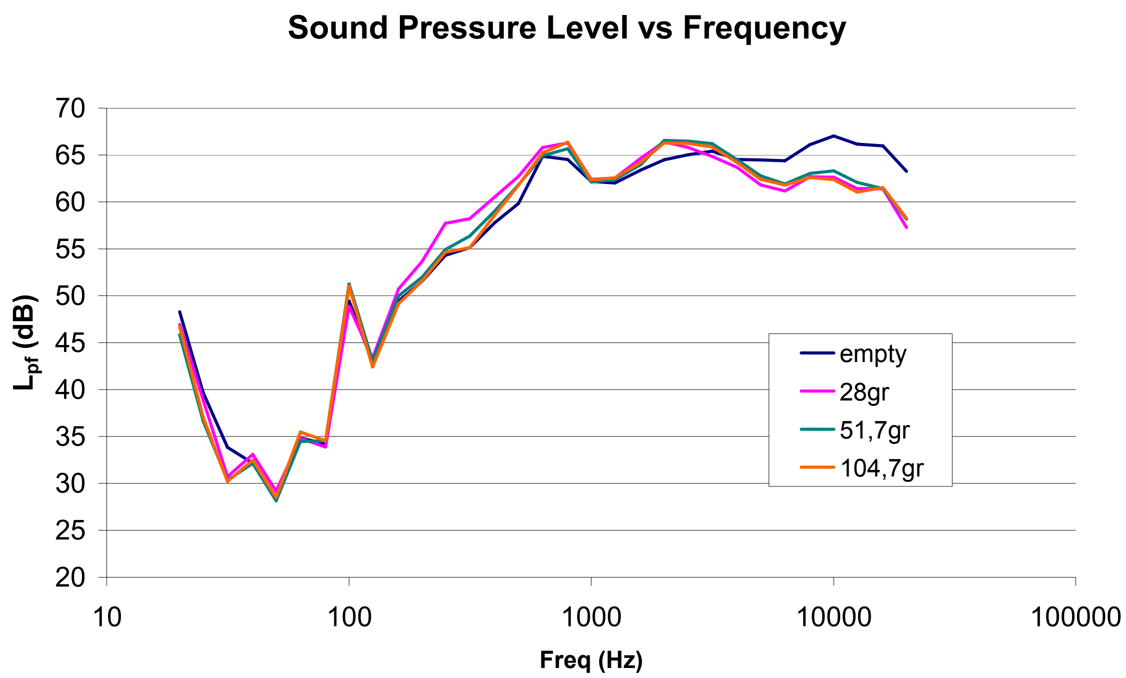

Figure B.10 1/3 octave band sound pressure level distributions

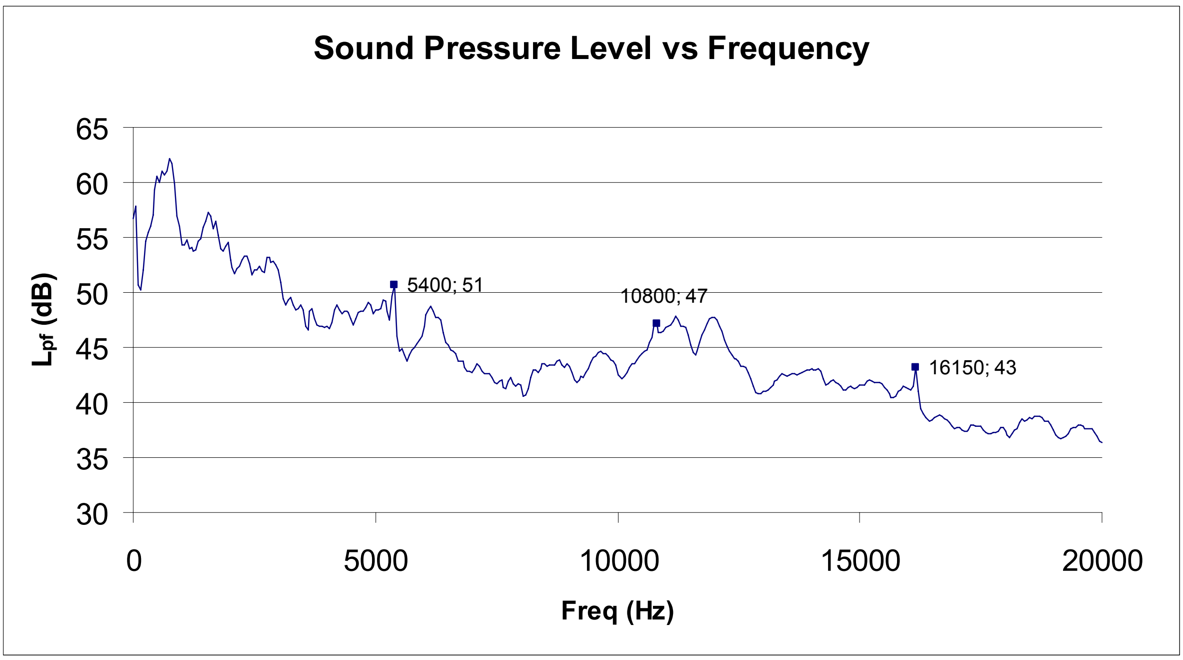

Figure B.11 Spectrum distribution for frequencies increasing 50 Hz of the full case

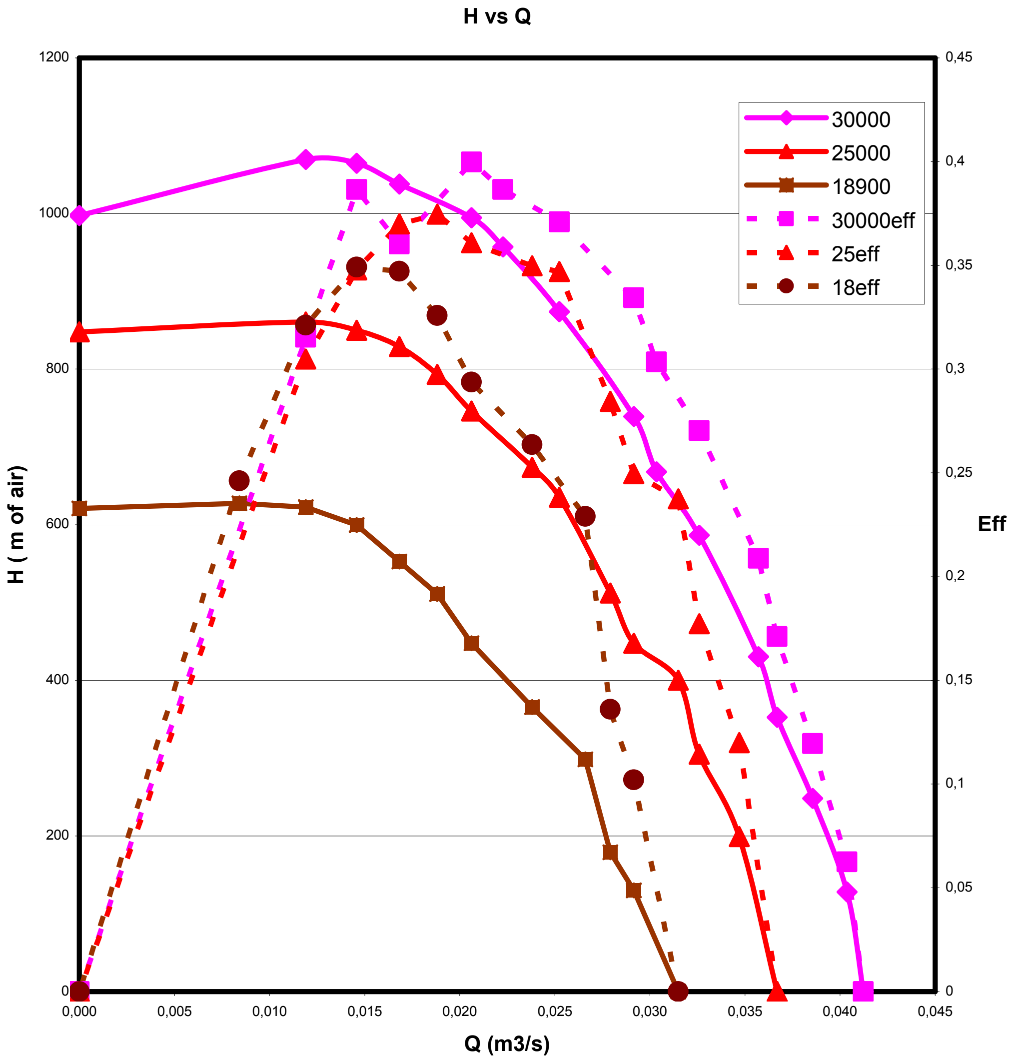

Figure B.12 Graphics of Operating Points for 30000, 25000 and 18900rpm;

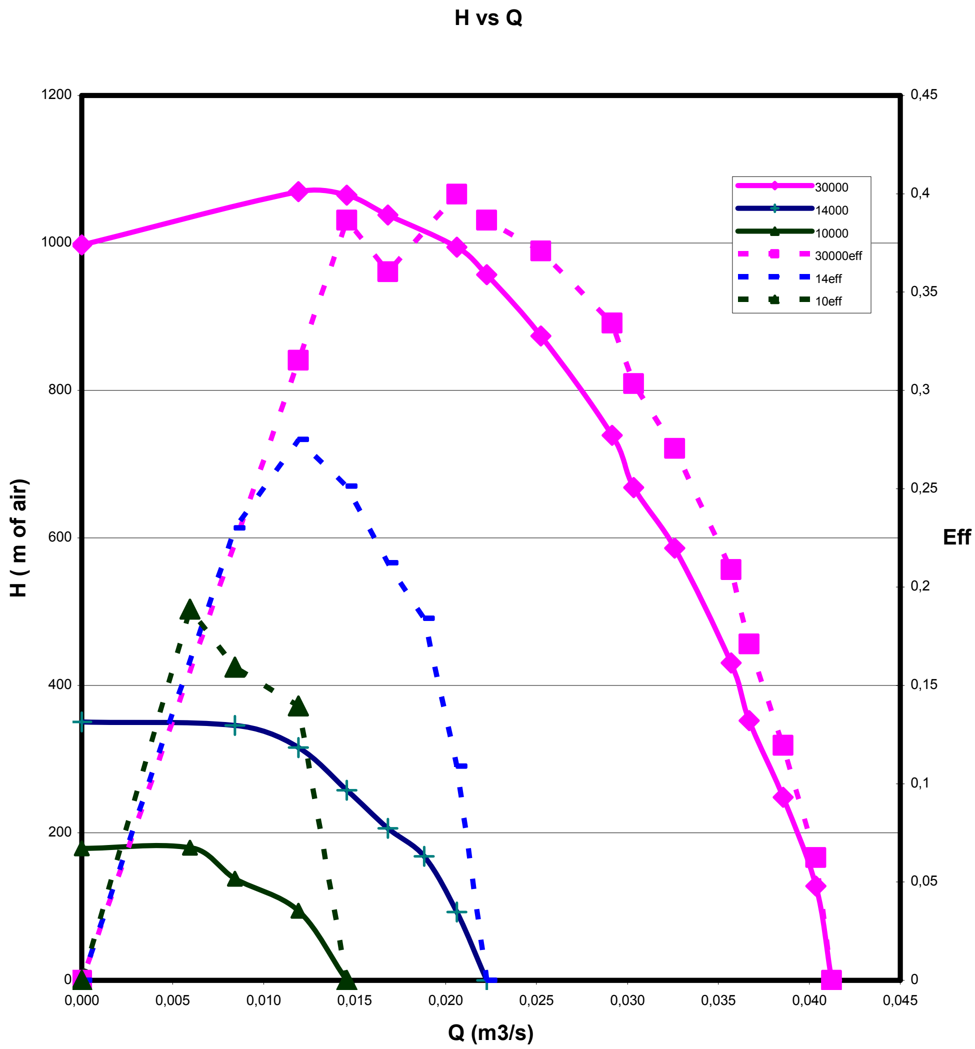

Figure B.14 Graphics of Operating Points for 30000, 14000 and 10000rpm;

APPENDIX C

FLUID MECHANICS OF VACUUM CLEANER

EXPERIMENT SETUPS

C: Detail Drawings of Set-up in Scale

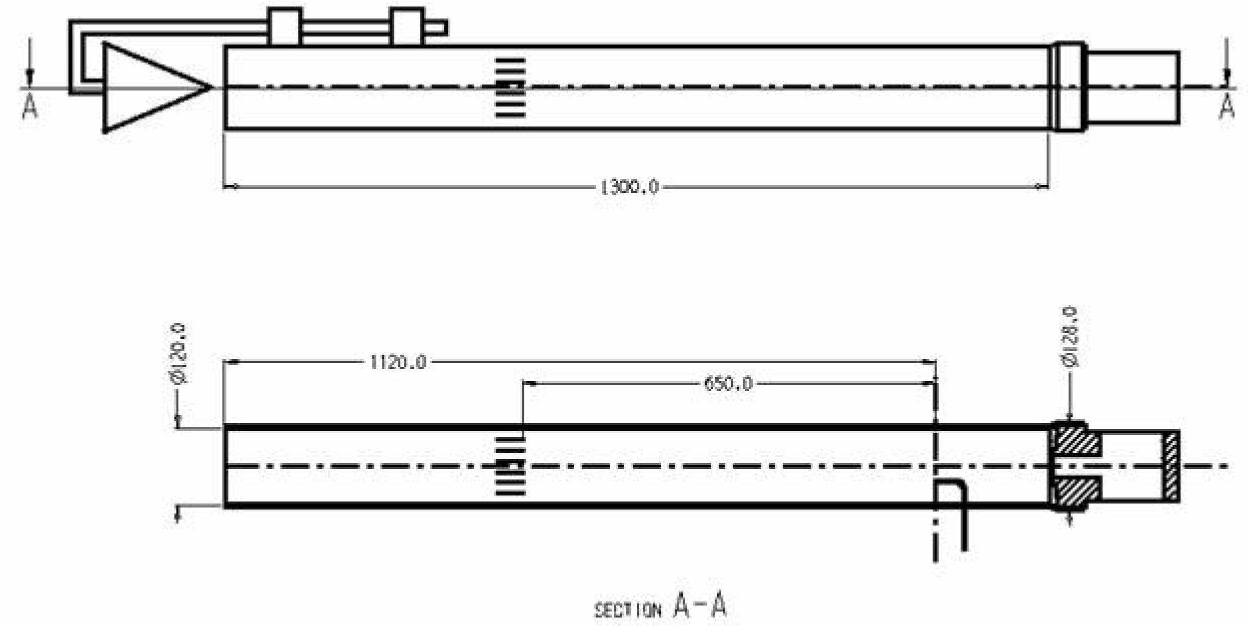

Figure C.1 The Fan Characteristics Set-Up and the Straightener

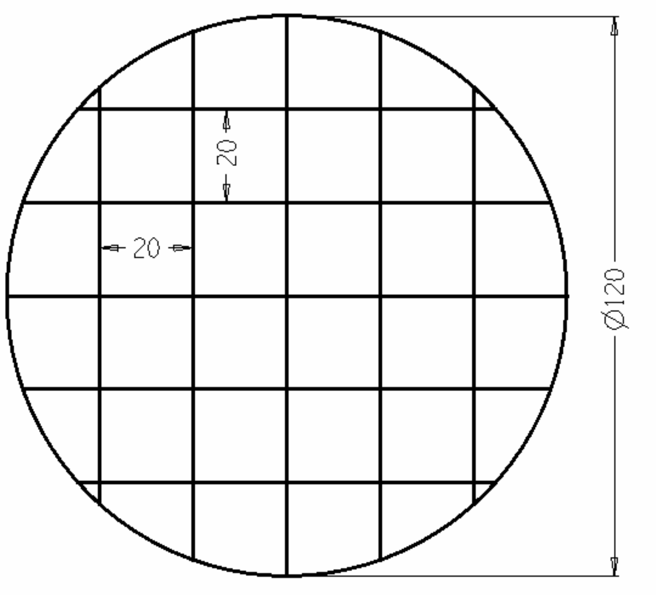

Figure C.2 The Pitot Tubes Locations

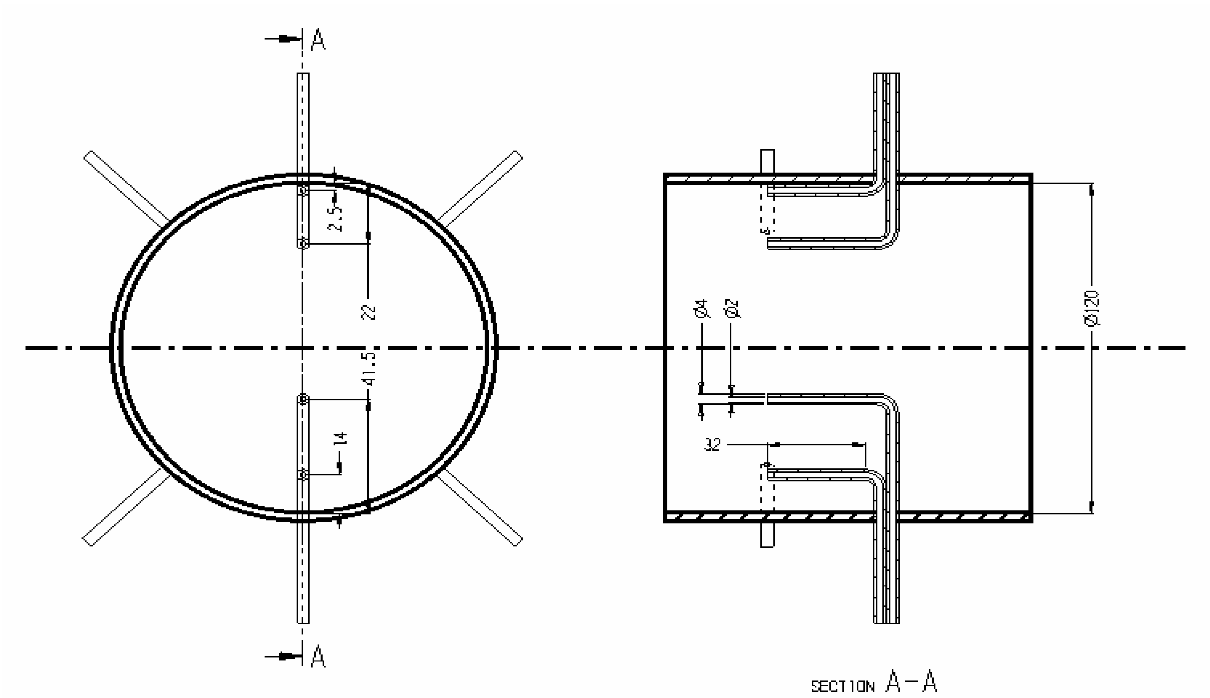

Figure C.4 Microphone Positions

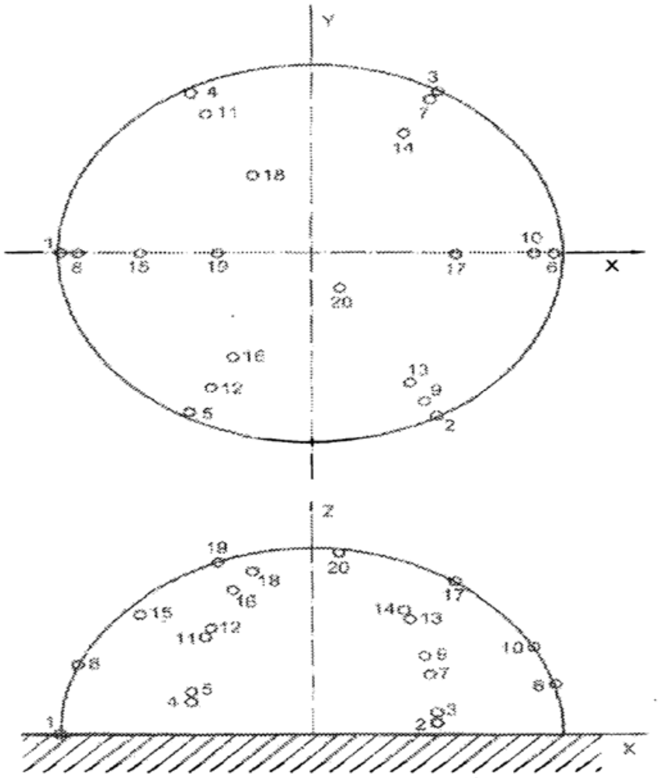

Figure C.5 Microphone Positions in the Experiment

APPENDIX D

COMPONENTS OF VACUUM CLEANER

The parts of a vacuum cleaner and their duty with pictures are in the order of air flow is;

Figure D.1. Cleaning head of the Prototype Vacuum Cleaner

Figure D.2. Hose and Tube

Figure D.3. Vacuum Tank and Dust Bag

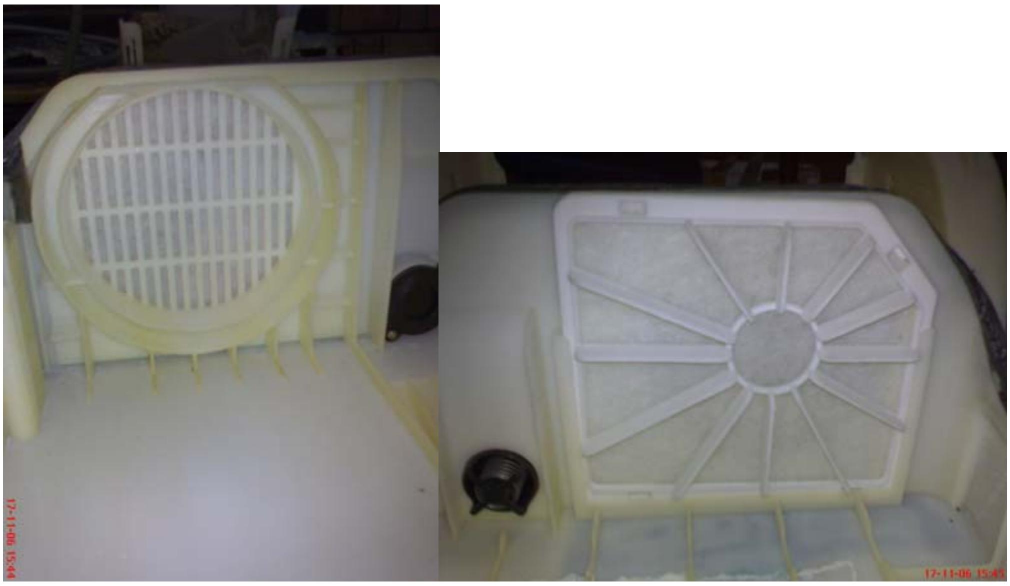

Figure D.4. Motor filter (a) from motor side (b) from dust bag side

Figure D.5. Electric motor with Damping Material

Figure D.6. Fan

Figure D.7. HEPA filter of Prototype (b) HEPA filter of the commercial product

Figure D.8 The Casing of the Vacuum Cleaner