University of Southern Queensland

Faculty of Engineering and Surveying

Reduction of Noise Transmission in The Adephi, Singapore

A dissertation submitted by

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In fulfillment of the requirement of

Course ENG4111 and 4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted : October, 2004

Abstract

Despite awareness of problems caused by noise, it is true to say that the average building & workplace almost everywhere is much noisier than it should be. In the past, noise has too often been accepted as a necessary evil. Today this need not be and building owners and industrialists are being pressed by tenants, governments and workers alike to ensure that a reasonable working environment is maintained. In most cases, acoustic technology and noise control hardware is sufficiently advanced to permit the efficient and econmic solution of the most daunting noise problem.

It is the aim of this research project to look into the noise transmission problem encountered in an existing plant room, before developing solution to achieve the requirement.

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ENG4111 & ENG4112 *Research Project*

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I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Oliver Goh Yaw Siong
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Signature

Date

Acknowledgement

This report was prepared in response to the building maintenance request to look into ways of reducing the noise transmission from an existing lourve in the Adephi building. I would like to take this opportunity to express my appreciation to the following person and company for the valuable helps and advise given to thank A/Prof. Fok Sai Cheong for his guidance and support. I would also like to thank Resma Properties Pte Ltd, a subsidiary of CapitaLand Commercial Properties Limited, for sponsoring this project.

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Chapter 1 Introduction

1.1 Background

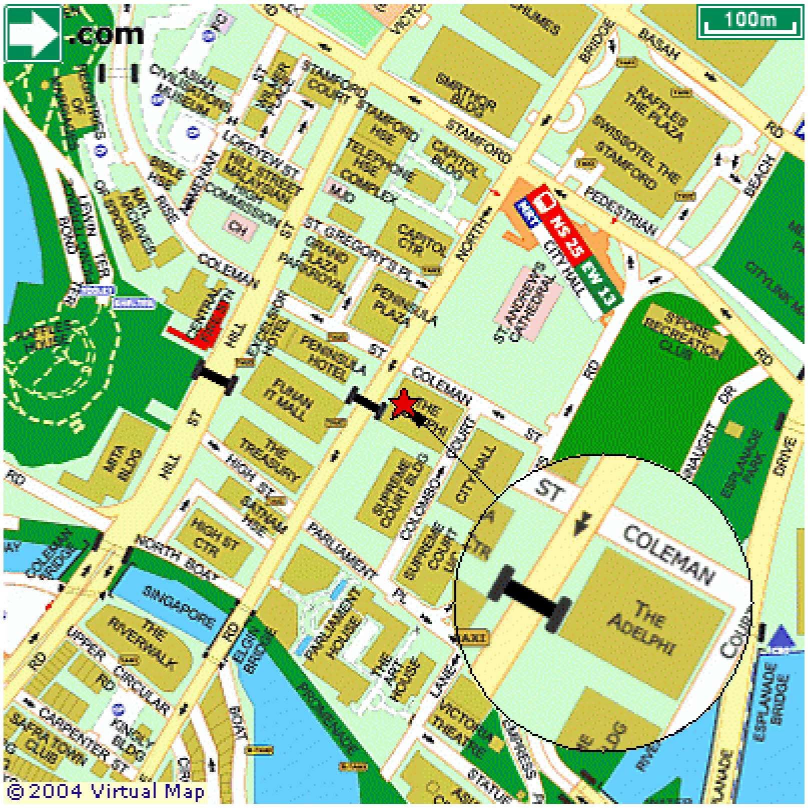
The Adephi is a 10-storey office cum retail development with 5-basement levels prominently located at the junction of Coleman Street and North Bridge Road in Singapore. Commanding prominent road frontages along two busy thoroughfares, it is only a short walk from the City Hall MRT station and within the Civic district where important judiciary and executive offices of the government are located.

Construction work for the building commenced in 1988 and was completed in 1991. It was originally intended to be used as a hotel but was later converted to an office cum retail development. This was due to its prominent location, which is near to the Singapore Supreme Court and the Singapore Parliament House as shown in Figure 1.1.

After the 1997 recession and the subsequent 9-11 bombing of the Twin Tower in the U.S.A (the main trading partner of Singapore), Singapore was badly hit by the downturn in its economy. The construction industry, which contributes to about 12% of the employment rate in Singapore, was severely affected. To help the construction industry, the government brought forward a number of the public works, including the constructions and refurbishment of government buildings. The government also introduced incentives for renovation and upgrading work to help boost the constructions in the private sectors.

In 2001, due to the government plan to build the New Singapore Parliament House and New Singapore Supreme Court, which are in close proximity to The Adephi. The building major shareholder, CapitaLand Commercial Limited, decided to conduct an upgrading work to ensure The Adephi would blend in well with these new buildings. The upgrading would also provide a better environment for its shoppers and tenants.

Figure 1.1 : Location of The Adephi



To facilitate the process, an Upgrading Management team for the upgrading work was formed. This includes the owner representative, the Project Management team, Interior Designer, M&E Consultant, Building Management team and other various trade specialists.

During the process of the work, the Upgrading Management team has received various feedbacks from tenants and shoppers. One of the feedback items was the disturbing noise that was emitting from an existing lourve along the side entrance to the building. This item was considered important, as over the years the main crowds have been coming in through the side entrance instead of the main entrance. This is mainly due to the erection of new shopping centres, malls, hotels and increase in transport facilities along the road facing the side entrance of the building. In fact, the upgrading management agrees that the continuous irritating noise emitting from the lourve might undermine the whole purpose of the upgrading work.

In order to eliminate this problem and to prevent future enquiry from the local authority, when the new government building are built. The management has decided to look into the complaints and to put in place the possible solution that could help to reduce the noise transmission from the existing lourve, located at the building side entrance.

1.2 Objectives

This research project aims to look into the source of the noise so as to reduce the noise transmission from the lourve. It would involve reviewing of the noise control design criteria for building, which includes ambient noise level, intrusive noise levels and reverberation times, as well as noise and vibration control for the mechanical and electrical installation in the existing plant room. The review will also focus on the noise control design criteria adopted in both Singapore and overseas. These noise control design criteria will set as an acoustic basis for the research project, in which the following objectives shall be achieved:

- To study the cause and the source of the noise
- To develop a feasible solution to minimize the disturbances caused by the noise to the tenants, shoppers and anyone using the walkway along the building side entrance, without modification to the existing equipment setup.
- To develop a feasible solution that meets the authority regulation standard requirement adopted in Singapore and oversea.
- To develop a proposal to meet the above requirement with a budget that does not exceed S$10,000.00. This tight constraint in budget is due to the recent investment in the major upgrading work to blend-in with the surrounding new building.

1.3 Design Assumptions

Due to a number of site and access constraints faced during the research, the investigation of this research project was based on the following assumptions:

- All penetrations or openings irregardless of slab, wall or door opening shall be properly sealed to isolate the plant room as an enclosures
- All seals at doors, etc. shall be able to provide equivalent performance with comparison to enclosure panels. This is to prevent noise leakage through any door gap.
- Ventilation ducting used in this research project shall be able to allow both natural or forced air extractions which is treated with silencer that has an insertion loss equivalent to enclosure panels.

1.4 Overview of the Dissertation

This report contains five major sections: An introduction and background information on the existing system, literature review on various method of noise control, overview of existing mechanical ventilation system setup, comparisons and analysis of the existing setup and proposed solution. Finally we would look into the conclusions and proposed further work.

Chapter One will look into the introduction and background information of the research project and also the reason why this project was created. Chapter Two contains the literatures review on noise control. It covers the definition of sound and noise, the standard units used for sound, the noise rating (NR), the weighting network, relationship between sound power and sound pressure, noise criteria and the various methods of noise control. Chapter three gives an overview of present mechanical ventilation system. This chapter will look into the existing system configuration, its condition and the possible problem that has caused the emission of the irritating noise from the lourve at the side of the entrance. Chapter Four gives the results of the compari47with the proposed new system configuration. The final Chapter will look into the overall achievement of the research project objectives and the recommendation of future work to be undertaken to further improve the research of the project.

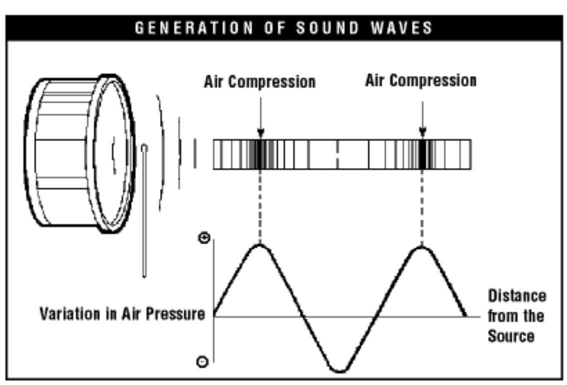
Chapter 2 Literatures Review

2.1 Definition of Sound and Noise

Sound is what we hear. Noise is unwanted sound. The difference between sound and noise depends upon the listener and the circumstances. Rock music can be pleasurable sound to one person and an annoying noise to another. In either case, it can be hazardous to a person's hearing if the sound is loud and if he or she is exposed long and often enough.

Sound is produced by vibrating objects and reaches the listener's ears as waves in the air or other media. When an object vibrates, it causes slight changes in air pressure. These air pressure changes travel as waves through the air and produce sound. To illustrate, imagine striking a drum surface with a stick. The drum surface vibrates back and forth. As it moves forward, it pushes the air in contact with the surface. This creates a positive (higher) pressure by compressing the air. When the surface moves in the opposite direction, it creates a negative (lower) pressure by decompressing the air. Thus, as the drum surface vibrates, it creates alternating regions of higher and lower air pressure. These pressure variations travel through the air as sound waves (Figure 2.1).

Figure 2.1 : Generation of Sound Waves



2.2 The bel and decibel

The unit bel (i.e. 10 dB) is borrowed from the field of telecommunication engineering and is a dimensionless unit expressing the logarithmic ratio of two quantities proportional to power. However, the quantities sound power, intensity and sound pressure vary over a very wide range (as can be seen from Table 2.1): such large numbers are therefore clumsy to handle in calculations or when describing a sound source. The use of a logarithmic scale conveniently helps to overcome this problem.

Table 2.1 : Linear, exponential and decibel scales for sound power

Radiated Sound Power (W)		Sound Power Level (dB)		
Usual Notation	Expressed as powers	Relative to 1W	Relative to 10^{-13} W	Relative to 10^{-12} W
100 000	10^5	50	180	170
10 000	10^4	40	170	160
1 000	10^3	30	160	150
100	10^2	20	150	140
10	10^1	10	140	130
1	1	0	130	120
0.1	10^{-1}	-10	120	110
0.01	10^{-2}	-20	110	100
0.001	10^{-3}	-30	100	90
0.0001	10^{-4}	-40	90	80
0.00001	10^{-5}	-50	80	70
0.000001	10^{-6}	-60	70	60

As an example a sound power of 100 000 W may be written more compactly as 10^5 W. If we take the logarithmic ratio which defines the bel, this becomes even more compact, i.e. 5 bels or 50 dB.

$$\text{Therefore sound power level in dB} = 10 \log_{10} \frac{\text{sound power}}{\text{Reference power}}$$

The reference power in the above example was taken as 1 W. The reference generally accepted and arrived at by international agreement is 10^{-12} W, although a reference of 10^{-13} W has been very widely used in the USA. The level in decibels relative to the former reference can be arrived at by subtracting 10 dB from the level relative to the latter reference.

Since sound power varies as square of sound pressure,

$$\text{Pressure level in dB} = 10 \log_{10} \frac{(\text{Sound pressure level})^2}{(\text{Reference pressure})}, \text{ or}$$

$$\text{SPL (dB)} = 20 \log_{10} \frac{(\quad p \quad)}{(\text{reference pressure})}$$

The acceptable reference used for pressure is 0.0002 microbars or dynes per square centimeter; this is the threshold of hearing for average subjects. It must be borne in mind that a decibel figure without any reference value is meaningless. In this report, the notation ‘dB’ is used for both sound power and pressure level. Where not stated explicitly, the reference implied is 10^{-12} W for sound power level and 0.0002 dyne/cm^2 for sound pressure level.

Sound power and sound pressure levels are quoted in decibels, dBW and dB respectively. The decibel is derived from values associated with power (watts) and pressure as discussed earlier. Being logarithmic scales, they cannot be added together or subtracted whenever one pleases. They must be combined according to certain simple rules given. However, if silencing (attenuation) is being dealt with, dB values may be directly subtracted because in this context they represent the ratio of power or pressure reduction through the silencer or lined duct. Sound power dBW scale is based on a reference level of 10^{-12} watts, the sound pressure dB scale is based on 2 x 10^{-15} Newtons/meter^{2}.

2.3 Noise Rating (NR) Curves

The frequency range to which the ear can respond is approximately from 20 to 20,000 cycles per second. In practical noise analysis, it is not necessary to cover more than from 45 to 11,000 cycles per second. This range is divided into eight groups of frequencies called octave bands. In each band the upper frequency is double the lower. The mid frequency is usually quoted to identify the octave band, for example, the 355 to 710 c/s octave band is labeled the 500 c/s octave band.

In 1971, International Organization for Standards published an International Standard – ISO 1996 and in 1973, the Australian Standard 1469 was published. Both of these Standards contained in internationally agreed set of empirical curves relating acceptable octave band sound pressure level to centre frequencies of these octave bands. The NR curve is a stringent criterion in that it requires all points to be on or below the selected NR Curve.

These curves are similar in various aspects to the set of Noise Criteria (NC) Curves. For the same numerical value, the NR Curve is approximately 1 to 3 dB lower than NC Curves (more stringent) in the mid to high frequency bands, whilst in the lower frequency bands, the NC Curve is 1 to 2 dB lower.

As the ISO Standards are widely accepted internationally and in Singapore, the NR Curves design criteria would be selected in this reduction of noise transmission research project.

To determine the Noise Rating (NR) rating of a noise:

- Plot measured octave band sound pressure levels against octave band centre frequencies. Table 2.2 shows the sound pressure levels corresponding to the various NR rating.
- The NR rating is determined by the highest curve which would envelop them

Table 2.2 : Sound pressure levels corresponding to various NR levels

Noise Rating	Octave band (Hz) – Sound Pressure Level (dB)							
	63	125	250	500	1000	2000	4000	8000
65	97	78	72	68	65	62	60	59
60	83	74	68	63	60	57	55	54
55	79	70	63	58	55	52	50	49
50	75	65	58	53	50	47	45	43
45	71	61	54	49	45	42	40	38
40	67	57	49	44	40	37	35	33
35	63	52	44	39	35	32	30	28
30	59	48	40	34	30	27	25	23
25	55	44	35	29	25	22	19	18

2.4 Weighting Network

As our measuring of noise usually involve people, the ultimately interest is in the human reaction to sound rather than in sound as a physical phenomenon. Sound pressure level, for instance, cannot be taken at face value as an indication of loudness because the frequency (or pitch) of a sound has quite a bit to do with how loud it sounds. For this and other reasons, it often helps to know something about the frequency of the noise to be measured. This is where weighting networks come in.

They are electronic filtering circuits built into the meter to attenuate certain frequencies. They permit the sound level meter to respond more to some frequencies than to others with prejudice something like that of the human ear.

There are basically three weighting characteristics that are normally used, A, B, and C. The chief difference among them is that very low frequencies are filtered quite severely by the A network, moderately by the B network, and hardly at all by the C network. Therefore, if the measured sound level of a noise is much higher on C weighting than on weighting, much of the noise is probably of low frequency. Readings taken when a network is in used are said to be "sound levels" rather than "sound pressure levels. " The readings taken are designated in decibels in one of the following forms: dB(A); dBa; dBA; dB(B); dBb; DBB; and so on. Tabular notations may also be referred to as L_A, L_B, L_C.[1]

Table 2.3 : Presents the A-weighting filter Characteristics by Octave Bands

dB (A) weighting, Relative Response	
Octave Band Centre Frequency (Hz)	Weighting (dB)
31.5	-39
63	-26
125	-16
250	-9
500	-3
1000	0
2000	+1
4000	+1
8000	-1

[1]. Davis Cornwell, *Introduction to Environmental Engineering*, Third Edition: McGraw-Hill, 1998, p559

The A network is the most widely used weighting network in Singapore and many other countries. This is because it closely corresponds to loudness or the response of the human ear. The A network is used in this project. Table 2.3 gives the A-weighting filter characteristics by Octave Bands.

2.5 Sound Power and Sound Pressure Relationship

Consider a room in which a heater consuming electricity releases energy into the room as heat. The rate of energy release will be power, that is, Btu/h or kWh. The temperature of the room at some point will depend on many factors. The most important factor will be the distance of the thermometer is from the heater and the amount of heat, which is absorbed by and transmitted through the walls, windows, ceiling and floor. The less the heat lost, the higher will be the room temperature. Now consider a room, which is being fed with sound power, say, through a ventilation inlet grille. This power will cause sound pressure to build up in the room. The sound pressure level at any point will be dependent on how far that point is from the grille through which sound power is emerging and on the amount of sound power absorbed, reflected and transmitted through the walls, windows, ceiling and floor. This analogy is given to explain the important relationship between sound power and sound pressure. It will be appreciated that the sound power level of a fan is dependent on the amount of sound power it generates in doing its work. The resulting sound pressure level, to which the ear and the sound level meter respond, is entirely dependent on the acoustic characteristics of the environment in which the sound power is propagating. In open air, clear of all buildings, walls or other surfaces which could reflect the sound pressure waves, the sound pressure level will fall away uniformly as the distance from the fan. In a room or hall the sound pressure waves will reach the occupants by two paths:

1) directly, failing off as the distance from the fan or the ventilation outlet; this is known as the direct sound pressure level (SPL Dir.) ;

2) by multiple reflections from the walls, ceiling, floor, etc, which will depend entirely on the size of the room and its acoustic absorbing or reflecting qualities; this is called the reverberant sound pressure level (SPL Rev.).

The two levels combine to give a resultant effect on the ears or on the sound level meter. The barer and more reflective the walls, ceiling and floor of the room, the higher will be the reverberant sound pressure level. The reverberant sound pressure level may completely swamp the direct sound pressure level. The more heavily furnished the room (for example a room with carpets, curtains, etc.), the more sound is absorbed and the lower the reverberant sound pressure level. In absorbent conditions the direct sound pressure level will usually be more prominent. For these reasons, the sound pressure levels stated by different equipment manufacturers cannot be compared unless it is known that the equipments being compared were tested under identical acoustic conditions. It is possible, however, to compare sound power levels, which are independent of the acoustic conditions, provided that the frequency content levels are specified.

2.6 Noise Criteria

In order to improve working environments, numerous guidelines have been developed in Singapore and overseas countries during the past three decades. These guidelines and recommendations were normally tailored to suit the needs of the country.

The partial recommended ambient noise level criteria table for continuous noise intrusion, are given in the Table 2.4. Each building space has been assigned an ambient noise rating (NR) level.

Table 2.4 Recommended Noise Rating inside Buildings

<table>
<tr><th rowspan="2">Description Area</th><th colspan="2">Criteria for continuous noise intrusion</th><th colspan="2">Criteria for traffic and other intermittent noise dB(A)</th></tr>
<tr><th>NR Curve</th><th>Approx. dB(A)</th><th>L_{90}</th><th>L_{10}</th></tr>
<tr><td colspan="5">Factory areas</td></tr>
<tr><td>Light maintenance shops</td><td>50</td><td>55</td><td colspan="2" rowspan="4">Not considered as background NR level high enough to mask external noises</td></tr>
<tr><td>Area for just acceptable speech and telephone conversation</td><td>55</td><td>60</td></tr>
<tr><td>Areas where speech or telephone conversation not required but where no risk of hearing damage – heavy industrial processing</td><td>60 ~ 75</td><td>65 ~ 80</td></tr>
<tr><td>Office areas, control rooms within factory area</td><td>40</td><td>45</td></tr>
<tr><td colspan="5">General service areas for all buildings</td></tr>
<tr><td>Corridors</td><td>45</td><td>50</td><td></td><td></td></tr>
<tr><td>Toilet, washrooms</td><td>45</td><td>50</td><td>50</td><td>60</td></tr>
<tr><td>Plant rooms</td><td>70</td><td>75</td><td>50</td><td>60</td></tr>
</table>

From Table 2.4, a portion of the recommended noise rating inside a building extracted out from the ASA Draft Standard DR75138 – Code of Practice for Ambient Sound Levels for area of occupancy within building, the NR curve for an area where speech or telephone conversation does not occurred (i.e. no risk of hearing damage) is 60 ~ 75, which is approximately 65 ~ 80 dB(A). Therefore, this research project will base on this noise control design criteria of 65 ~ 80 dB(A), to be the guide for the recommended ambient noise level criteria for continuous noise intrusion.

2.7 Methods of Noise Control

This section provides recommendations for controlling noise by engineering methods. In order to make use of the engineering methods to control noise, a detailed knowledge of the characteristics of the noise must be made.

2.7.1 Principles of Noise Control

The principles of noise control are most easily understood when the noise system is broken into its various elements. All noise systems may be viewed, as a system comprises mainly of three components:

1. The source that generates the noise
2. The noise transmission paths through which this noise must pass through to reach the observer
3. The receiver:
    - Level of noise hear and received by the receiver
    - The acceptable level of noise and the noise design criteria that is required for assessment (Please refer to Table 2.4)
    - The source and transmission characteristics to achieve an acceptable noise level at the point of the receiver

2.7.1.1 The Noise Source

Below are some of the ways, which noise can be reduced from the existing equipment or machinery system. These includes:

- Avoidance or reduction of metal-to-metal impact
- Use of vibration isolating mountings to reduce vibration transmission
- Suppression of vibration of the external surfaces of the equipment and machinery system e.g. by the selection of suitable material, stiffness and damping and by careful dynamic balancing

- Reduction of mechanical forces (such as the reduction of speed or power as sound level usually increases with increasing speed; reduction of relative acceleration between machine components)
- Reduction of aerodynamic forces (such as minimize flow velocity; reduction of discharge velocities, static pressure losses, eliminate shocks; avoid cavitation)

2.7.1.2 The Noise Transmission Paths

The control of noise along the path of transmission can be achieved by:

- Increasing the distance between the source and receiver. (This will not be effective in reverberant indoor conditions)
- Enclosing the noisy equipment with acoustic material, as complete enclosure of the noisy equipment is the most effective method of controlling airborne noise.
- Erecting partition wall / barrier to isolate or separate the noise source from the receivers. Separation should be done as complete as possible and the number of openings kept to a minimum.
- Placing an acoustic shield or barrier between the noise source and the receiver, to prevent the direct transmission of noise
- Applying acoustic absorbing material onto the walls and ceiling of the room to reduce reverberant noise.
- Make use of 'silencers' or attenuation in the intake and exhaust systems, which are associated with fluid flow processes.
- Applying active noise control for low frequency or broad band noise. [2]

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[2]. Standard Association of Australian, *Australian Standard Hearing Conservation,* p14, 1983

2.7.1.3 The receiver or observer

Noise control or reduction of noise control can be achieved at the receiver by the following methods:

- Placing the exposed persons in a sound insulated booth or control room to minimize the exposure of the person to the noise
- Rotating the receivers so as to reduce their exposure time to the noise
- Using of hearing protectors against the noise (The function of hearing protection device is to reduce the amount of noise reaching the inner ear of the wearer. This is achieved by completely covering the ear with an earmuff or helmet. It can also be achieved, by covering the entrance of the ear with a cap or by completely covering the ear canal with an earplug)

2.7.2 Noise Control Systems

There are six (06) types of noise control systems that may be used for the consideration of solving any noise problem:

1. Sound barriers
2. Sound absorbers
3. Vibration damping
4. Vibration isolation
5. Mufflers and silencers
6. Equipment / machine redesign, process modification, or noise source elimination or at least noise suppression

2.7.2.1 Sound barriers

The most fundamental approach to noise reduction is to interpose a barrier between the sound and the receiver. The barrier may take the form of:

1. an enclosure of the noise source
2. an enclosure of the receiver, or
3. a barrier between the noise source and the receiver

The noise reduction or difference between the sound levels on either side of the barrier will vary, depending upon the type of construction employed; however, the physical acoustical properties of the barrier is the key element in the calculation of the noise reduction for all the three of these designs.

The basic characteristic used to describe the acoustical performance of a barrier is the transmission loss. The transmission loss, abbreviated TL, is defined as 10 times the logarithm (base 10) of the incident sound energy on a barrier to the sound transmitted through it.

The Sound Transmission Class, STC, is a single numerical average of a barrier control of noise hazard that can be accomplished by three approaches; engineering methods, adminstrative measures and personal protection. The most desirable course of action for noise control is to apply engineering noise control measures to reduce the noise exposure level. Engineering controls are physical means that reduce the sound levels either at the source, along the path or in the hearing zone of the receiver. The most effective method of control is the method that is able to elimination or partial suppression of the noise at its source.

2.7.2.2 Sound Absorption

The amount of absorption provided by a material is characterized by its absorption coefficient. The sound absorption coefficient, α, represents the ratio of the sound absorbed to the sound incident on any material and varies from 0~1. By definition:

$$\text{Absorption Coefficient, } \alpha = (\text{Absorbed Energy} / \text{Incident Energy})$$

A α value equals to 1 indicates that all of the incident acoustic energy is absorbed, while a α value equal to 0 means that all of the incident acoustic energy is reflected. Sound absorption values vary with frequency and are dependent upon the material thickness.

Whenever equipment or machinery is operated within enclosed spaces, sound levels will increase to some extent due to reverberation. When this reverberant sound level increase becomes significant, it is appropriate to install sound absorptive materials on the ceiling to decrease the offending noise. The most convenient method of employing sound absorption is the installation of acoustical baffles.

The most important piece of knowledge required about acoustic materials is the realization that there are two kinds and that they are in no way of finding substitutes for each other. The acoustic materials mentioned above are insulating materials and absorbing materials respectively.

Insulating materials restrict the transmission of sound through themselves. A typical example is a brick wall. A less effective example is the metal wall of a duct. Note that these materials tend to be airtight and rather heavy. In fact a useful “rule of thumb” for checking insulation is first to check whether it is airtight (even small holes can have serious effects) and then to check the weight per square foot (superficial density) at the lightest point. The denser materials usually give greater insulation. For example, sheet lead can be particularly effective.

It is worth noting that although insulating materials restrict the passage of sound, they do not make it disappear. It is usually simply reflected back in roughly the same way as a mirror reflects light and this often causes a build up of sound on the source side.

Absorbent materials do not reflect sound (or only reflect a small proportion of it). Typical absorbers are mineral wool and open windows. When sound falls on these it does not return. In the case of mineral wool and similar materials, some of it tends to be lost in the interstices of the material, but in the case of open windows it simply passes though. In both cases a high proportion of the sound passes through to the other side. It is therefore obvious that absorbent materials are not the same as insulting materials and cannot be used as such.

The absorptive material methods may be grouped together as:

Porous absorber

- Bulk porous materials from 10mm to 150mm thickness, are faced over rigid surfaces or over air spaces
- Common material – fibreglass, mineral fibre, open and closed cell foams, cellulose fibres formed by binder, heavy drapery, deep pile carpets, sintered metal sheets
- May be faced with perforated sheets for protections and containment (perforate open area usually >30%)
- α depends only on ($R\iota$ / ρc) and ($f\iota$ / c),

where R is the material flow resistance and

ι is the material (or material + cavity) depth. For fibreglass, R depends only on the fibre diameter. $\iota > \lambda/5$ for maximum α

ρ is the density of material used (kg/m^3)

c is the velocity of the sound (m/s)

f is the flow rate of the sound

Table 2.5 : Appropriate values of directivity factor and directivity index Density, Acoustic Velocity and Characteristic Impedance of Various Materials (20°C and 1 Atmosphere)

Material	Density, ρ (kg/m^3)	Velocity, c (m/s)	Characteristic Impedance ρc (kg/m^2s)
Air	1.21	343	415
Carbon Dioxide	1.84	267	481
Methane	.68	445	304
Pure Water	998	1483	1.48×10^6
Aluminum	2700	6370	17.2×10^6
Crown glass	2500	5660	14.1×10^6
Perspex	1190	2700	3.21×10^6
Hard rubber	1100	2400	2.64×10^6
Mild steel	7800	5960	46.4×10^6
Concrete	2600	3100	8.00×10^6

Panel absorbers

- These are usually thin damped panel layer, suspended over backing air space
- The absorption peaks at the following frequency:

Absorption Peak Frequency, $f_p \sim 60 / (md)^{1/2}$

Where m is mass density of panel (kg/m^2) and

d is the cavity depth (m)

- It is normally used at frequencies below 200 Hz
- Some of the typical materials used include felt, plywood panels and lead vinyl

Volume Resonators

- These would help to reduce sound energy by the viscous dissipation at opening and absorptive material within cavity
- It is useful in high temperature applications, contaminated flow applications, as well as architectural applications

2.7.2.3 Vibration Damping

This is the most logical way to reduce noise radiation from vibrating structures. The term “damping” refers to the design property of materials which converts vibration energy into heat energy.

There are basically three measures that are frequently used to qualitatively describe a material’s damping characteristics:

1. Loss Factor, η
2. Decay Rate, Δ
3. Damping Factor, ζ

The Loss Factor, η, is the dimensionless damping-stiffness ratio of a visco elastic material.

Another common measure of vibration damping is the decay rates, Δ, expressed in decibels per sound, and is defined as the rate of natural attenuation for free vibrations within a material. The decay rate is related to the loss factor by an approximate relationship:

Constrained layer damping involves sandwiching a layer of visco elastic material between the structure being damped and an outer constraining layer. This type of damping finds application where structural members are quite thick, or where a large vibration reduction is required.

2.7.2.4 Vibration Isolation

To isolate unwanted fan vibration, which was once sufficient to mount the offending equipment on a piece of cork or felt. But today, with greater knowledge and higher expected standards, this traditional solution is usually inadequate, both in large building structures and in the general industrial environment.

In building construction, improved design allows lighter but inherently more flexible structures to be used. At the same time, increasingly powerful equipment is needed and may often be installed in upper level plant rooms. Fan vibration, therefore, can be a major problem in such structures.

Isolation of vibration is accomplished by supporting the equipment on resilient mounting elements such as springs or rubber, which compress under the equipment weight. The degree of isolation achieved is directly related to the amount of compression (i.e. static deflection) of the mounting.

The greater the static deflection, which can be achieved (without compressing to solid) the better will be the resulting vibration isolation.

When determining the level of isolation efficiency, which might be acceptable in any situation, consideration must be given to the following: Fan type and operating weight. Magnitude and nature of the vibrating forces. Restrictions on fan motion. Location of the fan in the building structure.

Some simple guidelines for successful vibration control:

1. Mounting support points and load ratings should be selected so that the static deflections of all mountings are as uniform as possible.
2. Unrestricted movement of resiliently mounted equipment is essential for effective isolation.

- Ensure that adequate clearance is maintained around the installed equipment, particularly underneath it, to permit free movement - especially where high deflection mountings are used.
- All connections to resiliently supported equipment should themselves be flexible. As well as restricting equipment motion, any fixed connection can offer a direct path for transmission of vibration to the surrounding structure, bypassing the isolation system.

3. Top-heavy machinery, especially when mounted on a narrow base, can become unstable if mountings are located too close to each other beneath

the equipment. Such instability can be avoided by use of outrigger brackets which space the mountings further apart and raise the mounting location points closer to the vertical centre of gravity of the equipment.

4. A rigid base is essential for resiliently supported equipment to avoid misalignment of drive components. Any flexibility in the machine base, should be eliminated by the addition of steel stiffeners or use of a concrete inertia base.

Ductwork and ancillary equipment connections should not impose dead loads on the resiliently mounted equipment. Their weight should be separately supported or allowed for when calculating the total weight.

The table provides a guide to the isolator type and necessary static deflection for given values of operating speed and isolation efficiency. It also suggests appropriate levels of isolation efficiency for various operating locations.

To use the table, read the minimum static deflection directly against machine disturbing frequency (operating speed) and required isolation efficiency.

Two static deflection figures appear, one for basement or on-grade installations; the other for upper level installations where some allowance is made for flexibility of the supporting structure. Select the appropriate figure.

The Isolation Efficiency Chart is a chart that illustrates the theoretical relationship between isolation efficiency, disturbing frequency and static deflection for a simple isolation system on a rigid foundation. It is also grouped into zones suggesting isolation efficiency ranges appropriate to different applications.

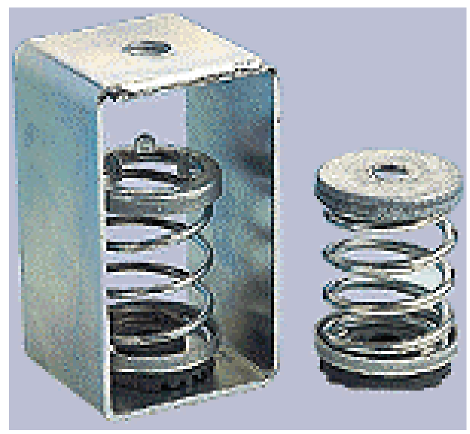
To use the chart, determine the lowest rotational speed of the equipment and consider this to be the disturbing frequency. Move vertically to intersect the diagonal line corresponding to the percentage isolation required; then move horizontally left and read the static deflection required of the mounting.

Table 2.6 The Isolation Efficiency Chart

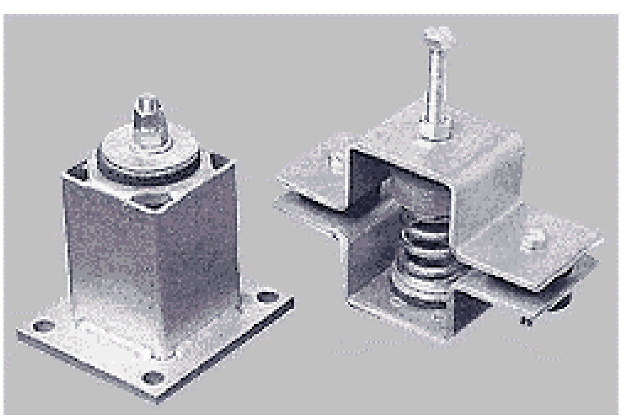
		Isolation Efficiency, 80%		Isolation Efficiency, 90%		Isolation Efficiency, 95%		Isolation Efficiency, 98%	
	Distribution Freqency	Ground Floor	Upper Floor	Ground Floor	Upper Floor	Ground Floor	Upper Floor	Ground Floor	Upper Floor
	Rps (Hz)	Isolator Static Deflection, mm							
High Defl. Springs	3.3	125	-	-	-	-	-	-	-
	5.0	60	90	110	150	-	-	-	-
25mm Defl. Springs	8.3	20	35	40	50	70	90	-	-
	11.7	11	18	20	27	0	50	100	120
Rubber Mount	16.7	6	10	10	15	18	25	50	60
	25.0	3	5	5	8	8	11	20	25
Pad Mount	33.3	2	4	4	6	6	8	11	15
	50.0	0.8	1.5	1.5	3	4	5	7	10
		Non-critical Areas		**General Areas**		**Critical Areas**		**Critical Areas**	
		Factories, Workshops, Garages, Warehouses, Laundries, Basements.		Schools, Dept. stores, Super-markets, Telephone exchanges, Hotels.		Multi-storey buildings, Offices, Hospitals – service area, Churches, Schools, Restaurants.		Multi-storey buildings, Hospitals - ward areas, Broadcastings studios, Theatres, Auditoriums, Libraries.	

Figure 2.2 Types of Vibration Isolators

(a) Rubber-in-shear Type



(b) Spring Mounted Type



(c) Seismic Mounted Type

2.7.2.5 Mufflers and Silencers

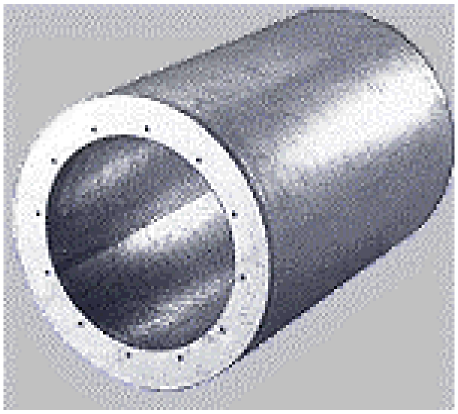
Mufflers and Silencers are of special interest to heating and ventilating engineers because they allow the passage of air while restricting the passage of sound. They usually subdivide the airflow into several passages each lined with perforated sheet backed by mineral wool, fibreglass or some other sound absorbing material.

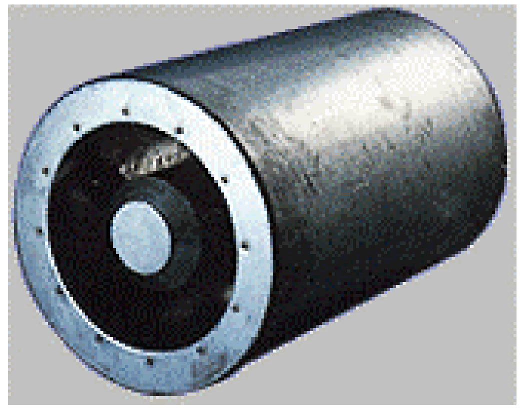
Silencers are generally specified by the sound attenuation in decibels which is provide in each octave band, so that the degree of attenuation matches with the sound power distribution of the noise source over the frequencies. This is known as the insertion loss of the silencer.

The other important parameter associated with silencers is the resistance to airflow. It would clearly be unsatisfactory to introduce so much resistance against the fan (in order to absorb the noise) that the fan speed had to be increased, thereby generating more sound and incurring additional power consumption which is both impractical and uneconomical.

Figure 2.3 shows the three of the available types of silencers that is available in the market.

Figure 2.3 Type of Silencers





(a) Circular Duct Silencers (Open Type)

(b) Circular Duct Silencers (Pod Type)

c) Rectangular Duct Silencer

With the ever-increasing air velocities in modern mechanical ventilation systems, it has become necessary to take into account the noise generated by the turbulence created due to the presence of the silencer in the air stream. This is known as the self-noise of the silencer.

To ensure maximum effectiveness, silencers must be correctly located in the duct system. The location in general must be such that the breakout noise or flanking does not present problems. The optimum position of silencers may vary from one installation to another, but the following points are usually applicable.

The silencers should be located as close to the fan as possible, particularly if the duct immediately downstream of the fan is over a critical area. This will reduce the in-duct power level and thereby ensure a minimum of breakout from that duct.

If the duct immediately downstream of the fan is over an area not sensitive to noise and the duct passes through a wall before it is over a relatively quieter area, the silencers may be placed in the partition as shown in Figure 2.4. This will prevent the noise from the plant entering the duct (i.e. break-in) reaching the adjacent area.

Frequently, however, fire dampers are required in the plant room wall and therefore the silencers may have to be located in a position other than as shown in Figure 2.4. In such situations the silencer should be located in the room side of the fire damper. The silencers and any duct between the silencers and the wall must be encased from the wall to the far end of the silencer by a suitable material.

The practice of locating the silencers as shown in Figure 2.5 should be avoided. Such a location is incapable of preventing noise breakout or break-in and reduces the effectiveness of the silencer.

Figure 2.4 Suitable location of a silencer

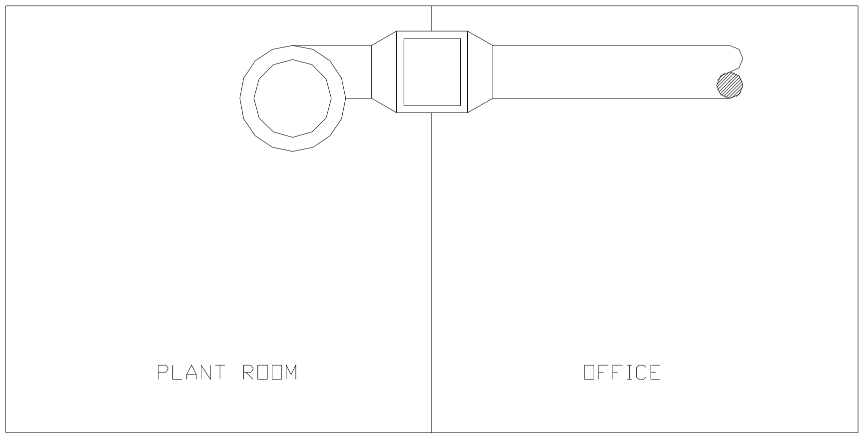
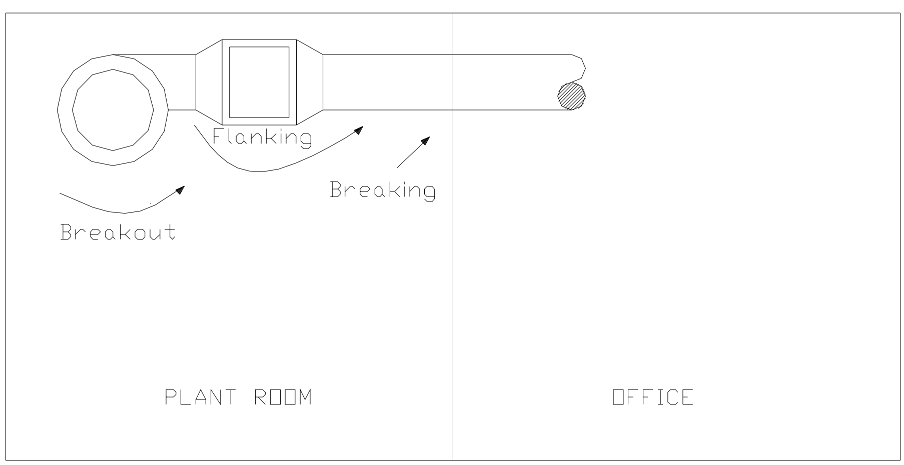


Figure 2.5 Unsuitable location for a silencer



There are six basic types of silencers:

1. Absorptive silencer – this is the most common type of silencer and takes the form of a duct lined on the interior with a sound absorptive material.
2. Reactive expansion chamber – this type reflects sound energy back toward the source, so as to cancel some of the oncoming sound energy.
3. Reactive resonator – this type functions in approximately the same way as the reactive expansion chamber type.
4. Plenum Chamber – this device allows the sound to enter a small opening in the chamber; that sound which has not been absorbed by the chamber's acoustical lining leaves by a second small opening, generally at the opposite end of the chamber.
5. Lined bend – sound energy flowing down a passage is forced to turn a corner, the walls of which as lined with acoustical material. The sound energy is thus forced to impinge directly on a sound absorbing surface as it reflects its way around the corner; each successive impingement takes sound energy from the traveling wave.
6. Diffuser – this device doesn't actually reduce noise. In effect, it prevents the generation of noise by disrupting high-velocity gas streams.

Chapter 3 Overview of Present Mechanical Ventilation System

3.1 System Configuration

As in most cases, the fan is the prime source of noise in any ventilation system. A small part of the horsepower supplied to the fan is radiated out as sound power. The higher the power supplied to the fan, the greater will be its acoustic power.

Induced draft, forced draft and ventilation fans emit noise that may be a source of community annoyance. Fan noise usually combines tones at the blade passage frequency with low frequency tones and rumbles. This noise may have substantial energy at low frequencies, manifesting itself in a characteristic rumble or it may be identified as the source of tones at the blade passage frequency.

Figure 3.1 : Plan Elevation of the Existing Plant Room

(All dimensions are in millimeter)

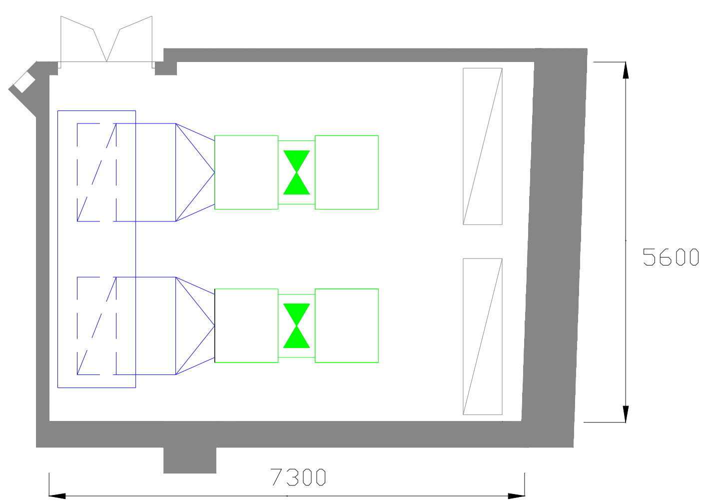


Figure 3.1 shows the plan elevation of the plant room layout and the indication of the parameters of the room. The number of existing components such as plenum box, mechanical ventilation fan, silencers and mansory shaft has also been indicated in the plan elevation.

Figure 3.2 : Isometric View of the existing Plant Room

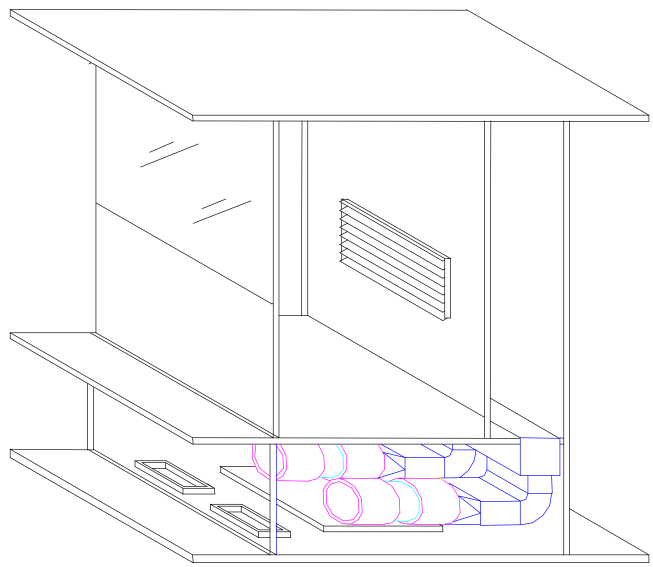


Figure 3.2 shows the isometric elevation of the plant room, the external lourve and the walkway. From the figure, it is noted that the plant room contains two sets of mechanical ventilation fans mounted on the slab. These ventilation fans are then coupled to a common exhaust air plenum box before it is being exhausted out to the walkway through the lourve located on the wall next to the building side entrance. The details of the various components are clearly shown in Figure 3.3 and Table 3.1.

Figure 3.3 : Part List of the various components in the existing plant room

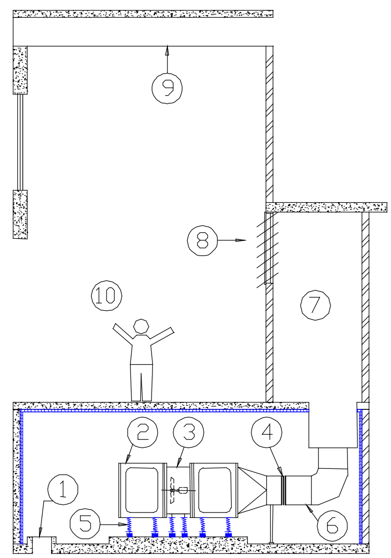
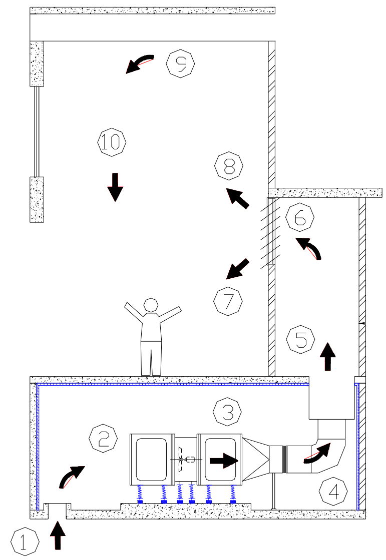


Table 3.1 : Label indication of the existing plant room for Figure 3.3

Label	Description
1	Mansory shaft (Exhaust Air)
2	Silencers / Attenuators
3	Mechanical Fan
4	Damping
5	Vibration Isolators
6	Exhaust Duct
7	Exhaust Chamber
8	Exhaust Air Lourve
9	False Ceiling
10	Walkway

Table 3.1, indicates clearly all the major components that is in the existing setup. This is to ensure that all components could be easily identified in the process of the analysis of the existing configuration.

Figure 3.4 : Flow path of air and noise



The Figure 3.5 illustrates the airflow path of the existing configuration. It shows the flow path of the noise transmission in the existing mechanical ventilation ducted system. First exhaust air is being extracted out from the basement carpark through the mechanical ventilation mansory shaft as shown in figure 3.4, indicated by (1) and (2). As the air is being drawn out from the mansory shaft, it passes through the mechanical ventilation fan and brings along the generated noise from the mechanical ventilation fan along its flows path as indicated by (2) and (3). Upon leaving the mechanical ventilation fan, part of the air is being absorbed by the silencers, which is located at both the upstream and the downstream of the fan. To prevent further transmission of noise from the vibration of the fan at this point, vibration isolators are used to isolate the vibration from the fan to the building structure. Flexible connector such as canvas is used as indicated by (3) and (4). This is to prevent the vibration transmission from the ventilation fan from the mechanical ductwork, which in term would transmit and generate the noise. Therefore, as the air and noise generated by the ventilation fan is being drawn through to the mechanical ventilation ductwork, noise is not re-created or re-generated in the process. As the exhaust air from the carpark is being extracted out of the ductwork, it is allowed to flow into another mansory shaft before flowing out of the lourve located along the way of the walkway, through the pressurization of the mansory shaft, indicated by (5) and (6). As the air flow through the lourve to the walkway, it transmitted the noise to it. The result of noises transmitted via the direct and reverberation paths, which is cumulative as are any other noises entering the space as indicated by (7), (8), (9) and (10).

The paths of noise transmission from the mechanical ventilation fan may be summarized as:

- Noise from the inlet and outlet
- Noise radiated through the fan casing
- Noise radiated through the ducts
- Noise induced by vibrations transmitted to adjoining structures

3.2 Consideration for Noise Control

All types of fans, whether axial, centrifugal or mixed flow, generate noise in the process of its operation. However, the character of the noise, that is, the frequency content, may vary considerably with each group according to the individual mechanical ventilation fan design and the respective manufacturer. Normally over the range of fans static pressures of 12 in. of water or more, it is usual for the centrifugal fan to produce more low frequency noise than the axial type. In the case of a fan, most of the noise is generated by the blades. As these blades move against the air resistance, a complex pattern of vortices and turbulence is set up. This causes a series of small air pressure waves (i.e. above and below atmospheric pressure) to radiate out from the fan. These pressure waves can be channelled down the connected ducts to other the ventilated zones.

These small pressure waves generated cause the ear to react and consequently the brain to register sound. Similarly the microphone of a sound level meter can pick up these pressure waves and register the sound pressure level (SPL). The generation of these sound pressure waves requires energy. The amount of energy required is very small in comparison with that required by the fan to its work, but nevertheless sound pressure requires energy. The rate of its expenditure is therefore power, sound power. It is not practical physically to measure sound power, but it can be readily calculated from the measurable sound pressure level (SPL) which a fan creates, provided that certain practical test conditions are observed by the fan manufacturers.

The first step in the reduction of noise is to define specifically how the acoustic energy is being generated. All noise sources generate sound by one of the following two mechanisms:

1. Acoustical radiation from a vibration surface
2. Aerodynamic turbulence

Vibration Radiation

Sound may be produced by the movement of a vibrating structure, which in turn sets into motion the air molecules that are coupled to it.

The following are the steps for the noise control of mechanical ventilation fan:

(a) All fan rooms shall be acoustically treated if it is necessary to minimize breakout noise

(b) All fans shall be provided with vibration isolators to minimize the generation and transmission of noise through vibration.

(c) For control of airborne noise upstream and downstream of the fans, attenuators shall be provided for high capacity fans

(d) For control of noise break out from fan casing, in critical areas, fan enclosures will be recommended

(e) Ducts connected to fans should be decoupled with flexible connectors to prevent the transmission of vibration

In the current case, noise control can be achieved by absorbing some of the sound power generated by the fan. Generally, the lower the frequency the more sound absorbent material is required and the more air space is needed to contain it. Both cost money and it is to the disadvantage of the centrifugal flow fan that it generates most of its sound power at lower frequencies than the axial type. Thus centrifugal flow fan requires more absorption material, more space and higher cost to make it acceptable in many systems. Absorption may be carried out by lining sections of distribution ductwork, particularly at bends, or by fitting packaged sound absorbers (attenuates, silencers, mufflers) immediately adjacent to the fan. Usually it is more convenient and cheaper to use a packaged silencer, and to place it immediately adjacent to the fan so that the sound power is reduced at source before reaching the distribution duct system.

Chapter 4 Comparisons and Analysis

The sound analysis of a mechanical ventilating system can be broken down into three parts :

a) The sound power source, which is the mechanical ventilation fan that is generating unwanted sound.
b) The reduction of sound power through the distribution ducts due to acoustic losses
c) The resultant sound pressure levels in the ventilated zones or the sound pressure received by the receiver

This project involves the appraisal of the acceptability of these sound pressure levels and corrective action if necessary to make them acceptable. The design criteria was identified earlier based on constant background noise sources to be 65 ~ 80 dB(A). The background noise mainly involves traffic noise. This background noise is not constant due to the periods of off-peak and peaks traffic. Furthermore, disturbances of the occasional “noisy” trunks, buses or motorcycles have to be taken into the consideration. These complex issues make the determination of the background noise source unfeasible. For this project, we would ignore the background noise as we have no control over these sources and cost incurred for the site measurement.

In order to be in-line with the list of objectives and the requirement specified by the sponsor to comply with the local government requirements, this research project will therefore, mainly focus on the calculation of the noise reduction emitting from the fan. This would be used as submission in the later date to the local authority for compliance to the local regulation.

4.1 Comparison of Designs

Figure 4.1 : Side elevation of the existing plant room

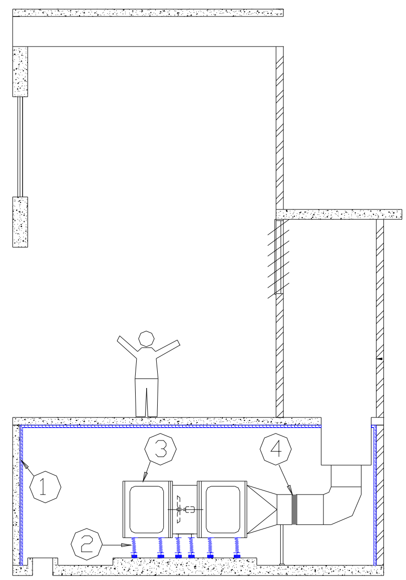


Table 4.1 : Label indication of the existing Plant Room for Figure 4.1

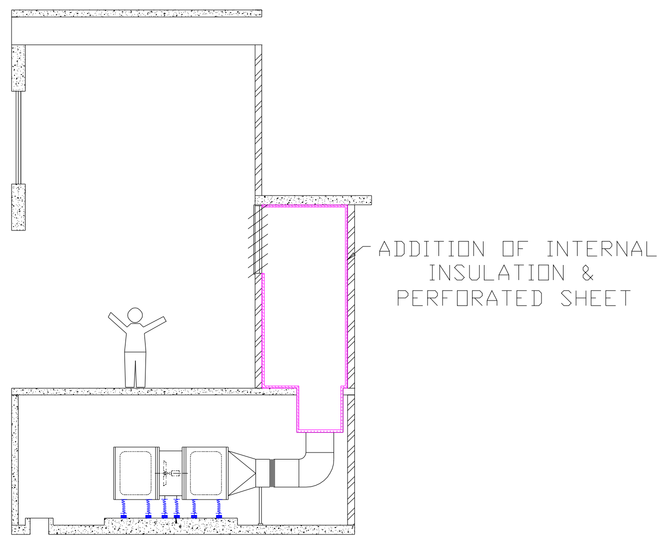
Label	Description
1	Acoustic Material with perforated sheet
2	Vibration Isolators
3	Silencers / Attenuators
4	Damping

Figure 4.1 shows the side elevation of the plant room. From the figure, it is noted that the existing setup has already equipped with the noise control items as listed in table 4.1. As discussed in the earlier chapter, these control items that is present in the existing plant room is sufficient to control the generation and transmission of noise in the plant room.

After analysis the existing setup, it is found that the mansory shaft before the external lourve is not complete with the installation of any acoustic material, therefore we would suggest that a suitable material be used for this case.

4.2 Proposed System Configuration

Figure 4.2 : Proposed System Configuration



In the new configuration, we would be looking into the details of treating the exhaust air mansory shaft, located just before the lourve. We would also be looking into the details of the acoustic material to be used.

Plenum Chambers

Lining a plenum chamber usually proves to be more economical than lining a section of duct. This is especially true if low frequency attenuation is wanted, and it may justify the insertion of a chamber into the air distribution system purely for that reason.

In the first place, there will be reflection of sound at the inlet to the plenum back along the inlet duct. Of the sound energy, which enters the chamber, part will be radiated directly to the outlets, part will enter the outlet ducts after repeated reflection from the chamber walls, and the lining will absorb the rest. The directly transmitted part is proportional to the area of the outlet and falls with the square of the distance of the inlet from the outlet. It is also reduced by a directional effect if the outlet is not immediately opposite the inlet, and it will disappear altogether if outlet and inlets are on the same wall.

This is because when sound waves strike the surface of a material, a fraction of the incident energy is absorbed by conversion to heat. All materials absorb sound to some extend; acoustical materials are those materials whose primary function is to absorb sound. Therefore they absorb a large fraction of the acoustical energy which strikes them.

The sound absorption coefficient of a material is a measure of the sound absorptive property of a material; ideally, it is the fraction (expressed as a decimal number) of the randomly incident sound power which is absorbed or otherwise not reflected by the material. The sound absorption coefficient of every material varies with frequency. It is common practice to list the coefficient of a material at six frequencies: 125, 250, 500, 1000, 2000 and 4000 Hz.

In addition to sound absorption, a number of other properties must be considered

In the selection of an acoustical material, the other acoustic properties to be considered include:

1. Flame spread and fire endurance
2. Mechanical strength, abuse resistance
3. Dimensional stability
4. Light reflectance
5. Maintenance, cleanability, paintability
6. Appearance
7. Cost
8. Ease of installation, method of mounting
9. Space availability for acoustical installation
10. Weight of acoustical installation
11. Compatibility with other materials and components

Specifications usually require that an acoustical material be rated for flame spread and fire endurance. The flame-spread index of a material is a measure of the rate at which flames will travel across the exposed surface of the material. This index compares the rate of flame travel with a selected species of untreated red oak flooring (assigned an index of 100) and with cement asbestos board (assigned an index of 0.) Since the ranges of performance are more significant than individual values the flame spread index is divided into the following classes:

Table 4.2 : Index of a material

Class	Flame spre1(a)d index	Federal Specification SS-S-118a
I	0-25	Class 25
II	26-75	Class 75
III	76-200	Class 200
IV	Over 200	-

Most acoustical materials are in Class I. The fire endurance of a floor/ceiling assembly or a roof/ceiling assembly, of which the acoustical ceiling is but one of the components that is rated in hours or fractions of hours.

The mechanical strength of most acoustical materials is relatively low (i.e. they are easily damaged when struck). The surface of an acoustical material that is subject to abuse may be protected by perforated facings fabricated of metal plywood or hardboard or by metal screen or strips of wood.

An acoustical material is said to be dimensionally stable if its physical dimension do not change significantly with changes in humidity with changes in humidity and temperature. The dimensional stability of an acoustical materials depends on the fiber of which it is fabricated (inorganic material are more stable than organic material) and the binder which holds the fibers together (waterproof binders are more stable than non-water resistant binders such as starch).

In order to determine the effectiveness of the proposal, the comparison of the result in this project would purely be based on calculation.

4.3 Calculation of Noise Reduction

4.3.1 Consideration of Noise Reduction

The analysis of sound reduction can be very easily and conveniently carried out, by the following steps and methods.

The starting point of any noise calculation is the fan itself. For this it is obviously necessary to have accurate sound power levels in frequency octave bands. Accurate data for the actual working point of the fan can be taken from the original equipment manufacturer (OEM) supplier fan manual or directly from the OEM supplier. It must be remembered that measuring instruments and techniques for fan data do not permit of significance being placed in differences of less than 3dB.

The OEM fan suppliers will normally provide the full performance data on the attenuation given by their ranges of standardized silencers. If none of the standard silencers provides sufficient attenuation, then additional attenuation can be provided in the distribution ductwork by lining some lengths of duct, or preferably bends, with absorbent material. Alternatively, specially designed silencers for the system can be supplied by fan supplier upon request.

The reduction in sound power through distribution ducts is due mainly to the division of sound power between the various ducts. It is also influenced by the loss in power, which occurs at the outlets from the system due to 'end reflection'. This happens whenever there is an abrupt change of sound path area, that is, from the end of a duct into a room. Additional attenuation can be achieved by lining with acoustic absorbent material, particularly immediately upstream and downstream of bends where, due to multiple reflections, it is more effective.

The design for sound system provides a simple method of calculating both the reverberant and direct sound pressure levels, which result from the sound power fed into the zone. In most cases it is sufficiently accurate to judge the mean absorption coefficient if the reverberation time is not known.

Finally the resultant sound pressure level is compared with the Noise Criteria, which define acceptable levels in each octave bands for various types of zones. It will be noted that higher levels are acceptable at lower frequencies but not at the higher frequencies. This is because the ear is less sensitive to sound pressures in this region.

It is strongly recommended that the six octave bands given on the working sheet are calculated out so as to ensure a complete check against noise criteria levels. Additional data are given on plenum chamber silencing and on out of door directivity effects.

4.3.2 Calculation of Sound Power Level

Noise levels can be calculated instead of being measured. In some cases, calculation is preferable and may be the only practical method, for example, where there are relatively high residual noise levels, where future levels need to be predicted or alternative scenarios need to be compared. Calculation is also useful for large-scale noise mapping and where there is limited access to the measurement position. Calculation is normally performed in accordance with a national and source specified standard.
Given the details of the existing mechanical ventilation setup, the calculation of the sound power level can be calculated using the calculation spread sheet that has been derived using the following methods and steps.

a) The Fan Sound Pressure Level (SPL). The value of the fan sound pressure level can be easily obtained from the fan performance characteristics in the supplier technical manual.

b) The Fan Sound Pressure Level (SPL). The fan sound pressure level obtained from the supplier technical manual has been measured under free field conditions at a distance of three fan diameters from the centre of the fan. For example, the SPL of a 48in. diam. Fan is measured at 12ft. from it centre. In order to calculate the attenuation of fan sound through a duct system, the rate at which sound energy is fed into the system must be known, that is, the FAN SOUND POWER LEVEL (SWL). Table 4.3 gives the conversion factors from fan SPL to SWL.

Table 4.3 : To obtain the fan SWL in dB re 10^{-12} watts add the following factors related to the fan diameter

In.	Mm	Factor
12	300	+ 10 dB
15	380	+ 12
19	480	+ 14
24	600	+ 16
30	760	+ 18
38	960	+ 20
48	1220	+ 22
60	1520	+ 24
75	1900	+ 26

c) The Sound Pressure Level (SPL) for Fans Operating Parallel in System. The quality of ventilating system noise is dictated by its frequency content, that is, the way in which the sound power is distributed over the frequency spectrum. The spectrum is divided into eight octave bands covering the audible range. Table 4.4 and 4.5 give the factors to be added to the fan SWL to give the WSL in each octave band. In practice the lowest and highest octave band need not be considered for axial flow fans but not for centrifugal fan. Therefore, they cannot be omitted. Fan spectra are also given for each fan size and speed in the technical manual. These spectra cover in addition all fractional solidity fans. Note that the 'in duct' spectra apply because the fan sound power is being fed into a duct. The free field spectra apply where the power is being radiated into the open air, for example where an extract fan is at the end of a system. This is described under 'Directivity'.

Table 4.4 : Obtain spectrum identity related to fan diameter and speed

In	mm	320 400	420 500	520 600	700 900	920 1200	1400 1800	2800 3600
12	300	-	-	-	-	-	H_3	K
15	380	-	-	-	-	-	H_3	K
19	480	-	-	-	G	G	H_3	K
24	600	-	F	G	G	G	H_3	K
30	760	E	F	F	G	G	H_3	K
38	960	F	F	F	F	G	H_5	-
48	1220	E	F	F	F	G	H_5	-
60	1520	E	E	F	F	F	-	-
75	1900	E	E	F	F	F	-	-

Table 4.5 : Add the following factors to the fan SWL to give the octave band SWL values

Spectrum	125	250	500	1000	2000	4000
E	- 5	- 7	- 8	- 12	- 18	- 24 dB
F	- 5	- 6	- 7	- 10	- 15	- 21
G	- 6	- 6	- 7	- 8	- 12	- 18
H_3	- 6	- 5	- 6	- 7	- 10	- 15
H_5	- 12	- 6	- 5	- 7	- 10	- 15
K	-10	- 7	- 5	- 7	- 8	- 12

d) The Insertion Loss of Silencers. A standard silencer may be used to reduce the amount of sound power fed into the duct system. Normally it is best to connect the silencer direct to the fan intake or discharge, or both, in order to reduce the sound level in the plant room or the zone in which the fan is installed. Other plant in the room may create sufficient sound power for part of it to be fed back into lightweight ducts within the plant room on the quiet side of the fan silencers, that is, by flanking the silencers. The sound power will combine with that already fed into the system and will cause a corresponding increase in the sound level near the final outlets. Therefore where the sound pressure level inside the plant room is high, any lightweight ducting within the room should be lagged, or the silencer should be located at the point where the duct leaves the room. Flexible connectors used with fans should be of noise insulting flexible material, such as lead-loaded rubber or PVC sheet, instead of canvas. Particular care should be taken to seal all joints to prevent further escape of sound energy. Most fan casings, due to their rigidity and weight, give an attenuation of at lease 25dBW in all octave bands to sound power escaping through the walls. Therefore they seldom require lagging to give additional attenuation.

e) The Sound Power Level Entering System. The sound power level fed into the duct system is therefore given by :

(Sound Power Level), e = a + b – c – d .

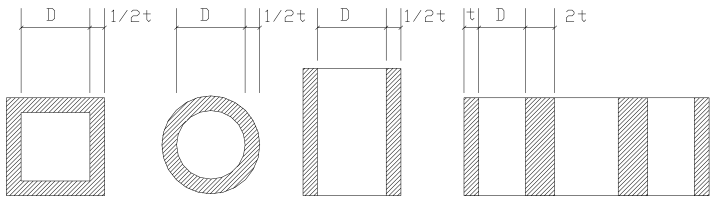
Sound reduction in the distribution ducts

Select the most critical ventilation outlet, normally the one nearest to the fan, and estimate the sound power reduction which occurs along the duct path to this outlet (see F and G) and at the outlet itself (H).

f) The Sound Reduction in Duct. Straight unlined ducts provide a small amount of sound power reduction dependent on the frequency and the duct size. An increase in sound reduction can be obtained by lining the duct with sound absorbent material, the thickness of which will influence the amount of reduction. The amount of sound power reduction in dB per foot length is given in Table 4.6. Note that the airway width dimension d is the minimum dimension of a rectangular duct or the diameter of a circular duct, taking into account the acoustic lining thickness, if applied.

The absorption coefficients of the lining material are given at the foot of Table 4.6 as a guide. It is not recommended that the dB per foot attenuation values be changed for materials with different absorption coefficients because their values may not truly reflect their performance under lined duct conditions. Remember that sound absorbent lining provides not only absorption coefficient but at the same time thermal insulation.

Table 4.6 : Straight Ducts approx. attenuation dB per foot length



Type	Width, d	Octave Band c/s					
		125	250	500	1000	2000	4000
Round and rigid walled ducts D = min, width	3" ~ 7"	0.03	0.05	0.05	0.1	0.1	0.1
	8" ~ 15"	0.03	0.03	0.05	0.07	0.07	0.07
	16" ~ 30"	0.02	0.02	0.03	0.05	0.05	0.05
	32" ~ 60"	0.01	0.01	0.02	0.02	0.02	0.02
Rectangular sheet-steel; see note (a) D = min, width	3" ~ 7"	0.2	0.15	0.1	0.1	0.1	0.1
	8" ~ 15"	0.2	0.15	0.1	0.07	0.07	0.07
	16" ~ 30"	0.2	0.1	0.05	0.05	0.05	0.05
	32" ~ 60"	0.1	0.05	0.03	0.02	0.02	0.02
Lined duct *(b) T = 1" 0.7 lb/sq ft rockwool	2"	0.2	1.8	5.5	11	17	19
	3"	0.2	0.8	3.6	8.5	12	12
	5"	0.2	0.4	2.4	5.5	7.6	6.4
	8"	0.2	0.2	1.6	3.8	4.6	2.0
	16"	0.1	0.1	1.1	2.2	1.0	0.1
Lined duct *(b) T = 2" 1.0 lb/sq ft rockwool	4"	0.9	2.7	5.7	9.0	10	8.5
	6"	0.4	1.8	4.2	6.2	6.2	4.4
	10"	0.2	1.2	2.8	3.8	3.2	1.0
	16"	0.1	0.8	1.9	2.3	1.0	0.1
	32"	0.1	0.5	1.1	0.5	0.0	0.0
Lined duct *(b) T = 4" 1.4 lb/sq ft rockwool	8"	1.4	2.8	4.4	4.7	4.3	1.9
	12"	0.9	2.1	3.1	3.1	2.2	0.4
	20"	0.6	1.4	1.9	1.6	0.5	0.1
	32"	0.4	0.9	1.1	0.5	0.0	0.0
	64"	0.2	0.5	0.3	0.0	0.0	0.0
Absorption coefficient of lining material		0.1	0.3	0.6	0.75	0.85	0.85

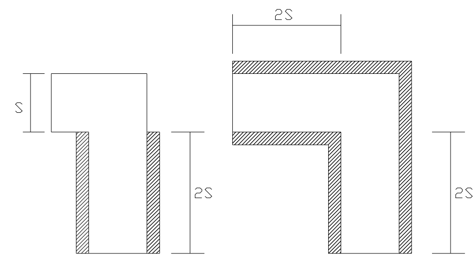
(i) Multiple by 2 at 125 and 250 c.p.s. if externally lagged

(ii) Linings assumed rigidly backed faced on inner surface with material of not less than 20% area perforation, the absorption coefficient at 1” thick being approximately as quoted above

(iii) Calculated values in excess of 40 dB may not be achieved without special precautions

g) The Sound Reduction in Bends. Bends with low loss, if viewed from the airflow point of view do not reduce sound power effectively. Elbows which provides a sharp 90 degrees turn in airflow direction, for example right-angled take-off branches, do provide a small measure of sound power reduction. This noise reduction can be increased appreciably by the installation of sound absorbent lining placed just after the elbow, where repeated takes place from wall to wall. Table 4.7 gives a summary of elbow attenuation related to the elbow minimum airway dimension.

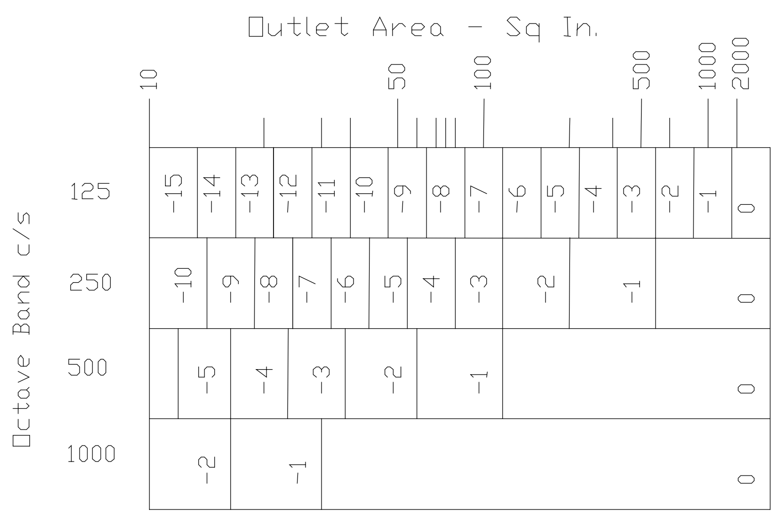
Table 4.7 : Bends



Widths	Spectrum	Octave band c/s					
3” ~ 4” 4/1/2” ~ 5/1/2”	A } B }	1000	2000	4000			
6” ~ 8” 9” ~ 11”	A } B }	500	1000	2000	4000		
12” ~ 16” 17” ~ 23”	A } B }	250	500	1000	2000	4000	
24” ~ 33” 34” ~ 38”	A } B }	125	250	500	1000	2000	4000
Long radius bends							
Turning vanes	A or B	0	-1	-2	-3	-3	-3 dB
Unlined 90° elbow	A B	-1 -4	-8 -7	-6 -4	-3 -3	-3 -3	-3 dB -3
Elbow lined after bend – Fig 1	A B	-2 -6	-11 -14	-13 -13	-10 -10	-10 -10	-10 dB -10
Elbow lined after bend – Fig 2	A B	-2 -7	-13 -16	-18 -18	-16 -17	-16 -17	-18 dB -18

h) Single Outlet Area. This is the sound power at high frequencies reaching the end of the Outlets. At high frequencies the whole of the sound power reaching the end of the duct emerges into the ventilated zone. At low frequencies, however, some of the sound power is reflected back and is absorbed within the duct system. This reflection effect is dependent on the gross area of the outlet and on the frequency. The relationship is given in Figure 4.3. Note that the gross outlet area means the whole cross-sectional area of the outlet when the subdividing bars, strips or louvers of the grille are removed.

Figure 4.3 : Single Outlet Area



i) Sound Power Level (SPL) leaving the system. The sound power emerging through the outlet is equal to the sound power fed into the duct system, less reductions which occur along the straight lengths of duct, at elbows and at the outlet itself :

i = e – f – g – h.

The ear and the sound level meter respond to sound pressure. The sound pressure level at the ear or at the meter microphone is depended on :

i) The reverberant sound pressure level set up by the total sound power fed into the room by all the ventilation outlets into the room, and the extent to which this sound power is reflected around the room or absorbed.

ii) The direct sound pressure level set up by direct transmission of sound power from the ventilation outlet nearest to the ear or the microphone.

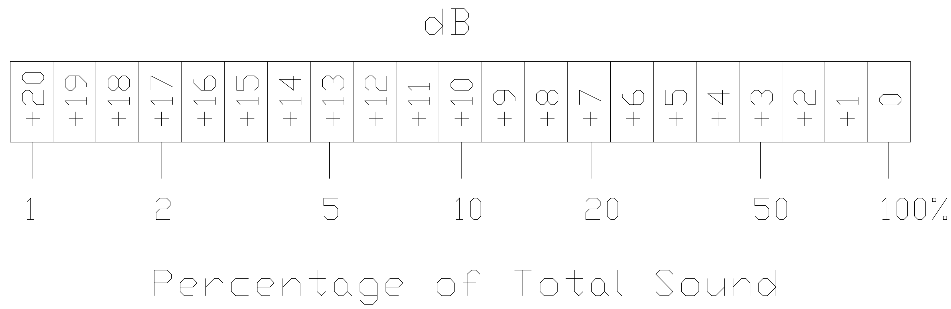
The combination of these two sound pressure levels is what is heard in a particular position in the room. If the room is live or reverberant the reverberant SPL will probably predominate. If, however, the ear is close to a ventilation outlet in an average or dead room the direct SPL is likely to predominate.

To estimate the reverberant sound pressure level

j) Percentage of Sound Reaching Room. First estimate the proportion of the total sound power created by the fan which will emerge from all the outlets into the room. The sound power from the outlets will be approximately in the same proportion as the total volume of airflow from these outlets to the total volume delivered in to the system by the fan.

Figure 4.4 gives the reduction in sound power level related to the percentage of total sound emerging from the outlets, that is, percentage of total air volume.

Figure 4.4 : Percentage of Sound Reaching Room



kl) Room Free Space and Absorption Coefficient or Reverberation Time.

The amount of reflection or absorption of the total sound power emerging from the outlets depends on the acoustic characteristics of the room. In terms of reverberant sound pressure level this may be estimated by : either Figure 4.5 and Figure 4.7 in which Figure 4.5 gives the effect of the area of the floor, walls and ceiling, and Figure 4.7 the effect of the mean sound absorption coefficient: Figure 4.6 and Figure 4.8 in which Figure 4.6 gives the effect of the volume of the room, and Figure 4.8 the effect of the rooms' reverberation time.

Figure 4.5 : Free Room Space - Area

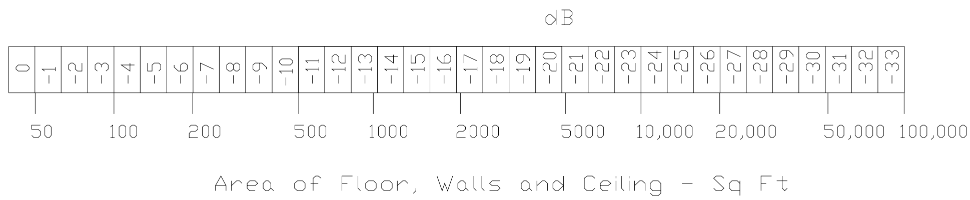


Figure 4.6 : Free Room Space - Volume

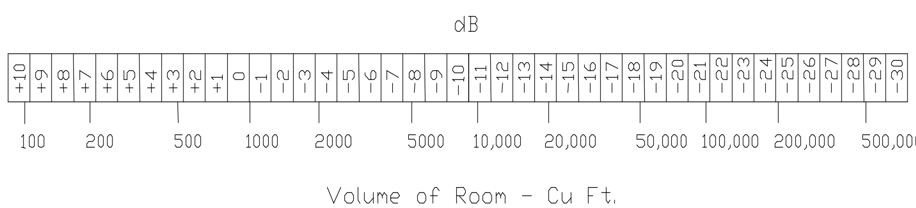


Figure 4.7 : Means Absorption Coefficient

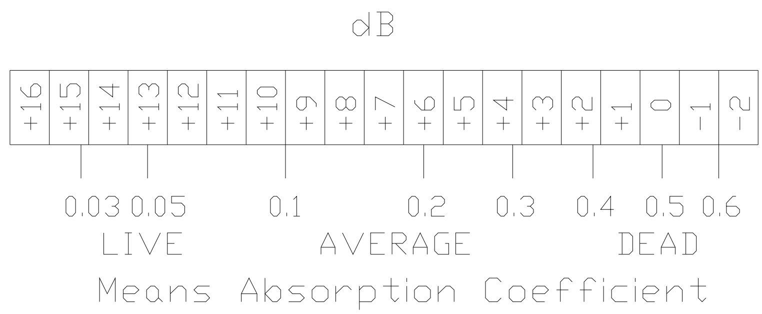
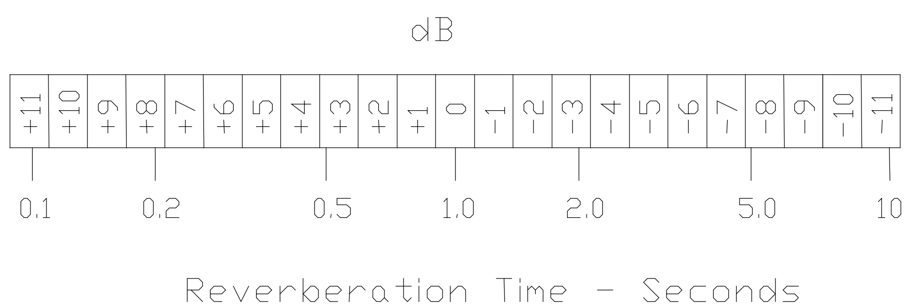


Figure 4.8 : Reverberation Time



m) The reverberant SPL is given by :

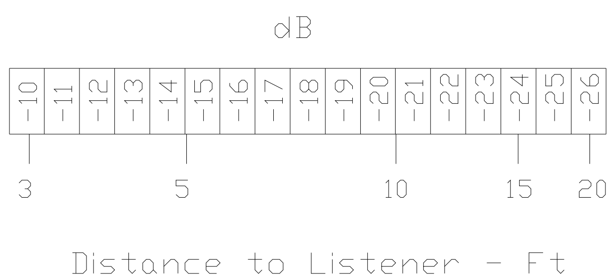
$$m = i - j + k + l$$

To estimate the direct pressure level

n) Percentage of Sound Reaching Outlet. First estimate the proportion of the total sound power created by the fan which emerges from the ventilation outlet nearest to the ear in the same way in J, but base this on the proportion of air volume flow from outlet only. Figure 4.4 again gives the reduction in sound power level for the single outlet.

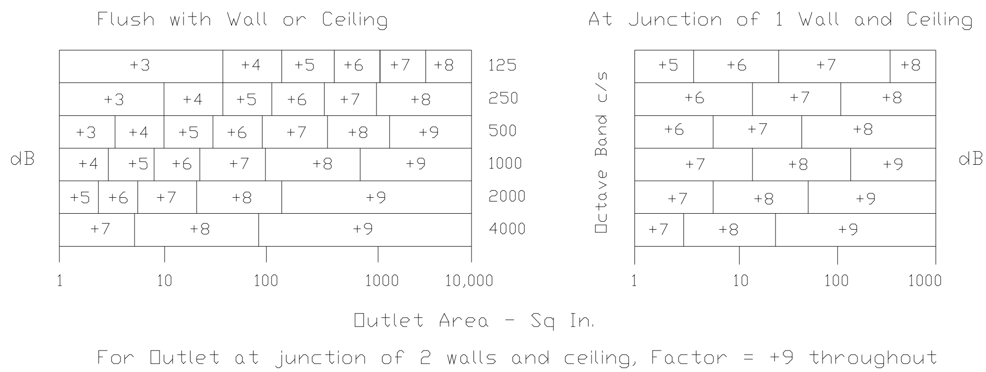
o) Distance from the listener. Estimate the effect on sound pressure level due to the distance between the receiver ear and the nearest ventilation outlet using Figure 4.9.

Figure 4.9 : SPL from a distance to the listener



p) The position of the nearest ventilation outlet in relation to the walls and ceiling of the room will affect the resultant sound pressure level, due to beaming or directivity. This effect will tend to increase the resultant sound pressure level, and may be estimated from Table 4.10.

Figure 4.10 : Directivity



q) The Direct Sound Pressure Level (SPL). The Direct SPL is given by :

q = i - n - o + p

r) Resultant Sound Pressure Level (SPL). To arrive at the resultant Room Sound Pressure Level it is necessary to combine the Reverberant and the Direct SPLs, using Table R. Since decibels are logarithmic units, arithmetic addition of sound power levels and sound pressure levels in dB cannot be carried out. Only when sound level reduction or attenuation is calculated may dB values be directly subtracted because in this context they represent a ratio of power or pressure reduction (Figure 4.9 and Figure 4.10)

s) Having arrived at the resultant room SPL it is necessary to consider whether this noise will be acceptable. Noise criteria (NC) curves enable one to assess acceptability. They weigh each part of the frequency spectrum separately and take account both of the varying sensitivity and take account both of the varying sensitivity of the ear to different frequencies and of annoyance factors. In particular they take account of the effect of Speech Interference that is of extraneous sound, which tend to interfere particularly with the transmission of speech in the 1000, 2000 and 4000 c/s octave bands. Noise criteria level, identified by NC numbers, together with a list of noise criteria acceptable for various environments. If the room SPL as calculated does not match the NC curve required, additional sound power reduction must be introduced into the system by one of the following means :

(i) Fit a fan silencer having greater sound attenuation, if neither of the silencers can provides the sufficient attenuation, the fans supplier can design a special silencer for the purpose.

(ii) Fit acoustic lining in part of the distribution ducting, preferable immediately downstream of a bend to obtain maximum effect

In the evaluating of any calculation, it is always noted that all calculated values would have a certain degree of error as compared to the actual site measurement values. This allowed variation is known as the tolerance.

In this case, calculation of sound pressure level, it is considered that sound pressure level measurement techniques do not justify significance being placed on differences of less than 5dB between calculated room SPL's and NC curves in octave bands of 125 c/s and higher.

Table 4.8 : Typical Sound Level Calculation Sheet

Sound Level Calculation (Discharge Side)

Fan Selection	Model	:
	Diameter	:
	Pitch Angle	:
	RPM	:
	SPL	: (a)
Duty	Air Flow	:
	Static Pressure	:

				Octave Band Mid-Frequency					
				125	250	500	1000	2000	4000
									(b)
(Add dB for fans operating parallel in system)									(c)
Insertion Loss of Silencers (Model :)									(d)
Add to obtain :	SPL entering system								(e)
Bend / Duct	Size	Length / Angle	Treat-ment						
Duct									(f)
Bend									(g)
Duct									(f)
Shaft									(f)
Single Outlet Area									(h)
SWL leaving system									(i)
Percentage of sound reaching room : 100%									(j)
Room (Area / Volume) : Free Space (cubic ft)									(k)
Absorption Coeff. or Reverberation Time - sec									(l)
Reverberant SPL									(m)
Percentage of sound reaching outlet – 100%									(n)
Distance to listener – 5 ft									(o)
Directivity : Outlet Area : (sq inch)									(p)
Direct SPL									(q)
Combine Reverberant SPL and Direct SPL : Resultant SPL									(r)
									(s)
Resultant SPL **dB(A)**									

Table 4.9 : Sound Level Calculation (before treatment of mansory shaft)

Sound Level Calculation (Discharge Side)									
Fan Selection	Model : AXV 1000-10/14 LH-4								
	Diameter : 1000 mm								
	Pitch Angle : 14 Degree								
	RPM : 1440 rpm								
	SPL : 107.5 dB								
Duty	Air Flow : 30,160 CMH x 2 nos								
	Static Pressure : 580 Pa								
				Octave Band Mid-Frequency					
				125	250	500	1000	2000	4000
				95	101	102	100	97	92
(Add 3dB for 2 fans operating parallel in system)				3	3	3	3	3	3
Insertion Loss of Silencers (Model 1000-1D)				-6	-12	-20	-19	-15	-14
Add to obtain :	SPL entering system			92	92	85	84	85	81
Bend / Duct	Size	Length / Angle	Treat-ment						
Duct	48” x 32”	3ft	None	-1	0	0	0	0	0
Bend	48” x 32”	90°	None	0	0	0	0	0	0
Duct	170” x 48”	3 ft	None	-1	0	0	0	0	0
Shaft	612” x 65”	13 ft	None	-2	-1	-1	-1	-1	-1
Single Outlet Area = 120” x 60” = 7200 sq inch				0	0	0	0	0	0
SWL leaving system				88	91	84	83	84	80
Percentage of sound reaching room : 100%				0	0	0	0	0	0
Room (Area / Volume) : Free Space (cubic ft)				-30	-30	-30	-30	-30	-30
Absorption Coeff. or Reverberation Time - 4sec				7	7	7	7	7	7
Reverberant SPL				65	68	61	60	61	57
Percentage of sound reaching outlet – 100%				0	0	0	0	0	0
Distance to listener – 5 ft				-15	-15	-15	-15	-15	-15
Directivity : Outlet Area = 7200 sq inch				8	8	8	8	9	9

Direct SPL	81	84	78	77	78	74
Combine Reverberant SPL and Direct SPL : Resultant SPL	81	84	78	77	78	74
Resultant SPL **88 dB(A)**	81	84	78	77	78	74

Table 4.10 : Sound Level Calculation (Mansory shaft with acoustic treatment)

Sound Level Calculation (Discharge Side)

Fan Selection	Model	:	AXV 1000-10/14 LH-4
	Diameter	:	1000 mm
	Pitch Angle	:	14 Degree
	RPM	:	1440 rpm
	SPL	:	107.5 dB
Duty	Air Flow	:	30,160 CMH x 2 nos
	Static Pressure	:	580 Pa

				Octave Band Mid-Frequency					
				125	250	500	1000	2000	4000
				95	101	102	100	97	92
(Add 3dB for 2 fans operating parallel in system)				3	3	3	3	3	3
Insertion Loss of Silencers (Model 1000-1D)				-6	-12	-20	-19	-15	-14
Add to obtain :	SPL entering system			92	92	85	84	85	81
Bend / Duct	Size	Length / Angle	Treat-ment						
Duct	48” x 32”	3ft	None	-1	0	0	0	0	0
Bend	48” x 32”	90°	None	0	0	0	0	0	0
Duct	170” x 48”	3 ft	None	-1	0	0	0	0	0
Plenum Chamber	612” x 65”	13 ft	2” int. Insulat-ion	-11	-16	-20	-21	-21	-21
Single Outlet Area = 120” x 60” = 7200 sq inch				0	0	0	0	0	0
SWL leaving system				79	76	65	63	64	60
Percentage of sound reaching room : 100%				0	0	0	0	0	0
Room (Area / Volume) : Free Space (cubic ft)				-30	-30	-30	-30	-30	-30
Absorption Coeff. or Reverberation Time - 4sec				7	7	7	7	7	7
Reverberant SPL				56	53	42	40	41	37
Percentage of sound reaching outlet – 100%				0	0	0	0	0	0
Distance to listener – 5 ft				-15	-15	-15	-15	-15	-15

Directivity : Outlet Area = 7200 sq inch	8	8	9	9	9	9
Direct SPL	72	69	59	57	58	54
Combine Reverberant SPL and Direct SPL : Resultant SPL	72	69	59	57	58	54
Resultant SPL **74 dB(A)**	72	69	59	57	58	54

4.4 Cost Evaluation

Basing on the requirement of the acoustic treatment to the existing mansory shaft, we have seek the assistance of the contractors in the field of mechanical ventilation acoustic treatment to advise on the installation cost.

Table 4.11, is a summary break down of the quotation which clearly indicate the budget cost for specified work. As this is a budgeted price, it is a normal practice in the market that it is about 20% higher than the usual price.

Table 4.11 : Summary break down of the quotation.

Item	Description	Unit Rate (S$)	Qty		Price (S$)
1	Modification of existing ductwork	100.00	8	Man-day	$ 800.00
2	Installation of rockwool and perforated sheet	50.00	172	Sq m	$8,600.00

Total Price $9,400.00

Chapter 5 Conclusions and Further Work

5.1 Achievement of Project Objectives

In summary, I would like to conclude that the list of objectives and requirements specified in the section 2.1 have been achieved.

These include the investigating the cause and the source of the noise and deriving the source of the noise at the mechanical ventilation system. Before a feasible solutions was developed to add on absorbing material on the mansory shaft for the improvement work and complying to the local authority regulation. Last but not least, the proposed addition and alternation work of the existing plant room is well within the budget of the sponsor.

5.2 Further Work

As most of the effort has already been spent in the designing of the system in this research project, suggestion that future work could be more focus on the on the checking area to improve on the current condition.

The following are the nine-point check plan:

1) When the plant room has been completed, check to ensure that no holes or gaps exist in it. If these are found, they should be filled with concrete or a similar impervious material. All holes through which ducts, pipes, etc. pass should be filled with a suitable material. Note that fiberglass, polystyrene or newspaper, etc are not suitable materials.
2) Adequate care must be taken to ensure that the anti-vibration mounting used with a compressors, pumps, etc. are not short circuited by rigidly fixed pipe work, electrical conduit, or builder's rubble.

3) Loose dampers in the ducts must be avoided, as these are a common source of noise. If found loose, they must be properly tightened. Check that the damper indicators indicate the damper position correctly.
4) Large unstiffened duct walls must be avoided, as they may result in drumming. Where this is so, adequate stiffeners of substantial rigidity should be provided. Where lagging of ducts is required, care must be taken to lag the entire surface. It is quite common to find the top of the duct, close to the ceiling, left unlagged.
5) Check for misaligned duct joints, and rectify.
6) Check for leaks in ducts. All holes required for pressure or velocity measuring apparatus or any other purpose should be sealed after tests have been completed.
7) Check that lighting and ceilings are not suspended from ducts.
8) Check that the ventilation system is properly balanced .
9) Installation of ductwork must be strictly supervised. In any one mechanical ventilation system, plastic coffee cups and lunch paper bags thrown in the duct during installation, if not this would cause considerable noise. Furthermore, clearing out such refuse can be very difficult once the system is in operation.

(15,500 words)

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Appendix A : Project Specification

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR	:	CapitaLand Commerical Limited
TOPIC	:	**Reduction of Noise Transmission at The Adephi**
SUPERVISOR	:	Dr Fok Sai Cheong
ASSOCAIATE SUPERVISOR	:	Mr Marcus Lim
PROJECT AIM	:	The project aims to study the cause of the noise and to develop solutions to minimize the disturbances caused by the noise to the tenants and shoppers at The Adephi, Singapore.
SPONSORSHIP	:	RESMA Property Services Pte Ltd

PROGRAMME: **Issue A, 18th April 2004**

1. Gather existing exhaust fan performance data (such as the breakout noise, and the airflow of the fan). This would also include gathering the various parameters (such as the exhaust air opening, duct size and the noise level both inside and outside the fan room)

2. Gather existing constructed As-Built Drawing

3. Analyze existing performance data, the various parameters and constructed drawing

4. Perform theoretical analysis using computer software

5. Submit proposals complete with theoretical analysis report on the achievable noise reduction.

AGREED: __________ (Student) ________________ (Supervisors)
Oliver Goh Dr Fok Sai Cheong

(Dated) 18 / 04 / 2004

Appendix B : Supporting Information