COMPUTER AIDED NOISE PREDICTION
IN
HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS

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FARUK EMRE GÜNGÖR

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Prof. Dr. Canan ÖZGEN
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

Prof. Dr. Kemal İDER
Head of Department

This is to certify that we have read this thesis and that in our opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

Prof. Dr. Mehmet ÇALIŞKAN
Supervisor

Examining Committee Members

Prof. Dr. Samim ÜNLÜSOY (chairman)

Prof. Dr. Kahraman ALBAYRAK

Prof. Dr. Gülin BİRLİK (ES)

Prof. Dr. Mehmet ÇALIŞKAN

Assoc. Prof. Dr. Cemil YAMALI

ABSTRACT

COMPUTER AIDED NOISE PREDICTION
IN
HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS

Güngör, Faruk Emre

M.Sc., Department of Mechanical Engineering

Supervisor: Prof. Dr. Mehmet Çalışkan

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This thesis aims at preparing a user-friendly software tool for the prediction and analysis of the noise generated in Heating, Ventilating and Air Conditioning (HVAC) Systems elaborating the standardized prediction formulae and data coming from the research studies. For the analysis portion of the software, different types of indoor noise criteria are introduced and implemented in the software to ease the investigation of the level and the quality of the sound perceived by the occupant in a room through such criteria. General software structure and implementation of HVAC elements are explained by different user-interface samples in the thesis. Several case studies are presented to demonstrate the capabilities of the tool prepared in VISUAL BASIC programming language within the scope of the study.

Keywords: Heating, Ventilating and Air Conditioning (HVAC) Systems, HVAC noise, noise prediction software, indoor noise criteria.

ÖZ

ISITMA, HAVALANDIRMA VE İKLİMLENDİRME SİSTEMLERİNDE BİLGİSAYAR DESTEKLİ GÜRÜLTÜ TAHMİNİ

Güngör, Faruk Emre

Yüksek Lisans, Makine Mühendisliği Bölümü

Tez Yöneticisi: Prof. Dr. Mehmet Çalışkan

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Bu çalışma; çeşitli araştırmalar sonucu elde edilmiş olan standardize bilgi ve formülleri kullanarak, Isıtma, Havalandırma ve İklimlendirme Sistemlerinde ortaya çıkan gürültünün tahminine ve analizine yönelik, kullanıcı arayüzü kolay bir yazılım programı hazırlamayı hedeflemiştir. Yazılımın analiz bölümüyle ilgili olarak; ortamda bulunan kişilerin algıladıkları sesleri ve bu seslerin kalitelerini incelemek amacıyla çeşitli tipteki iç mekan gürültü kriterleri tez kapsamında tanıtılmış ve yazılıma uygulanmıştır. Sözkonusu yazılımın genel yapısı ve yazılıma uygulanan farklı tipteki Isıtma, Havalandırma ve İklimlendirme Sistemi elemenlarının yazılımdaki uygulamaları kullanıcı arayüz örnekleri verilerek çalışmada anlatılmaktadır. Yazılımın yeteneklerini sergilemek amacıyla çeşitli senaryolar tez kapsamında incelenmiştir.

Anahtar Kelimeler: Isıtma, Havalandırma ve İklimlendirme Sistemleri, gürültü tahmini, iç mekan gürültü kriterleri, gürültü tahmin yazılımı.

To My Wife

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CHAPTER 1

INTRODUCTION

1.1. SCOPE AND OBJECTIVE

For environmental considerations in critical listening spaces, like conference rooms and auditoria, and for many other spaces with light building structures, the sound generated by mechanical equipment and its effects on the overall acoustical environment must be considered to enable the design of the Heating, Ventilating and Air Conditioning (HVAC) system of a building acceptable in terms of noise standards according to the international noise criteria. Thus, the selection of the mechanical equipment and the design of spaces should be undertaken with an emphasis on the goal of providing acceptable sound levels in occupied spaces of the buildings in which the HVAC equipment is in service.

Due to the necessity of a through noise analysis in the HVAC system design, it is intended to prepare a user-friendly noise prediction program within the scope of this study. This will also help the designer to overcome the difficulties encountered during the design phase of an HVAC system for evaluation of the resultant noise according to the several noise criteria.

While preparing the code of the software, the system concept of noise control is employed throughout the study such that each of the components in the HVAC system is related to the source-path-receiver chain. The noise is generated in the source; it travels from the source via a path, either airborne or structure born and it reaches the ears of the receiver. Several mechanical components, from dampers to diffuser junctions as well as prime movers or air handling equipment, are treated as sources of noise by the nature of the airflow through and around them. Sound attenuation provided by duct linings are also included in the procedure. Acoustical characteristics of noise receiving spaces are also considered in the prediction procedure. The resulting noise levels are compared to internationally accepted noise criteria.

Consequently, the primary concern of the thesis study is to prepare a prediction software to estimate the sound pressure levels due to certain noise sources at the prespecified receiver locations and to evaluate the overall system noise according to the several indoor noise criteria based on data and empirical results from previous researchers and experiences in the field. Analysis of the sound generated in HVAC components and elaboration of different types of sound rating methods are realized during the preparation of the infrastructure of the prediction software.

1.2. OUTLINE OF THE STUDY

In Chapter 2, previous studies and standards related to the noise of several HVAC components and studies related to the indoor rating of HVAC noise are explained. The chronographic progress of researches realized is examined in this section. Similar prediction software are also presented. The advantages of the prediction software in the study are explained.

In Chapter 3, primary noise sources and attenuators existing in the majority of HVAC systems are examined. The related prediction formulae and data come from standards and research studies are outlined. The data demonstrated hereby is also the backbone of the algorithm of the prepared prediction software. Since it is possible to face any type of equipment in a HVAC system, a wide variety of HVAC equipment is included in this chapter.

The second major objective of the study is to evaluate the overall noise levels of the HVAC systems in accordance with the international noise rating procedures. In this respect, four different noise rating procedure, namely Noise Criteria (NC), RC Room Criteria (RC), A-Weighted Sound Level Criteria and RC Mark II Room Criteria methods are explained in details in Chapter 4.

In Chapter 5, introduction of the software written within the scope of the study, its general structure, and implementation of HVAC elements employed are explained giving different user-interface samples. The sources used during the preparation of the software are also introduced in this section.

In Chapter 6, three different cases are elaborated gradually from a simple case to more complicated and real life case so as to demonstrate the capabilities and structure of the program in different manners. The scenarios are chosen so that different rating values in total are obtained like “neutral”, “hissy” and “rumble” tonal spectra. The explanation of the scenarios used in the cases, every calculation step executed in those cases and the result screens associated to each case are demonstrated in this section.

Finally, in Chapter 7, acquisitions gathered from the prediction software, an overall evaluation of the program depending on the case study results, and possible future enhancements that can be applied to the program are explained.

CHAPTER 2

LITERATURE SURVEY

2.1. GENERAL

Adequate noise control in heating, ventilating, and air-conditioning (HVAC) system is not difficult to achieve during the design of the system, provided that basic noise control principles are understood and exercised in the design phase. Based on this purpose, different kind of studies or researches has been realized for many years not only by institutes but also by many researchers. Additionally, some of these studies have given the way to some standards related to the HVAC systems still being used by the designers and manufacturers.

In this respect, this chapter gives information about publications, studies of researchers who studied on HVAC system noise, several standards issued relating to the noise of several HVAC components and the studies related to the indoor rating of HVAC noise. Besides, in this chapter, similar prediction software are introduced and criticized.

2.2. REVIEW OF THE LITERATURE

The control of background noise from HVAC systems in buildings has been of serious concern for many years. In the late 1940’s the boom in construction of high-rise centrally air-conditioned buildings was accompanied by an acute awareness of the excessive noise and vibration that HVAC systems were capable of producing. During this early period there was very little information available to answer what is the source components of this noise, to predict how people responded to noises of different levels. Thus, the HVAC design engineer was faced with the dilemma of needing to find solutions to his noise problems largely by trial and error; not only was the noise control technology in this field in its infancy, but practical guidelines and criteria for determining just how much noise was acceptable, marginal or objectionable were virtually unavailable.

By the late 1950's, however, considerable progress had been made in the development of HVAC noise control technology, and in the introduction of criteria for predicting the subjective response of people to noise.

In 1957, **Beranek** introduced the well-known Noise Criteria (NC) curves [1] to rate the HVAC system noises in terms of an NC level. The introduction of these curves represented a significant advancement in the state of the art at that time, and initially, there were great hopes for the success which would be found in application of these criteria in diagnosing noise problems and in setting system design goals. However, the field experience over the years has not been really very good with respect to correlating an NC rating of a given noise with people's attitudes about it.

In 1971 **Beranek, Blazier and Figwer** sought to correct the shape problems which had been found with NC curves by the introduction of a new family of contours termed PNC (Preferred Noise Criteria) curves [2]. These curves differed from the NC curves in two respects: firstly, a level was defined for the 31.5 Hz octave-band and, secondly, the spectrum slope was made less steep below 250 Hz and more steep about 1000 Hz. The experience with the PNC curves did not yield satisfactory results to most people's surprise.

Following years, several studies have been realized to add "quality" as a second agent to the "level" feature of sound rating methods. As a result, in 1981, Room Criteria (RC) rating procedure was introduced for rating HVAC systems including the quality feature [3]. Sound quality is the qualitative descriptor that identifies the character of the sound as perceived by the listener. The rating of the sound quality can be grouped in the following categories:

1. Neutral (N): a bland, unobtrusive sound that has no particular identity with frequency;
2. Rumble (R): an obtrusive sound in which there is an excess of low-frequency energy;
3. Hiss (H): an obtrusive sound in which there is an excess of high-frequency energy;
4. Tonal: an obtrusive sound which is distinctly tonal in character.

An enlarged methodology for the rating of the HVAC noise was initiated by an **American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE)** sponsored research project [4]. The refined methodology is identified as the RC Mark II procedure for rating HVAC-system related noise in buildings. Furthermore, ASHRAE has endorsed this refined methodology, and recommended as the preferred procedure for HVAC system sound rating in the edition of the ASHRAE Applications Handbook [5].

While sound rating methods are studied within the years, other types of researches and publications are issued like noise characteristics of HVAC system's components, i.e fan, variable air volume system, elbow, duct, return air system noise, etc. While some of these studies are executed by individual researchers, manufacturers' test data on HVAC system components are issued as an important data source. Even, some of those data are standardized and issued as formal data.

Basic sound and vibration principles and data needed by HVAC designers, information on acoustic design guidelines and system design requirements, most of the equations and tables associated with the sound control design in HVAC systems were published between 1991 and 1999 in **ASHRAE**

Handbooks [5], [6]. In addition to the fundamental ASHRAE publications, several other publications were also issued. Among those, **AMCA Standard 300** [7] or **ASHRAE Standard 68/AMCA Standard 330** [8] provided the approved test conditions for the sound power generated by a fan performing at a given duty. **Air Conditioning&Refrigeration Institute (ARI) Standard 880** [9] specified the procedures in order to obtain the sound data of Variable Air Volume (VAV) systems. **ASHRAE Standard 70** [10] or **ARI Standard 890 (P)** [11] provided manufacturers' test data for room air terminal devices such as grilles, registers, diffusers, air handling light fixtures, and air-handling suspension bars. **ARI Standard 575** [12] and **ARI Standard 370** [13] made possible to obtain factory sound data for indoor and outdoor chillers, respectively. **ASHRAE 1987** [14] handbook listed sound attenuation values for unlined round circular ducts. **ARI Standard 885** [15] provided insertion loss values for specified duct diameters and lengths for non-metallic insulated flexible ducts. **ASTM Standard E477** [16] outlined the test procedures to obtain sound data for dissipative and reactive silencers. **AMCA Standard 300** [7] gave the sound attenuation values associated with duct end reflection losses for ducts terminated into free space.

A wide variety of studies executed by researchers and universities in parallel to the studies of ASHRAE and similar institutions that work on HVAC systems. Among those, **Reynolds** and **Bledsoe** issued a publication covering technical discussions, detailed HVAC component and system design examples in 1991 [17]. **Schaffer** [18] provided specific guidelines for the acoustic design and related construction phases associated with HVAC systems, troubleshooting sound and vibration problems, and HVAC sound and vibration specifications in 1991. **Ebbing** and **Blazier** [19] clarified how users can make the best use of HVAC manufacturers' acoustical data and application information in 1998. **Reynolds** and **Bevirt** [20] studied on the instrument requirements, instrument and measurement calibration procedures, measurement procedures, and specification and construction installation review procedures associated with sound and vibration measurements relative to HVAC systems. **Beranek** [21], **Reynolds** and **Bledsoe** [17], **Reynolds** and **Bevirt** [20], and **Wells** [22] made studies about the transmission losses associated with the plenums. The attenuation values of straight unlined rectangular sheet metal ducts were introduced to the literature by **Cummings** [23], **Reynolds** and **Bledsoe** [24], and **Ver** [25]. **Kuntz** and **Hoover** [26], **Reynolds** and **Bledsoe** [24] tabulated the attenuation values of several rectangular sheet metal ducts for 25 mm and 50 mm duct lining, respectively. **Reynolds** and **Bledsoe** [27] tabulated the insertion loss values for dual-wall round sheet metal ducts with 25 mm and 50 mm acoustical lining, respectively. **Beranek** [28] studied on the insertion loss values for unlined and lined square elbows without turning vanes. **Ver** [29] issued the principle of distribution of the sound power contained in the incident sound waves in the main feeder duct between the branches associated with the junction. **Ebbing** [30], **Cummings** [23], **Lilly** [31] made some studies and put into forward some theories related to the sound radiated through HVAC system duct walls. **Schultz** [32] and **Thomson** [33] found that diffuse-field theory does not apply in real-world rooms with furniture or other sound scattering objects and they brought a new approach to the sound level predictions at a specific point in a room.

All these studies both realized by institutions and individual researchers show the historical and rapid development of HVAC industry in explanatory manner.

2.3. REVIEW OF EXISTING SOFTWARE

Some noise prediction programs were written for use in the HVAC system design studies and to reduce the burden of the HVAC designer to overcome the difficulties and time consumption encountered in making a tremendous amount of noise prediction calculations. The standardized studies of some institutions and the results of the previous research studies outlined above are used in a large scale in the preparation of such software.

Presently, there exist three major noise prediction programs extensively used for the noise prediction studies of HVAC systems. Those are V-

A Select Release 4.0, Trane Acoustics Program and the Applied Acoustics Program.

V-A Select Release 4.0 was developed as air-handling noise control software for use by HVAC designers and acoustical consultants to enable user-friendly acoustical analysis of HVAC duct systems. Since Vibro-Acoustics, that is the developer of the software, has been working on HVAC systems for 40 years, it has an extensive database of product testing and that is why V-A Select Release 4.0 software provides cost-effective silencer selections to meet project requirements in its software. This is the distinguishing side of the program to select a silencer to achieve specific insertion loss and pressure drop requirements and to perform silencer selections in advance of further acoustical and aerodynamic analysis in HVAC systems.

Trane Acoustics program was developed to help designers accurately model the sound level reaching building tenant's ears, the Trane Acoustics Program (TAP) projects equipment sound power data through the surroundings (e.g., floors, ductwork, walls), to estimate the sound level that will be heard. Industry-standard calculations published by ASHRAE's 1991 Algorithms for HVAC Acoustics handbook are the basis for this estimate. The conditions of an HVAC system can be modelled choosing specific equipment and building component criteria in the program. Program analyses the sound paths and calculates the total effect for the enclosed space.

The Applied Acoustics Program was developed to determine the acoustic quality of indoor and outdoor spaces. The program estimates the sound pressure level at a receiver location in response to one or more sound sources. It computes Noise Criteria, Room Criteria and A-Weighted Sound Level (dBA) acoustic ratings. The program allows the user to define one or more sound sources (e.g., HVAC equipment), the path(s) sound energy travels to a receiver, and one or more sound attenuation or regeneration elements in the path of the sound. From this information, the program then provides an estimate of sound pressure at each receiver using ASHRAE and ARI procedures.

All of the existing software for the prediction of the noise levels in HVAC systems show both some similarities and differences with the program written within the scope of this study. Since different companies developed different software products to serve themselves for their specific purposes in noise prediction applications, every program gives different flexibilities to the users. For example; Vibro-Acoustics company uses its years of experience to offer users silencer selections to be used in their projects. Similar to this concept, the program written in this study intended to enable the user to perform an acoustical analysis to estimate the resultant noise levels in occupied spaces in accordance with the latest standards and to provide the user the capability of comparing these noise levels to desired noise criteria including the latest updated ones. The program written in this thesis enables the user to compare the resultant noise levels even with a procedure including sound quality assessment named as RC Mark II method in addition to the conventional Noise Criteria, Room Criteria and A-Weighted Sound Level (dBA) Criteria already existing in the program.

In conclusion, the program uses the latest ASHRAE and ARI standards in its algorithms so as to give more accurate and up-to-date results. Additionally, it uses the latest noise criterion that includes sound quality assessment as a diverse feature during the comparison of the resultant noise levels to different noise criteria. Furthermore, it is aimed to provide a user-friendly interface for easy use of features including a built-in graph capability.

It shall be noted that new programs for the estimation purposes of HVAC system noise will definitely be developed in accordance with the developing technological conditions in HVAC systems by closely following new studies and standards that will be implemented by different institutes, researchers and manufacturers in the future.

CHAPTER 3

NOISE IN HVAC SYSTEMS

3.1. PRIMARY NOISE SOURCES

3.1.1. FAN NOISE

Aerodynamic noise from all types of fans may be broadly divided into a rotational component and vortex component. The rotational component is associated with the impulse given to the air each time a blade passes a given point and is hence a series of discrete tones at the fundamental blade passing frequency and harmonics thereof. The vortex component of noise is largely due to the shedding of vortexes from the fan blades. It is in random character and has a continuous spectrum over a wide range of frequencies determined by the fan geometry and operation. It may also have many non-harmonic single-frequency components determined by blade geometry and local air velocity. Since the laws of generation of these two types of noise are different, they will vary in importance for different types of fans and operating conditions.

In addition to aerodynamic noise, there are usually several non-aerodynamic sources of noise in equipment involving fans. Such sources include noise resulting from unbalance, bearing noise, brush noise, magnetic noise, and belt noise.

The number of blades of a centrifugal fan generally is governed by optimum airflow design. The noise generation decreases but slightly for more than the optimum number of blades.

Powerful tone of definite pitch may be generated in fans when the wakes from the impeller blades impinge on an obstruction such as a badly located or shaped cut-off in a centrifugal fan, or struts or motor supports which are badly shaped and too close to the impeller of an axial flow or propeller fan. This interference noise has the same frequency characteristics as rotation noise.

A shroud around a propeller fan may serve to reduce noise considerably if it is working properly. Such reduction is generally most effective at the higher harmonics. However, if the flow breaks down over part of the shroud, the noise may become considerably worse than for an unshrouded case.

As the operating pressure across axial fans is increased, the maximum sound intensity is shifted from the fundamental to higher harmonics. This effect is not observed for centrifugal fans.

The sound power generated by a fan performing at a given duty is best obtained from manufacturer's test data taken under approved test conditions. However, if such data are not readily available, the octave band sound power levels for various fans can be estimated by the procedure described below:

Fan noise can be rated in terms of the specific sound power level, defined as the sound power level generated by a fan operating at a specific capacity and pressure. By reducing all fan noise data to this common denominator, the specific sound power level serves as a basis for direct comparison of the octave band levels of various fans and as the basis for a conventional method of calculating the noise levels of fans at actual operating conditions.

On a specific sound power level basis, small fans are noisier than large fans. While size division is necessarily arbitrary, the size divisions indicated are practical for estimating fan noise. Fans generate a tone at the blade passage frequency, and the strength of this tone depends partly on the type of the fan. To account for this blade passage frequency, an increase in sound pressure should be made in the octave band in which the blade frequency falls into. The number of decibels added to the sound pressure level in this band is the blade frequency increment [6].

Blade Passage Frequency (B_f): The number of times per second a fan's impeller passes a stationary item. It is described in Hz.

Blade Passage Frequency can be calculated using the formula stated below.

$$B_f = \text{(rpm of the fan)} \times \text{(no. of impeller blades of fan)} / 60 \qquad (3.\ 1)$$

All fans generate a tone at this blade passage frequency and its multiples. Whether this tone is objectionable or barely noticeable depends on the type and design of the fan and the point of operation. Some types of commonly used fans are summarized below;

Housed Centrifugal Fans: Forward curved (FC) fans are commonly used in many air handlers. The blade pass of FC fans is typically less prominent and at a higher frequency than other fans. The most distinguishing acoustical concern of FC fans is the prevalent occurrence of low frequency rumble. FC fans are commonly thought to have 16 Hz, 31.5 Hz and 63 Hz rumble, particularly operating to the left of the maximum efficiency point.

Backward inclined (BI) and airfoil fans are generally louder at the blade pass-frequency than a given FC fan selected for the same duty; but BI fans are much more energy efficient at higher pressure and airflow. The blade pass tone generally increases in prominence with increasing fan speed and is typically in a frequency range that is difficult to attenuate. Below the blade pass frequency, these fans are generally have lower sound amplitude than FC fans and are often quieter at the high frequencies.

Plenum fans produce substantially lower discharge sound power levels if the fan plenum is appropriately sized and acoustically treated with sound absorptive material.

Vaneaxial fans generally have the lowest amplitudes of noise at low frequency of any fan. For this reason they are often used in applications where the higher frequency noise can be managed with attenuation devices. In the useful operating range, the noise from axial fans is a strong function of the inlet airflow symmetry and blade tip speed.

Propeller fan's generated noise generally has a low frequency dominated spectrum shape, and the blade passage frequency is typically prominent and occurs in the low frequency bands due to small number of blades. Propeller fans are most commonly used on condensers and for power exhausts [5].

The number of blades and the fan rpm can be obtained from the selection catalog. Table 3.1 lists specific sound power levels and blade frequency increments. Table 3.2 lists the octave band in which the BFI occurs. At present, the sound power level data for forwardly curved fans varies widely; the specific sound power levels given in Table 3.1 are an average of that data. For critical applications, the sound power levels for a particular fan should be obtained from the manufacturer.

Sound power levels at actual operating conditions can be estimated by the actual fan volume flow rate and fan pressure, as;

$$L_W = K_W + 10\log\frac{Q}{Q_1} + 20\log\frac{P}{P_1} + C + BFI \tag{3. 2}$$

where;

L_W : estimated sound power level of the fan in dB re 10^{-12} W

K_W : specific sound power level in dB

Q : flow rate in m^3/s

Q_1 : 1 m^3/s

P: pressure drop in Pa

P_1 : 1 Pa

C: correction factor for point of fan operation in dB

BFI: blade frequency increment to be added only to octave band containing blade pass frequency in dB

Values of the estimated sound power level are calculated for all eight bands, and the BFI is added to the sound pressure level in the octave band in which the blade passage frequency falls.

Table 3.1 Specific Sound Power Levels of Typical Fans in dB [6]

		octave band center frequency, Hz								
Fan Type	**Wheel Size**	63	125	250	500	1000	2000	4000	8000	BFI
centrifugal										
Airfoil, backward-	over 900 mm	40 45	40	39	34	30	23	19	17	3
curved, backward-	under 900 mm		45	43	39	34	28	24	19	3
inclined										
Forward-curved	All	53	53	43	36	36	31	26	21	2
Radial blade and	low pressure (1 to 2.5 kPa)	56	47	43	39	37	32	29	26	7
pressure blower	Medium pressure (1.5 to 3.7 kPa)	58	54.	45	42	38	33	29	26	8
	High Pressure (3.7 to 15 kPa)	61	58	53	48	46	44	41	38	8
Vaneaxial	Hub ratio 0.3 to 0.4	49	43	43	48	47	45	38	34	6
	Hub ratio 0.4 to 0.6	49	43	46	43	41	36	30	28	6
	Hub ratio 0.6 to 0.8	53	52	51	51	49	47	43	40	6
Tubeaxial	over 1000 mm	51	46	47	49	47	46	39	37	7
	under 1000 mm	48	47	49	53	52	51	43	40	7
Propeller General Ventilation	All	48	51	58	56	55	52	46	42	5

Note : Includes total sound power level in dB for both inlet and outlet. Values are for fans only-not packaged equipment.

Table 3.2 Typical Octave Band in which Blade Frequency Increment (BFI) Occurs [6]

Fan Type	Octave Band in which BFI occurs*
Centrifugal	
Airfoil, backward-curved, backward-inclined	125-250 Hz
Forward-curved	500 Hz
Radial blade and pressure blower	125 Hz
Vaneaxial	125-250-500 Hz
Tubeaxial	63 Hz
Propeller	63 Hz

* Use for estimating purposes. For speeds of 1750 rpm (29 rps) or more, move the BFI to the next higher octave band. Where the actual fan is known, use the manufacturer's data.

Table 3.3 Correction Factor C for Off-Peak Operation [6]

Static efficiecy, % of peak	Correction factor,dB
90 to 100	0
85 to 89	3
75 to 84	6
65 to 74	9
55 to 64	12
50 to 54	15

3.1.1.1. Point of Operation

Fan selection at the calculated point of maximum efficiency is common practice to ensure minimum power consumption. In general, fan sound is at a minimum near the point of maximum efficiency. Noise increases as the operating point shifts to the right (higher airflow and lower static pressure) and low frequency noise can increase substantially at operating points to the left of maximum efficiency (lower airflow and higher static pressure) [5].

The specific sound power levels given in Table 3.1 are for fans operating at or near the peak efficiency point of the fan performance curves. This conforms with the recommended practice of selecting fan size and speed so that operation falls at or near this point; it is advantageous for energy conservation and corresponds to the lowest noise levels for that fan. If, for any reason, a fan is not or cannot be selected optimally, the noise level produced will increase. Hence, correction factor C in Equation (3. 2) must be applied to account for this increase. This correction factor should be applied to all octave bands. For off-peak operation, correction factor values are given in Table 3.3 [6].

3.1.1.2. General Discussion of Fan Noise

To minimize the required duct sound attenuation, the proper selection and installation of the fan (or fans) is vitally important. The following factors should be considered:

1. Design the air distribution system for minimum resistance, since the sound power generated by a fan, regardless of type, increases by the square of the static pressure.

2. Examine the specific sound power levels of the fan design for any given job. Different fans generate different levels of sound and produce different octave-band spectra. Select a fan that will generate the lowest possible sound level, commensurate with other fan selection parameters.

3. Fans with relatively fewer number of blades (less than 15) tend to generate tones, which may dominate the spectrum. These tones

occur at the blade passage frequency, i.e, Equation (3. 1) and its harmonics. The intensity of these tones depends on resonance within the duct system, fan design, and inlet flow distortions.

4. Design duct connections at both the fan inlet and outlet for uniform and straight airflow. Deviation from accepted applications can severely degrade both the aerodynamic and acoustic performance of any fan and invalidate manufacturer's ratings.

5. In variable air volume systems, consider the effect of changes in volume on the sound power. Reducing volume flow by changes in inlet vane settings may substantially increase the low frequency fan sound power levels of backwardly inclined and axial-flow fans [6].

3.1.2. VARIABLE AIR VOLUME (VAV) SYSTEM NOISE

VAV systems can significantly reduce energy cost due to their ability to modulate air capacity. However, they can be the source of fan noise that is very difficult to mitigate. To avoid these potential problems, the designer should carefully design the ductwork and the static pressure control systems, and select the fan or air handling unit and its modulation devices properly.

As in other aspects of HVAC design, the duct system should be designed for the lowest practical static pressure loss, especially in the ductwork closest to the fan or air handling unit. High airflow velocity and convoluted duct routing can cause airflow distortions that result in excessive pressure drop and fan instabilities that are responsible for excessive noise, fan stall, or both. Many VAV problems have been traced to improper air balancing. If the duct system is balanced with at least one balancing damper wide open, the fan speed could be reduced with a corresponding reduction in fan noise. Lower sound levels will result if most balancing dampers are wide open or eliminated.

For constant volume systems, fans should be selected to operate at maximum efficiency at the fan design airflow rate. However, VAV systems must be selected to operate with efficiency and stability throughout its range of modulation.

In general, the fan for a VAV system should be selected for peak efficiency at an operating point of around 70 to 80% of the maximum required system capacity. This usually means selecting a fan that is one size smaller than that required for peak efficiency at 100% of maximum required system capacity. When the smaller fan is operated at higher capacities, it will produce up to 5 dB more noise. This occasional increase in sound level is usually more tolerable than the stall related sound problems that can occur with a larger fan operating at less than 100% design capacity most of the time.

3.1.2.1. Air Modulation Devices

Variable capacity control methods can be divided into three general categories:

1. Variable inlet vanes or discharge dampers, which yield a new fan system curve at each vane or damper setting;

2. Variable pitch fan blades (usually used on in-line axial fans), which adjust the blade angle for optimum efficiency at varying capacity requirements; and

3. Variable speed motor drives where the motor speed is varied by modulation of the power line frequency or by mechanical means such as gears or continuous belt adjustment

3.1.2.1.1. Variable Inlet Vanes and Discharge Dampers

Variable inlet vanes vary airflow capacity by changing the inlet airflow to a fan wheel. This type of modulation varies the total air volume and pressure at the fan while the fan speed remains constant. While fan pressure and air volume reductions at the fan result in duct system noise reductions by reduced air velocity and pressures in the duct work, there is an associated increase in fan noise caused by the air-flow turbulence and flow distortions at the inlet vanes acting as a fan inlet obstruction.

Fan manufacturers' test data have shown that, on airfoil type centrifugal fans, as vanes mounted inside the fan inlet (nested inlet vanes) close, the sound level at the blade passing frequency of the fan increases by 2 to 8 dB, depending on the amount of total air volume restricted. For inlet vanes that are mounted externally the increase is in the order of 2 to 3 dB. Forward curved fan wheels with inlet vanes are about 1 to 2 dB quieter than airfoil fan wheels. In line axial type fans with inlet vanes generate increased noise levels of 2 to 8 dB in the low frequency octave bands for a 25% to 50% closed vane position.

Discharge dampers are typically located immediately downstream of the supply air fan and reduce airflow and increase pressure drop across the fan while the fan speed remains constant. Because of the air turbulence and flow distortions created by the high pressure drop across discharge dampers, there is a high probability that duct rumble will occur near the damper location. If the dampers are throttled to a very low flow, a stall condition can occur at the fan also resulting in an increase in low frequency noise.

3.1.2.1.2. Variable Pitch Fan Blades for Capacity Control

Variable pitch fan blade controls vary the fan blade angle in order to reduce the overall airflow through the fan. This type of capacity control system is predominantly used in axial type fans. As air volume and pressure are reduced at the fan, the corresponding noise reduction is usually 2 to 5 dB in the 125 through 4000 Hz octave bands for an 80% to 40% air volume reduction.

3.1.2.1.3. Variable Speed Motor Controlled Fan

Three types of electronic variable speed control units are used with fans; (1) current source inverter, (2) voltage source inverter, (3) pulse width modulation (PWM). The current source inverter and third generation PWM control units are usually the quietest of these controls. The primary acoustic advantage of a variable speed controlled fan is the reduction of fan speed, which translates into reduced noise where dB reduction is approximately $50[\log(higherspeed / lowerspeed)]$. Because this speed reduction generally follows the fan system curve, a fan selected at optimum efficiency initially (lowest noise) does not lose that efficiency as the speed is reduced.

3.1.3. ROOFTOP MOUNTED AIR HANDLER NOISE

Large roof openings are often required for supply and return air duct connections. These ducts run directly from noise-generating rooftop air handlers to the building interior. The four common sound transmission paths associated with rooftop air handlers are

1. Airborne through the bottom of the rooftop unit to spaces below

2. Structure-born from vibrating equipment in the rooftop unit to the building structure

3. Duct-borne through the supply air duct from the air handler.

4. Duct-borne through the return air duct from the air handler.

The sound radiated from the duct transmission paths is of interest in the scope of this study.

Duct-borne transmission of sound through the supply air duct consists of two components; sound transmitted from the air handler through the supply air duct system to occupied areas and sound transmitted via duct breakout through a section or sections of the supply air duct close to the air handler to occupied areas. Experience has indicated that sound transmission below 250 Hz via duct breakout is often a major acoustical limitation for many rooftop installations. Excessive low frequency noise associated with fan noise and air turbulence in the region of the discharge section of the fan and the first duct elbow results in duct rumble, which is difficult to attenuate. This problem is often made worse by the presence of a duct with a high aspect ratio at the discharge section of the fan.

Rectangular ducts with duct lagging are often ineffective in reducing duct breakout noise. Using either a single or dual wall round duct with a radiused elbow coming off the discharge section of the fan can control duct breakout. If space does not allow for the use of a single duct, the duct can be split into several parallel round ducts. Another method that is effective is the use of an acoustic plenum chamber constructed with a minimum 50 mm thick, dual wall plenum panel, which is lined with fiberglass and which has a perforated inner liner at the discharge section of the fan. Either round or rectangular ducts can be taken off the plenum as necessary to supply the rest of the air distribution system.

Duct-borne transmission of sound through the return air duct of a rooftop unit is often is a problem. Generally only one short return air duct section runs from the plenum space above a ceiling and the return air section of the air handler. This short run does not adequately attenuate the sound between the fan inlet and the spaces below the air handler. The sound attenuation through the return air duct can be improved by adding at least one (more if possible) branch division where the return air duct is split into two sections that extend several duct diameters before they terminate into the plenum space above the ceiling. The inside surfaces of all the return air ducts should be lined with a minimum of 25 mm thick duct liner. If conditions permit, duct silencers in the duct branches or an acoustic plenum chamber at the air handler inlet give better sound conditions [5].

3.1.4. CHILLER AND AIR COOLED CONDENSER NOISE

All chillers and their associated equipment produce significant amounts of both broadband and tonal noise. The broadband noise is due to flows of both refrigerant and water. While the tonal noise is caused by the rotation of compressors, motors and fans (in fan cooled equipment). Chiller noise is usually significant in the octave bands from 250 through 1000 Hz.

In most water-cooled indoor chillers, the compressor is the dominant noise source. For a given chiller at a given operating point, a small equipment room, or one with mostly hard surface finishes, has a higher L_p value than a room that is large or has sound absorbing treatments on its ceiling and walls. The approximate reverberant L_p values can be used along with the sound transmission loss information to estimate the transmitted L_p values in rooms adjacent to a chiller room. Indoor chillers are offered with various types of factory noise reduction options ranging from compressor blankets (2 to 6 dBA reduction), to steel panel enclosures with sound absorbing inner surfaces (up to 18 dBA reduction).

Most air-cooled chillers use either reciprocating, scroll, or screw compressors. These chillers are also used as the chiller portion of rooftop packaged units. The dominant noise sources in outdoor air-cooled chillers are compressors and the condenser fans, which are typically low-cost, high-speed propeller fans. The determination of the factory sound data for outdoor equipment requires the A-weighted and octave band sound power level value of the equipment be determined. Factory supplied noise reduction options for outdoor equipment include compressor enclosures, oversized condenser fans, and variable speed condenser fans. Because air cooled equipment needs a free flow of cooling air, full enclosures are not feasible; however, strategically placed barriers can help reduce the noise propagation on a selective basis [5].

3.1.5. AIRFLOW NOISE

3.1.5.1. Aerodynamically Generated Sound in Ducts

Although fans are a major source of sound in HVAC systems, they are not the only sound source. Aerodynamic sound is generated at duct elbows, dampers, branch takeoffs, air modulation units, sound attenuators, and other duct elements. Produced by the interaction of moving air with the structure, the sound power levels in each octave frequency band depend on the duct element geometry and the turbulence of the air flow and the air flow velocity in the vicinity of the duct element. Duct related aerodynamic noise problems can be avoided by

1. Sizing ductwork or duct configurations so that air velocity is kept low,

2. Avoiding abrupt changes in duct cross section area,

3. Providing smooth transitions at duct branches, takeoffs, and bends,

4. Attenuating sound generated at duct fittings with sufficient sound attenuation elements between a fitting and corresponding air terminal device.

Elbows and other fittings can increase airflow noise substantially, depending on the type. Thus, duct airflow velocities should be reduced accordingly. The presence of diffusers or grilles can increase sound levels a little or a lot, depending on how many diffusers or grilles are installed and on their design, construction, installation etc. Thus, allowable outlet or opening airflow velocities should be reduced accordingly.

The amplitude of aerodynamically generated sound in ducts is generally proportional to between fifth and sixth power of the duct airflow velocity in the vicinity of a duct fitting. So reducing duct airflow velocity significantly reduces flow generated noise [5].

Sizing duct work for low velocities is expensive and space consuming. In practice, space is often insufficient to allow smooth air flow, area and direction change abruptly, and duct lengths between fittings and terminal devices are not ideal. Because duct elements can have an unlimited number of geometry, a universal law for aerodynamic noise generation has not been developed [6].

3.1.6. DAMPER, ELBOW, AND JUNCTION NOISE

Octave band sound power levels generated at single and multi-blade dampers, at elbows, with and without turning vanes, and at junctions can be predicted by following this generalized equation [6]:

$$PWL_{f_0} = K + 10\log f_0 + 50\log U + 10\log S + 10\log D + \text{Special Parameters} \quad (3.\ 3)$$

where;

f_0 : octave band center frequency in Hz

K: characteristic spectrum of the fitting, based on Strouhal Number (Figure 3.1 and Figure 3.2)

U: velocity factor; the velocity in the constricted part of the flow field or the velocity in the branch duct in m/s

S: cross-sectional area of the duct in which the dampers are installed, cross-sectional are of the elbow or cross-sectional are of the branch duct at a junction in m^2

D: duct height normal to the damper axis, the cord length of a typical turning vane, height of the elbow, or in the case of junctions, $D = \sqrt{4S/\pi}$ in m

Special parameters have the following values:

- Dampers: -18 dB
- Elbows with turning vanes: 10logn-18 dB

where;

n: number of turning vanes.

- For elbows without turning vanes and junctions: $18 + \Delta r + \Delta T$

where;

Δr : correction for rounding
ΔT : correction for upstream turbulence

Prior to solving Equation (3. 3), preliminary calculations stated in 3.1.6.1 and 3.1.6.2 are to be performed.

3.1.6.1. Preliminary Calculations for Dampers and Elbows with Turning Vanes [6]

1. Determine pressure loss coefficient C

$$C = 1.67 \Delta p S^2 / Q^2 \qquad (3.\ 4)$$

where;

Δp :pressure drop across fitting in Pa

Q: volume flow rate in m^3/s

2. Determine blockage factor BF

- For multiple dampers and elbows with turning vanes:

BF = 0.5 if C = 1

$$BF = (C^{05} - 1)/(C - 1), \text{ for } C \neq 1 \tag{3. 5}$$

- For single blade dampers:

$$BF = (C^{0.5} - 1)/(C - 1), \text{ if C<4}$$

$$BF = 0.68C^{-0.15} - 0.22, \text{ if C>4} \tag{3. 6}$$

3. Determine velocity term U

$$U = (Q/S)/BF \tag{3. 7}$$

4. Determine Strouhal number St

$$St = f_0 D/U \tag{3. 8}$$

5. Enter Figure 3.1 to determine the characteristic spectrum for dampers.

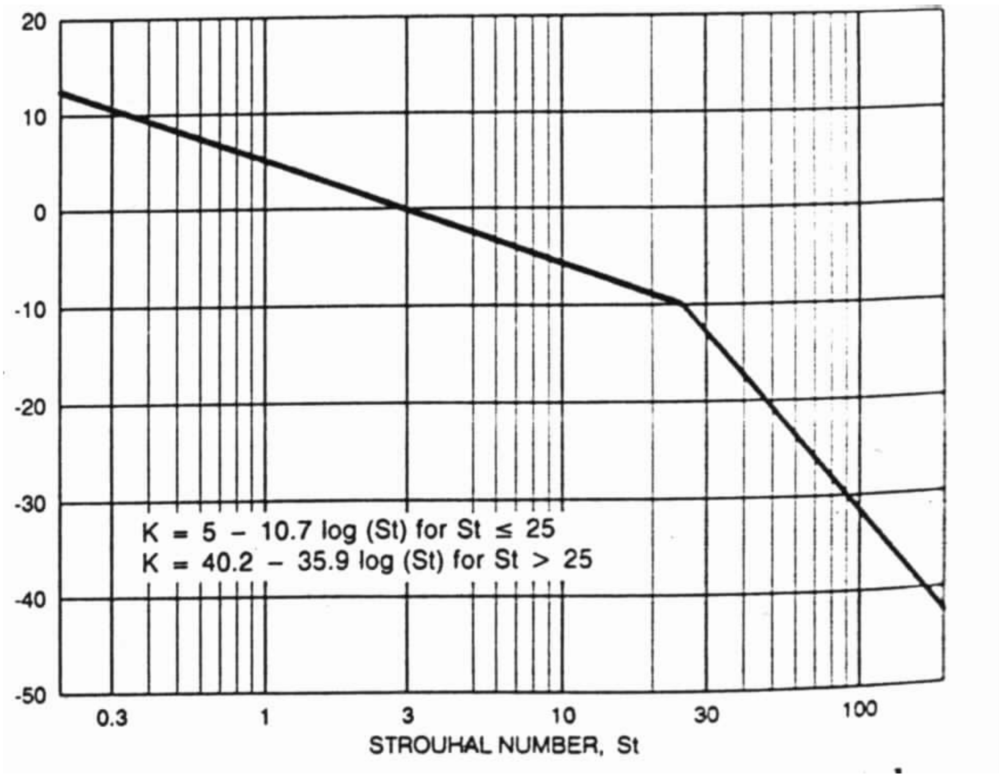


Figure 3.1 Characteristic Spectrum of K of Flow-Generated Noise of Dampers for use with Equation (3. 3) [6]

Enter Figure 3.2 to determine the characteristic spectrum for elbows with turning vanes.

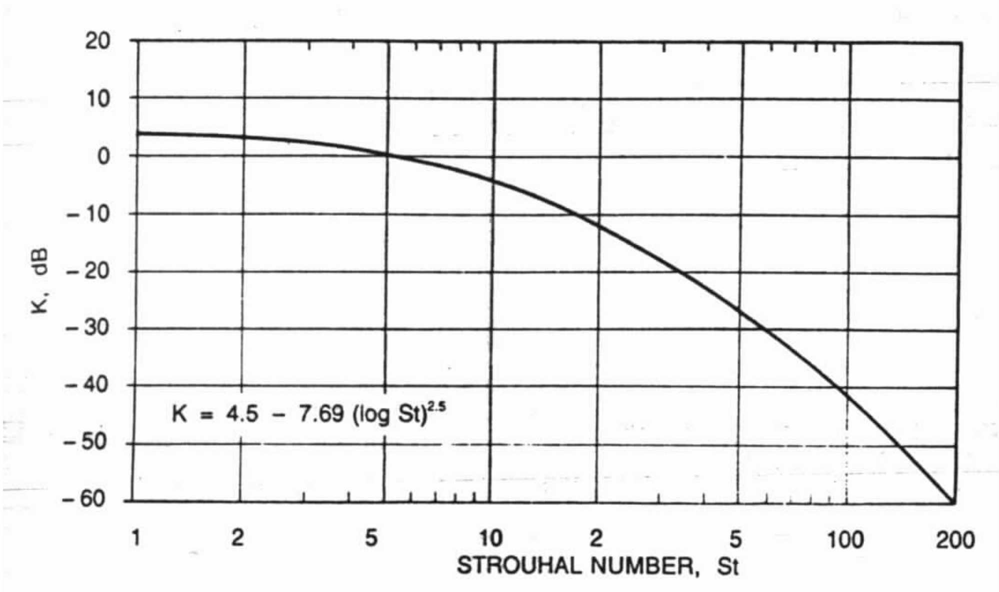


Figure 3.2 Characteristic Spectrum of K of Flow-Generated Noise for Bends Fitted with Curved-Blade Turning Vanes [6]

3.1.6.2. Preliminary Calculations for Elbows and Junctions without Splitter Dampers [6]

1. Determine velocity factor $M = U_M / U_B$ (3. 9)

where;

U_M : velocity in main duct in m/s

U_B : velocity in branch duct in m/s

2. Determine Strouhal number $St = f_0 D / U_B$ (3. 10)

3. Enter Figure 3.3 to determine the characteristic spectrum K.

4. Determine the rounding correction from Figure 3.4. If the elbow is not rounded, then:

$$r / D_{BE} = 0$$

5. Determine the correction for upstream turbulence T from Figure 3.5. This correction should be applied only if duct upstream has dampers, turns, or take offs within a length of 5 main duct diameters.

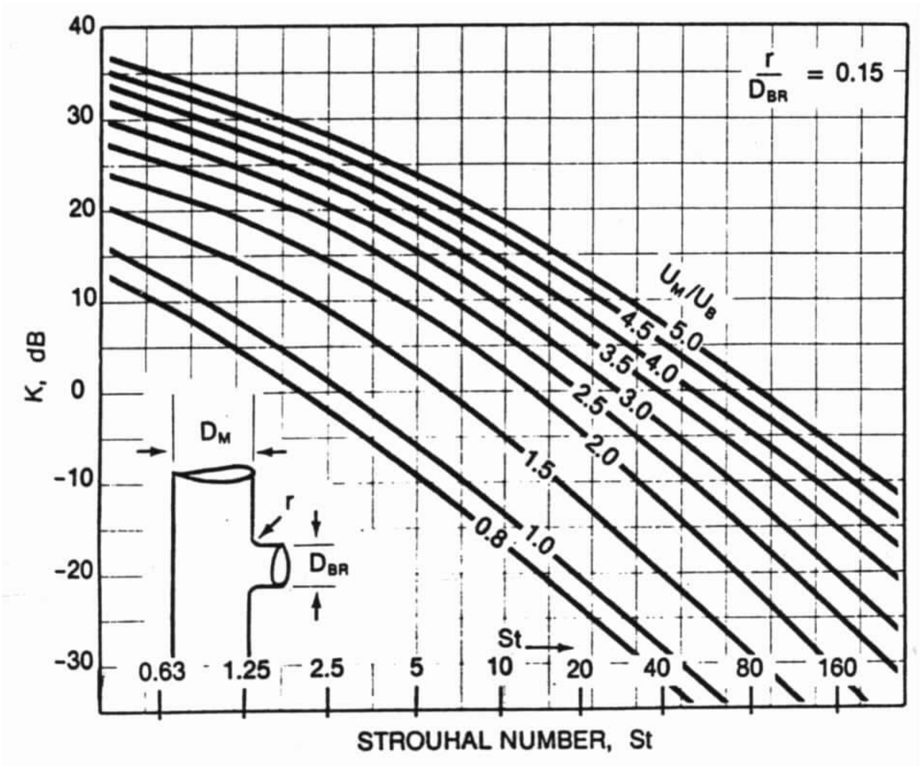


Figure 3.3 Normalized Spectra for Flow-Generated Noise of Takeoffs and Junctions without Splitter Damper or Scoop [6]

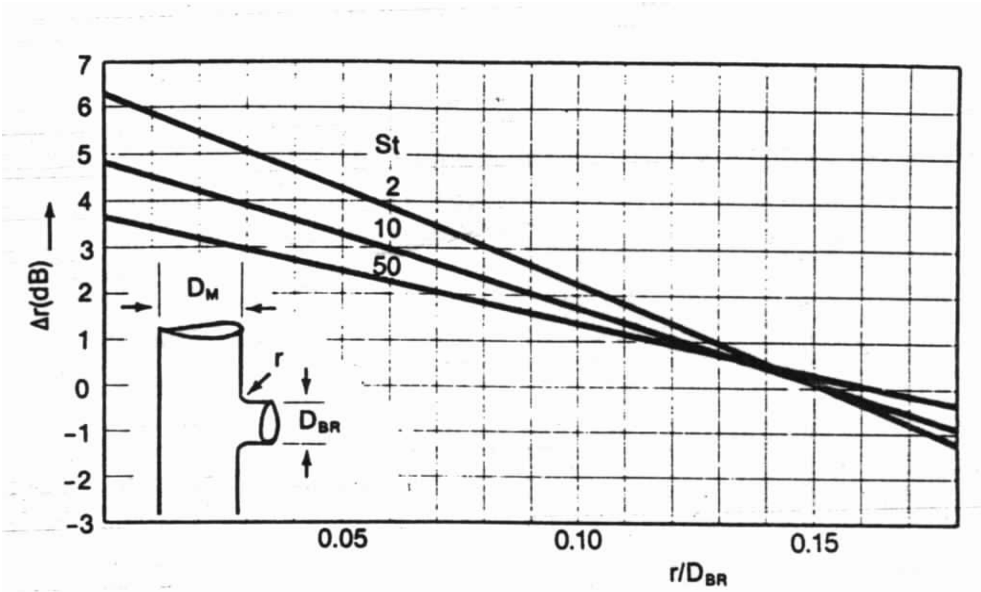


Figure 3.4 Rounding Correction [6]

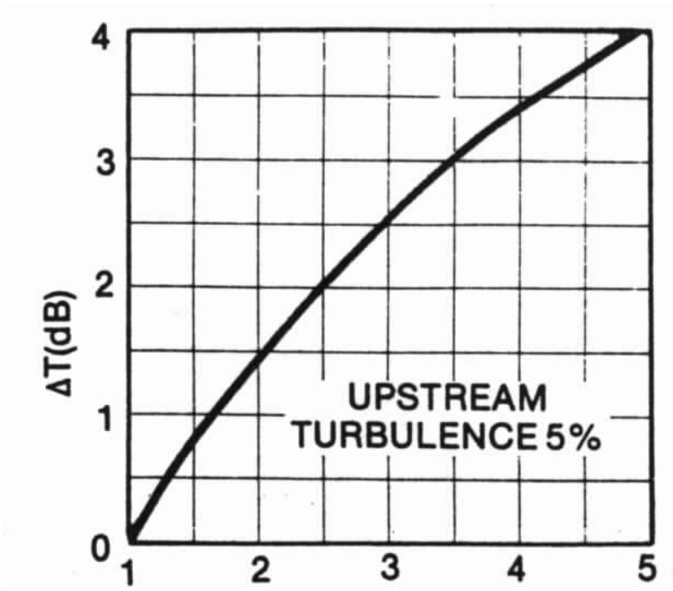


Figure 3.5 Correction for Upstream Turbulence [6]

3.1.7. DAMPER – DUCT TERMINAL DEVICE NOISE

Depending on its location relative to a duct terminal device, a damper can generate unwanted noise into an occupied area of a building. The noise can be transmitted down the duct to the discharge, or through the ceiling space into the occupied space below. Volume dampers should not be placed closer than 1.5 m from an air outlet for good design. When a volume control damper is installed close to an air outlet, the acoustic performance of the air outlet must be based on the air volume handled and on the pressure drop across the damper. The sound level produced by the damper is accounted for by adding a quantity to the diffuser sound rating. This quantity is proportional to the pressure ratio, which is the throttled pressure drop across the damper. Table 3.4 and Figure 3.6 provide quantities to determine the effect of damper location on diffuser sound ratings.

Table 3.4 Decibels to be Added to Diffuser Sound Rating to Allow for Throttling of Volume Damper [6]

Pressure Ratio (PR) = $\frac{Throttled\ \mathrm{Pr}essure}{Minimum\ \mathrm{Pr}essure}$						
Location of volume damper in m from air outlet	1.5	2	2.5	3	4	6
(A) In neck of linear diffuser	5	9	12	15	18	24
(B) In inlet of plenum of linear diffuser	2	3	4	5	6	9
(C) In supply duct at least 1.5 m from inlet plenum of linear diffuser	0	0	0	2	3	5

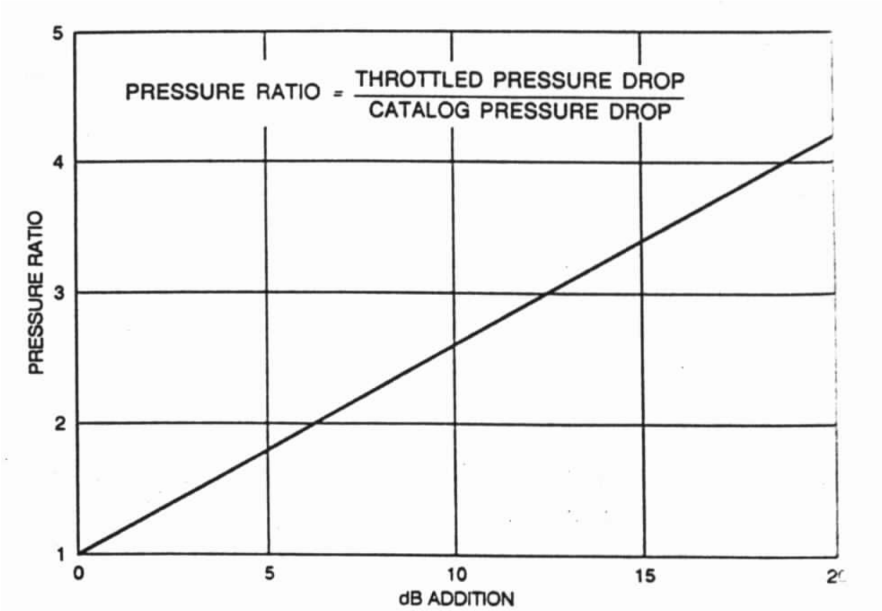


Figure 3.6 Decibels to be Added to Outlet Sound for Throttled Damper Close to Outlet [6]

Balancing dampers, equalizers, and other similar devices should not be placed directly upstream of air devices or open ended ducts in acoustically critical spaces. They should be located 5 to 10 duct diameters from the termination device with acoustically lined duct joining the damper and duct termination device.

Plenums may be used to keep dampers further away from diffusers. Dampers may be installed at the plenum entrance with linear diffusers installed in the distribution plenum. The further a damper is installed from the outlet, the lower the resultant sound level [5].

3.1.8. AIR TERMINAL NOISE

Air terminal devices can be assorted as grilles, registers, diffusers, air handling light fixtures, and air handling suspension bars. The room duct termination device should be selected to meet the noise criterion required or specified for the room. If a duct turn precedes the entrance to the diffuser or if a balancing damper is installed immediately before the diffuser, the airflow will be turbulent and the noise generated by the device will be substantially higher than the manufacturer's data by as much as 12 dB. In some cases, placing an equalizer grid in the neck of the diffuser reduces this turbulence substantially. The equalizer grid can help provide a uniform velocity gradient in the neck of the diffuser so the sound power generated in the field is closer to that listed in the manufacturer's catalog.

At present, diffusers are rated in terms of noise criterion (NC) levels, which include a receiver room sound correction of usually 8 to 10 dB. The ratings may be useful for comparison between and among different diffusers, but are not helpful for design. The designer should request from the diffuser manufacturer the component sound power level data in octave bands which are tested in accordance with ASHRAE 130 (for Air Terminals) and ASHRAE 70 (for Air Outlets) [15], and use the sound power to estimate the effects of the diffusers on the sound levels in different spaces [5].

Air terminals-devices used to deliver air into a space to control air volume and/or temperature by directional vanes, dampers or valves, heat exchangers, and fan control fall into two categories. The first category, air terminal devices, comprises diffusers, grilles, and registers that radiate sound directly into an air-conditioned space. Allowing no opportunity for sound attenuation along the duct path, air terminal devices require selection for noise generation that meets the established room design criterion values as well as other factors. The second category, terminal vanes, comprises terminal boxes and air valves that are separated from the space they serve by ductwork, ceiling plenums, or ceilings. These devices are selected so that their sound power ratings do not exceed the sum of the established room design criterion and the attenuation provided between the air-conditioned room and the device.

While fan generated noise is most critical in the low frequency bands, many air terminals such as diffusers, registers, grilles, and terminal boxes without fans do not contribute greatly to low frequency noise levels. In these cases, an allowance for fan noise may be unnecessary in selecting the terminals. However, both fan and terminal unit noise must be considered to obtain the balanced sound spectrum implied by the RC curves.

The sound level output of an air diffuser or grille depends not only on the air quantity and the size and design of the outlets, but also on the air approach configuration. Manufacturer's rating apply only to outlets installed as recommended i.e., with a uniform air velocity distribution throughout the neck of the unit. Poor approach conditions can easily increase sound levels by 12 to 15 dB above the manufacturer's ratings. Poor approach conditions can sometimes be overcome with properly adjusted accessories such as turning vanes or equalizing grids. However, using these accessories in critical, low noise level projects should be avoided.

Flexible ducts are often used to correct misalignment between the supply duct and the diffuser ceiling location. A misalignment for offset that exceeds one-fourth of a diffuser diameter in a diffuser collar length of 2 diameters significantly increases the diffuser sound level. There is no appreciable chance in diffuser performance with an offset less than one-eight the length of the collar.

When a volume control damper is installed close to an air out-let to achieve system balance, the acoustic performance of the air outlet must be based not only on the air volume handled, but also on the magnitude of the pressure drop across the damper. The sound level change is proportional to the pressure ratio

(PR) of the throttled pressure drop to the catalog pressure drop of the outlet as depicted in 3.1.7.

Balancing dampers, equalizers, and so forth should not be placed directly behind terminal devices or open-ended ducts in acoustically critical spaces, such as concert halls. They should be located 5 to 10 duct diameters from the opening, followed by lined duct to the terminal or open duct end [6].

Linear diffusers are often installed in distribution plenums so that the damper may be installed at the plenum entrance. The further a damper is installed from the outlet, the lower the resultant sound level will be (Table 3.4).

3.1.9. FAN POWERED TERMINAL VALVE NOISE

A fan in the terminal valve unit draws in heated air from the ceiling plenum which mixes with the primary air to control the delivered air temperature. In addition to the potential acoustic problems, these devices can transmit fan noise into the occupied space by radiating noise into the ceiling plenum as well as along the duct path.

3.2. PRIMARY SOUND ATTENUATORS

Sound attenuators which are explained in details in this section are classified basically as duct element sound attenuation, sound radiation through duct walls, sound transmission through ceilings, return air systems and receiver sound room correction. All these headings cover sub topics related to more specific HVAC elements which have attenuation features.

3.2.1. DUCT ELEMENT SOUND ATTENUATION

The duct elements covered hereby include sound plenums, unlined rectangular ducts, unlined round ducts, acoustically lined round ducts, elbows, acoustically lined round radiused elbows, duct silencers, duct branch power division, duct and reflection loss, and terminal volume regulation units. Procedures for obtaining the sound attenuation associated with these elements are presented in this section.

3.2.1.1. Natural Attenuation in Ducts

Even if ductwork contains no sound attenuators (acoustical linings or sound traps), only a fraction of the acoustic energy generated by the fan, the duct fittings, and so forth reaches any one room because of, firstly, the combined effects of energy division at branch takeoffs and, secondly, energy losses because of duct wall vibration and sound reflections at elbows and other outlets.

To avoid over-designing the acoustic duct treatment of the duct system, natural attenuation should be considered. The natural attenuation for rectangular duct and circular ducts without internal insulation is given in Table 3.5 and Table 3.6 [6].

Table 3.5 Natural Sound Attenuation in Unlined Rectangular Sheet Metal Ducts [6]

	Octave band center frequency, Hz		
P/A ratio, a*	63	125	250 and over
mm/mm^2	Attenuation, dB/m b*		
Over 0.012	0	0.98	0.33
0.012 to 0.005	0.98	0.33	0.33
Under 0.005	0.33	0.33	0.33

a perimeter divided by area

b double this values if the duct is externally insulated

Table 3.6 Natural Sound Attenuation in Unlined Straight Round Ducts [6]

Diameter mm	Approximate attenuation, dB/m for 25 mm duct liner *						
	Octave band center frequency, Hz						
	63	125	250	500	1000	2000	4000
150	0.66	1.6	3.3	5.9	7.2	7.2	6.6
300	0.49	0.98	2.3	4.9	7.2	7.2	4.9
600	0.33	0.66	1.6	3.3	5.6	3.0	1.6
1200	0.13	0.33	0.1	2	2	2.6	1.6

* Test data based on 0.55 to 0.85 mm (26 to 22 gage) spiral wound duct with perforated spiral wound steel liner 7.2 m long. Diameter shown is free area. Data are for no airflow circumstances.

3.2.1.2. Plenums

Plenums are often used to smooth the turbulent airflow associated with air as it leaves the outlet section of a fan and before it enters the air distribution ducts. These chambers are usually lined with acoustically absorbent material to reduce fan and other noise. Plenums are usually large rectangular enclosures with an inlet and one or more outlets. The transmission loss associated with a plenum can be expressed as;

$$TL = -10\log_{10}\left[S_{out}\left(\frac{Q\cos\theta}{4\pi r^2} + \frac{1-\alpha_A}{s\alpha_A}\right)\right] \tag{3. 11}$$

where (reference to Figure 3.7);

S_{out} : area of output section of plenum in m^2

s: total inside surface area of plenum minus inlet and outlet areas in m^2

r: distance between centers of inlet and outlet sections of plenum in m

Q: directivity factor, which may be taken as 4

α_A : average absorption coefficient of the plenum lining

θ: angle of vector representing the r to along axis l of duct (see Equation (3. 13))

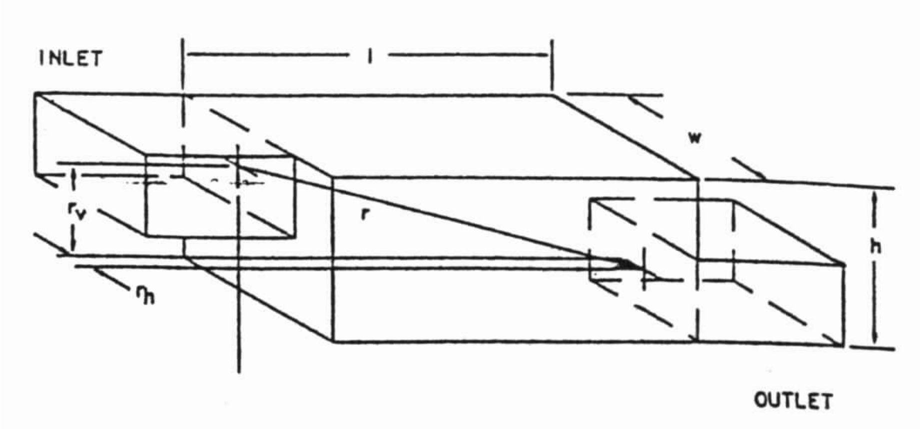


Figure 3.7 Schematic of Plenum Chamber [5]

The average absorption coefficient α_A of plenum lining is

$$\alpha_A = \frac{S_1\alpha_1 + S_2\alpha_2}{S} \tag{3. 12}$$

where;

α_1 : sound absorption coefficient of any bare or unlined inside surfaces of plenum

S_1 : surface area of any bare or unlined inside surfaces of plenum in m^2

α_2 : sound absorption coefficient of acoustically lined inside surfaces of plenum

S_2 : surface area of acoustically lined inside surfaces of plenum in m^2

In many situations, all of the inside surfaces of a plenum chamber are lined with a sound absorbing material. For these situations $\alpha_A = \alpha_2$.

The value of cosθ is obtained from

$$\cos\theta = \frac{l}{r} = \frac{l}{\sqrt{l^2 + r_v^{\ 2} + r_h^{\ 2}}} \tag{3. 13}$$

where (refer to Figure 3.7);

l: length of plenum in m

r_v : vertical offset between axes of plenum inlet and outlet in m

r_h : horizontal offset between axes of plenum inlet and outlet in m

Equation (3. 11) treats a plenum as if it is a large enclosure. Thus, Equation (3. 11) is valid only for the case where the wavelength of sound is small compared to the characteristic dimensions of the plenum. For frequencies that correspond to plane wave propagation in the duct, the results predicted by Equation (3. 11) are usually not valid. Plane wave propagation in a duct exists at frequencies below;

$$f_{co} = \frac{c_o}{2a} \tag{3. 14}$$

or

$$f_{co} = 0.586\frac{c_o}{d} \tag{3. 15}$$

where;

f_{co} : cutoff frequency in Hz

c_o : speed of sound in air in m/s

a: larger cross-section dimensions of a rectangular duct in m

d: diameter of a round duct in m

The cut-off frequency f_{co} is the frequency above which plane waves no longer propagate in a duct. At these higher frequencies the waves that propagate in the duct are referred to as cross or spinning modes. At the frequencies below f_{co}, Equation (3. 11) yields conservative results. The actual attenuation usually exceeds the values given by Equation (3. 11) by 5 to 10 dB. Equation (3. 11) usually applies at frequencies of 1000 Hz and higher [5].

3.2.1.3. Unlined Rectangular Sheet Metal Ducts

Straight unlined rectangular sheet metal ducts provide a fairly significant amount of low frequency sound attenuation. Table 3.5 shows the results of selected unlined rectangular sheet metal ducts. The attenuation values shown in Table 3.5 apply only to rectangular sheet metal ducts with the lightest gages allowed according to SMACNA duct construction standards. Sound attenuation at low frequencies in rectangular ducts may manifest itself as breakout noise elsewhere along the duct. Low-frequency breakout noise should therefore be checked [5]. Beakout noise in duct elements is elaborated in section 3.2.2.3.

3.2.1.4. Acoustically Lined Rectangular Sheet Metal Ducts

Internal duct lining for rectangular sheet metal ducts can be used to attenuate sound in ducts and to thermally insulate ducts. For fiberglass duct lining to be effective for attenuating fan sound, it must have a minimum thickness of 25 mm. The density of the fiberglass lining used in lined rectangular sheet metal ducts usually varies between 24 and 48 kg/m3 [5]. The sound absorption of such relatively thin linings is limited, especially at low frequencies below about 250 Hz [6].

As sound travels down a duct, some acoustic energy is absorbed by the duct or its lining, or it is radiated by the duct walls. The result is that the acoustic energy at the end of a section of duct is less than at the entrance. The factors for determining the loss of acoustic energy are dependent on the lining, if any, and the type and geometry [15]. The insertion loss values are the difference in the sound pressure level measured in a reverberation room with sound propagating through an unlined section of rectangular duct minus the corresponding sound pressure level that is measured when the unlined section of rectangular duct is replaced with a similar section of acoustically lined rectangular duct. The attenuation of the unlined duct is subtracted out during the process of calculating the insertion loss from the measured data. Insertion loss and attenuation values discussed in this section apply only to rectangular sheet metal ducts made with the lightest gages allowed according to the SMACNA duct construction standards [5].

Equation (3. 16) was developed to determine the attenuation provided by the acoustical lining in lined rectangular or square ducts. Equation (3. 16) may be used to calculate the total attenuation or insertion loss for any duct length and lining dimensions [15]. Coefficients to be used in Equation (3. 16) are given in Table 3.7.

$$InsertionLoss / Attenuation = 10^{CoeffA}.(P / A)^{CoeffB}.t^{CoeffC} \qquad (3.\ 16)$$

where;

P/A: Perimeter/Area in 1/ft

t: material thickness in in

Table 3.7 Coefficients to be used in Insertion Loss/Attenuation Calculations for Lined Rectangular Ducts [15]

Coefficients	Coefficients for Equation (3. 16)						
	Octave band center frequency, Hz						
	125	250	500	1000	2000	4000	8000
Coeff A	-0.865	-0.582	-0.0121	0.298	0.089	0.0649	0.15
Coeff B	0.723	0.826	0.487	0.513	0.862	0.629	0.166
Coeff C	0.375	0.975	0.868	0.317	0	0	0

3.2.1.5. Unlined Round Sheet Metal Ducts

As with unlined rectangular ducts, unlined rectangular ducts provide some natural sound attenuation that should be considered when designing a duct system. In contrast to rectangular ducts, round ducts are much more rigid and, therefore, do not resonate or absorb as much sound energy. Because of this, round ducts will only provide about 1/10 the sound attenuation at low frequencies as compared to the sound attenuation associated with rectangular ducts. Sound attenuation values for unlined round circular ducts are listed in Table 3.8 [5].

Table 3.8 Sound Attenuation in Straight Round Ducts [5]

	Attenuation, dB/m						
	Octave band center frequency, Hz						
Diameter, mm	63	125	250	500	1000	2000	4000
$D \leq 180$	0.03	0.03	0.05	0.05	0.10	0.10	0.10
$180 < D \leq 380$	0.03	0.03	0.03	0.05	0.07	0.07	0.07
$380 < D \leq 760$	0.02	0.02	0.02	0.03	0.05	0.05	0.05
$760 < D \leq 1520$	0.01	0.01	0.01	0.02	0.02	0.02	0.02

3.2.1.6. Acoustically Lined Round Sheet Metal Ducts

The literature has little data for the insertion loss of acoustically lined round ducts. The data that are available are usually manufacturer's product data. Table 3.9 and Table 3.10 give the insertion loss values for dual-wall round sheet metal ducts with 25 mm and 50 mm acoustical lining, respectively. The acoustical lining for the ducts is a 12 kg/m3 density fiberglass blanket, and the fiberglass is covered with an internal liner of perforated galvanized sheet metal that has an open area of 25% [5].

Table 3.9 Insertion Loss for Acoustically Lined Round Ducts with 25 mm Lining [5]

Diameter mm	Insertion Loss, dB/m							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
150	0.38	0.59	0.93	1.53	2.17	2.31	2.04	1.26
205	0.32	0.54	0.89	1.50	2.19	2.17	1.83	1.18
255	0.27	0.50	0.85	1.48	2.20	2.04	1.64	1.12
305	0.23	0.46	0.81	1.45	2.18	1.91	1.48	1.05
355	0.19	0.42	0.77	1.43	2.14	1.79	1.34	1.00
405	0.16	0.38	0.73	1.40	2.08	1.67	1.21	0.95
460	0.13	0.35	0.69	1.37	2.01	1.56	1.10	0.90
510	0.11	0.31	0.65	1.34	1.92	1.45	1.00	0.87
560	0.08	0.28	0.61	1.31	1.82	1.34	0.92	0.83
610	0.07	0.25	0.57	1.28	1.71	1.24	0.85	0.80
660	0.05	0.22	0.53	1.24	1.59	1.14	0.79	0.77
710	0.03	0.19	0.49	1.20	1.46	1.04	0.74	0.74
760	0.02	0.16	0.45	1.16	1.33	0.95	0.69	0.71
815	0.01	0.14	0.42	1.12	1.20	0.87	0.66	0.69
865	0	0.11	0.38	1.07	1.07	0.79	0.63	0.66
915	0	0.08	0.35	1.02	0.93	0.71	0.60	0.64
965	0	0.06	0.31	0.96	0.80	0.64	0.58	0.61
1015	0	0.03	0.28	0.91	0.68	0.57	0.55	0.58
1070	0	0.01	0.25	0.84	0.56	0.50	0.53	0.55
1120	0	0	0.23	0.78	0.45	0.44	0.51	0.52
1170	0	0	0.20	0.71	0.35	0.39	0.48	0.48

Table 3.9 (cont.)

Diameter mm	Insertion Loss, dB/m							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
1220	0	0	0.18	0.63	0.26	0.34	0.45	0.44
1270	0	0	0.15	0.55	0.19	0.29	0.41	0.40
1320	0	0	0.14	0.46	0.13	0.25	0.37	0.34
1370	0	0	0.12	0.37	0.09	0.22	0.31	0.29
1420	0	0	0.10	0.28	0.08	0.18	0.25	0.22
1475	0	0	0.09	0.17	0.08	0.16	0.18	0.15
1525	0	0	0.08	0.06	0.10	0.14	0.09	0.07

Table 3.10 Insertion Loss for Acoustically Lined Round Ducts with 50 mm Lining [5]

Diameter mm	Insertion Loss, dB/m							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
150	0.56	0.80	1.37	2.25	2.17	2.31	2.04	1.26
205	0.51	0.75	1.33	2.23	2.19	2.17	1.83	1.18
255	0.46	0.71	1.29	2.20	2.20	2.04	1.64	1.12
305	0.42	0.67	1.25	2.18	2.18	1.91	1.48	1.05
355	0.38	0.63	1.21	2.15	2.14	1.79	1.34	1.00
405	0.35	0.59	1.17	2.12	2.08	1.67	1.21	0.95
460	0.32	0.56	1.13	2.10	2.01	1.56	1.10	0.90
510	0.29	0.52	1.09	2.07	1.92	1.45	1.00	0.87
560	0.27	0.49	1.05	2.03	1.82	1.34	0.92	0.83
610	0.25	0.46	1.01	2.00	1.71	1.24	0.85	0.80
660	0.24	0.43	0.97	1.96	1.59	1.14	0.79	0.77
710	0.22	0.40	0.93	1.93	1.46	1.04	0.74	0.74
760	0.21	0.37	0.90	1.88	1.33	0.95	0.69	0.71
815	0.20	0.34	0.86	1.84	1.20	0.87	0.66	0.69
865	0.19	0.32	0.82	1.79	1.07	0.79	0.63	0.66
915	0.18	0.29	0.79	1.74	0.93	0.71	0.60	0.64
965	0.17	0.27	0.76	1.69	0.80	0.64	0.58	0.61
1015	0.16	0.24	0.73	1.63	0.68	0.57	0.55	0.58
1070	0.15	0.22	0.70	1.57	0.56	0.50	0.53	0.55
1120	0.13	0.20	0.67	1.50	0.45	0.44	0.51	0.52
1170	0.12	0.17	0.64	1.43	0.35	0.39	0.48	0.48

Table 3.10 (cont.)

Diameter mm	Insertion Loss, dB/m							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
1220	0.11	0.15	0.62	1.36	0.26	0.34	0.45	0.44
1270	0.09	0.12	0.60	1.28	0.19	0.29	0.41	0.40
1320	0.07	0.10	0.58	1.19	0.13	0.25	0.37	0.34
1370	0.05	0.08	0.56	1.10	0.09	0.22	0.31	0.29
1420	0.02	0.05	0.55	1.00	0.08	0.18	0.25	0.22
1475	0	0.03	0.53	0.90	0.08	0.16	0.18	0.15
1525	0	0	0.53	0.79	0.10	0.14	0.09	0.07

3.2.1.7. Rectangular Sheet Metal Duct Elbows

Table 3.11 displays insertion loss values for unlined and lined square elbows without turning vanes. For lined square elbows, the duct lining must extend at least two duct widths beyond the elbow. Table 3.11 applies only where the duct is lined before and after the elbow. Table 3.12 gives the insertion loss values associated with radiused elbows. Table 3.13 gives the insertion loss values for unlined and lined square elbows with turning vanes. The value of $f \times w = (f_w)$ in Tables 3.11 through 3.13 is the center frequency of the octave frequency band times the width of the elbow [5].

Table 3.11 Insertion Loss of Unlined and Lined Square Elbows without Turning Vanes [5]

	Insertion Loss, dB	
	Unlined Elbows	**Lined Elbows**
$fw < 48$	0	0
$48 \leq fw < 96$	1	1
$96 \leq fw < 190$	5	6
$190 \leq fw < 380$	8	11
$380 \leq fw < 760$	4	10
$fw > 760$	3	10

Note: fw = f x w where f = center frequency, kHz, and w = width, mm

Table 3.12 Insertion Loss of Round Elbows [5]

	Insertion loss, dB
$fw < 48$	0
$48 \leq fw < 96$	1
$96 \leq fw < 190$	2
Fw>190	3

Note: fw = f x w where f = center frequency, kHz, and w = width, mm

Table 3.13 Insertion Loss of Unlined and Lined Square Elbows with Turning Vanes [5]

	Insertion Loss, dB	
	Unlined Elbows	**Lined Elbows**
$fw < 48$	0	0
$48 \leq fw < 96$	1	1
$96 \leq fw < 190$	4	4
$190 \leq fw < 380$	6	7
$fw > 380$	4	7

Note: fw = f x w where f = center frequency, kHz, and w = width, mm

3.2.1.8. Non-metallic Insulated Flexible Ducts

Nonmetallic insulated flexible ducts can significantly reduce airborne noise. Insertion loss values for specified duct diameters and lengths are given in Table 3.14. Recommended duct lengths are normally from 1 to 2 m. Care should be taken to keep flexible ducts straight; bends should have as long a radius as possible. While an abrupt bend may provide some additional insertion loss, the airflow generated noise associated with the airflow in the bend may be unacceptably high. Because of potentially high breakout sound levels associated with flexible ducts, care should be exercised when using flexible ducts above sound sensitive spaces [5].

Table 3.14 Lined Flexible Duct Insertion Loss [5]

Diameter mm	Length m	Insertion Loss, dB						
		Octave band center frequency, Hz						
		63	125	250	500	1000	2000	4000
100	3.7	6	11	12	31	37	42	27
	2.7	5	8	9	23	28	32	20
	1.8	3	6	6	16	19	21	14
	0.9	2	3	3	8	9	11	7
125	3.7	7	12	14	32	38	41	26
	2.7	5	9	11	24	29	31	20
	1.8	4	6	7	16	19	21	13
	0.9	2	3	4	8	10	10	7
150	3.7	8	12	17	33	38	40	26
	2.7	6	9	13	25	29	30	20
	1.8	4	6	9	17	19	20	13
	0.9	2	3	4	8	10	10	7
175	3.7	9	12	19	33	37	38	25
	2.7	6	9	14	25	28	29	19
	1.8	4	6	10	17	19	19	13
	0.9	2	3	5	8	9	10	6
200	3.7	8	11	21	33	37	37	24
	2.7	6	8	16	25	28	28	18
	1.8	4	6	11	17	19	19	12
	0.9	2	3	5	8	9	9	6

Table 3.14 (cont.)

Diameter mm	Length m	Insertion Loss, dB						
		Octave band center frequency, Hz						
		63	125	250	500	1000	2000	4000
225	3.7	8	11	22	33	37	36	22
	2.7	6	8	17	25	28	27	17
	1.8	4	6	11	17	19	18	11
	0.9	2	3	6	8	9	9	6
250	3.7	8	10	22	32	36	34	21
	2.7	6	8	17	24	27	26	16
	1.8	4	5	11	16	18	17	11
	0.9	2	3	6	8	9	9	5
300	3.7	7	9	20	30	34	31	18
	2.7	5	7	15	23	26	23	14
	1.8	3	5	10	15	17	16	9
	0.9	2	2	5	8	9	8	5
350	3.7	5	7	16	27	31	27	14
	2.7	4	5	12	20	23	20	11
	1.8	3	4	8	14	16	14	7
	0.9	1	2	4	7	8	7	4
400	3.7	2	4	9	23	28	23	9
	2.7	2	3	7	17	21	17	7
	1.8	1	2	5	12	14	12	6
	0.9	1	1	2	6	7	6	2

Note : The 63 Hz insertion loss values are estimated from higher frequency insertion loss values.

3.2.1.9. Duct Silencers

Duct silencers are used to attenuate sound that is transmitted through HVAC systems; particularly duct systems. There are various sizes of both rectangular and circular sizes of mufflers. They can add pressure and energy losses. Therefore; when selecting silencers, the following parameters should be considered;

- **Airflow pressure drop**, including system effects for less than ideal flow conditions
- **Insertion loss,** which is the reduction in sound power level at a given location due solely to the placement of a sound attenuating device in the transmission path between the sound source and the given location

Airflow generated noise is created as the air flows into, through, and out of the silencer. In the majority of installations the airflow generated noise is much less than, and does not contribute to, the silenced noise level on the quiet side of the silencer. In general, airflow generated noise should be evaluated if static pressure drops exceed 87 Pa, the noise criteria is below RC-35, or the silencer is located very close to or in occupied space.

There are three types of HVAC duct silencers; dissipative (with acoustic media), reactive (no media), and active silencers.

Dissipative Silencers: Generally use perforated metal surfaces covering acoustic grade fiberglass to attenuate sound over a broad range of frequencies. Airflow does not significantly affect the insertion loss if pressure drops are under 87 Pa [5]. As with most dissipative mufflers, the attenuation is highest in the mid-frequency range; it is limited in the low frequency range (63 Hz octave), and limited to a lesser degree in the high frequency range (8000 Hz octave) [6].

Reactive Silencers: use tuned perforated metal facings covering tuned chambers void of any fibrous material. Because of tuning broadband insertion loss is more difficult to achieve than with dissipative silencers. Longer lengths may be required to achieve similar insertion loss performance as dissipative silencers. Airflow generally increases the insertion loss of reactive silencers.

Active Silencers: Reduce noise at lower frequencies by producing inverse sound waves that cancel the unwanted noise. An input microphone measures the noise in the duct and converts it to electrical signals. These signals are processed by a digital computer where exact opposite, mirror image sound waves of equal amplitude are generated. This secondary noise source destructively interferes with the noise and cancels a significant portion of the unwanted sound. Performance is limited, however, by the presence of excessive turbulence in the airflow detected by the microphones.

Manufacturers recommend using active silencers where duct velocity is less than 7.6 m/s and where the duct configurations are conductive to smooth evenly distributed airflow. Since insertion loss measurements use a substitution technique, reasonable (± 3 dB) insertion loss values can be achieved down to 63 Hz. Active systems can cancel low frequency random broadband and/or tonal repetitive noise. Typical attenuations of 20 to 38 dB have been reported in the 40 through 4000 Hz frequency range.

Standard silencers should be located at least three duct diameters from a fan, coil, elbow, branch takeoff, or other duct element. Locating a standard silencer closer than three duct diameters can result in a significant increase in both the pressure loss across the silencer and the generated noise [5].

3.2.1.10. Prediction of Noise Along Other Branches at Junctions

Knowing the octave band sound power level at a specific branch of a junction makes it possible to predict the flow generated noise along other branches or in the main duct, using Figure 3.8.

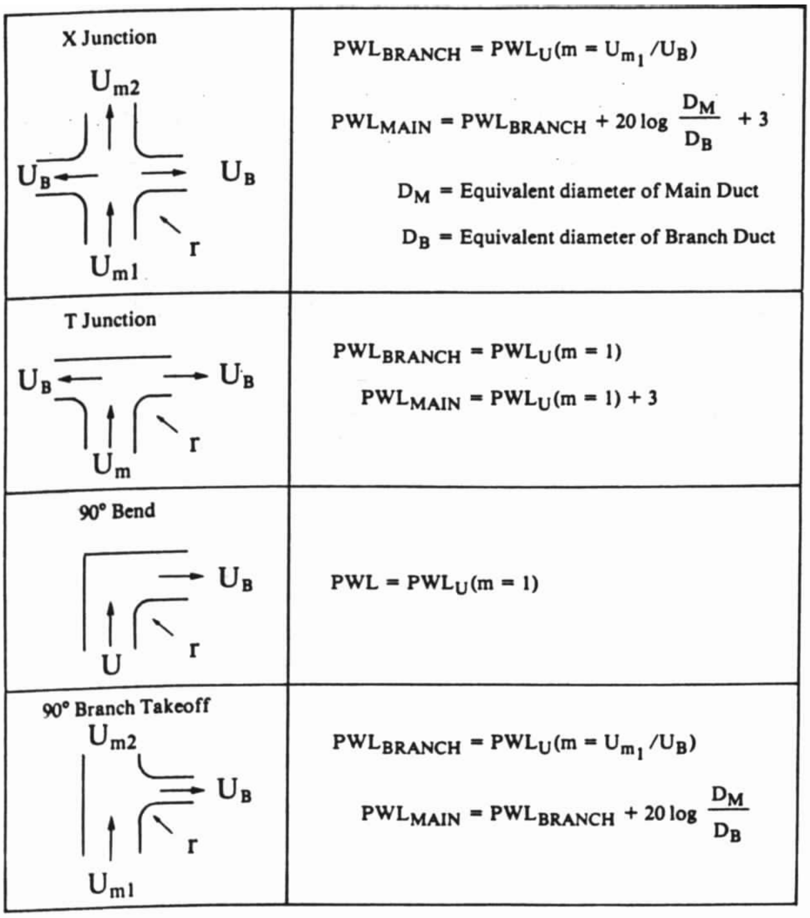


Figure 3.8 Prediction of Flow-Generated Noise for Junctions and Turns [6]

3.2.1.11. Duct Branch Sound Power Division

When sound travelling in a duct encounters a junction, the sound power contained in the incident sound waves in the main feeder duct is distributed between the branches associated with the junction. This division of sound power is referred to as the branch sound power division. The corresponding attenuation of sound power that is transmitted down each branch of the junction is comprised of two components. The first is associated with the reflection of the incident sound wave if the sum of the cross sectional areas of the individual branches $\sum SB_i$ differs from the cross sectional area S_M of the main feeder duct. The second component is associated with the energy division according to the ratio of the cross sectional area $\sum SB_i$ of an individual branch divided by the sum of the cross sectional areas of the individual branches $\sum SB_i$. The second component is the dominant component. Values for the attenuation of sound power ΔLB_i at a junction that are related to the sound power transmitted down an individual branch of the junction are given in Table 3.15 [5].

Table 3.15 Duct Branch Sound Power Division in dB [5]

$S_i / \sum S_{Bi}$	ΔL_{Bi}	$S_i / \sum S_{Bi}$	ΔL_{Bi}
1.00	0	0.10	10
0.80	1	0.08	11
0.63	2	0.063	12
0.50	3	0.50	13
0.40	4	0.040	14
0.32	5	0.032	15
0.25	6	0.025	16
0.20	7	0.020	17
0.16	8	0.016	18
0.12	9	0.012	19

3.2.1.12. Duct End Reflection Loss

When low frequency plane sound waves interact with openings that discharge into a large room, a significant amount of the sound energy incident on this surface is reflected back into the duct. In other words; when plane wave sound passes from a small space, such as a duct, into a large space, such as a room, a certain amount of sound is reflected back into the duct. The sound attenuation values ΔL associated with duct end reflection losses for ducts terminated in free space are given in Table 3.16, and for ducts terminated flush with a wall are given in Table 3.17. Diffusers that terminate in a suspended lay-in acoustic ceiling can be treated as terminating in free space. To use Table 3.16, if the duct terminating into a diffuser is rectangular, the duct diameter D is given by

$$D = \sqrt{4A/\pi} \tag{3. 17}$$

where;

A: area of the rectangular duct in m^2

Table 3.16 Duct End Reflection Loss - Duct Terminated in Free Space [5]

Duct Diameter, mm	**End reflection Loss, dB Octave Band center frequency, Hz**					
	63	**125**	**250**	**500**	**1000**	**2000**
150	20	14	9	5	2	1
200	18	12	7	3	1	0
250	16	11	6	2	1	0
300	14	9	5	2	1	0
400	12	7	3	1	0	0
510	10	6	2	1	0	0
610	9	5	2	1	0	0
710	8	4	1	0	0	0
810	7	3	1	0	0	0
910	6	3	1	0	0	0
1220	5	2	1	0	0	0
1830	3	1	0	0	0	0

Table 3.17 Duct End Reflection Loss - Duct Terminated Flush with Wall [5]

Duct Diameter, mm	**End reflection Loss, dB Octave Band center frequency, Hz**				
	63	**125**	**250**	**500**	**1000**
150	18	13	8	4	1
200	16	11	6	2	1
250	14	9	5	2	1
300	13	8	4	1	0
400	10	6	2	1	0
510	9	5	2	1	0
610	8	4	1	0	0
710	7	3	1	0	0
810	6	2	1	0	0
910	5	2	1	0	0
1220	4	1	0	0	0
1830	2	1	0	0	0

Table 3.16 and Table 3.17 have some limitations. The tests on which these equations are based were conducted with straight sections of round ducts. These ducts directly terminated into a reverberation chamber with no restriction on the end of the duct or with a round orifice placed over the end of the duct. Diffusers can be either round or rectangular. They usually have a restriction associated with them that may either be a damper, guide vanes to direct airflow, a perforated metal facing, or a combination of these elements. Currently, there is no data that indicate the effects of these elements. It is not known whether or not these elements react similar to the orifices used in the above mentioned tests. As a

result, the effects of an orifice placed over the end of a duct are not included in Table 12 and Table 13. Equation (3.17) does yield reasonable results with diffusers that have low aspect ratios (length/width). However, many types of diffusers (particularly slot diffusers) have high aspect ratios. Many diffusers do not have long straight sections (greater than three duct diameters) before they terminate into a room. Many duct sections between a main feed branch and a diffuser are curved or have short, stubby sections. The effects of these configurations on the duct end reflection loss are not known. However; Table 12 and 13 can be used with reasonable accuracy for many diffuser configurations [5].

The sound attenuation associated with duct end reflection losses can be approximated by [15]:

$$\Delta L = 10\,log\left[1 + (1.150/(4*3.14159*f))^2 * (2*3.14159/A)\right] \qquad (3.18)$$

where;

ΔL : attenuation in dB

A: cross sectional area of the duct in m^2

f: frequency in Hz

For rectangular/square ducts;

$A = d^2$

For round ducts;

$A = \pi d^2 / 4$

where;

d: diameter of a circular duct or effective diameter of a rectangular duct in m

End reflection loss should not be included in the attenuation of any system where linear diffusers are tapped directly into plenums, where diffusers are connected to primary ductwork with curved elements (flexible ductwork), or where the distances between diffusers and the primary duct are less than 3 equivalent duct diameters [6].

3.2.2. SOUND RADIATION THROUGH DUCT WALLS

3.2.2.1. Airflow Generated Duct Rumble

An HVAC fan and its connected ductwork can act as a semi-closed, compressible fluid pumping system where both acoustic and aerodynamic air pressure fluctuations at the fan are transmitted to other locations in the duct system. Air pressure fluctuations can be caused by variations in fan, motor or fan belt RPM or by airflow instabilities transmitted to the fan housing or nearby connected ductwork. When the air pressure fluctuations encounter large, flat, unreinforced duct surfaces that have resonance frequencies near or equal to the disturbing frequencies, the duct surfaces will vibrate. In typical HVAC duct systems, duct wall vibration can produce sound pressure levels of the order of 65 to 95 dB at frequencies that range from 10 Hz to 100 Hz. This type of duct generated sound is generally called duct rumble [5].

Low frequency noise that breaks out of ductwork can give the perception of low frequency noise, or rumble. The noise might be due to one or a combination of aerodynamic conditions or to fan noise.

The reduction of energy passing through a typical rectangular duct wall is less than the TL of the duct up to the limit of TL-3 dB for large S/A (long ducts). The measure of the breakout is the duct breakout noise reduction loss, and can be estimated from the tabulated TL and the duct dimensions as follows [6]:

$$\text{Br Noise Reduction} = TL - 10\log(S/A) \text{ in dB} \qquad (3.\ 19)$$

where;

TL: sound transmission loss in Table 3.18 for rectangular ducts in dB
S: surface area of length of duct being considered in m^2
A: cross-sectional area of duct in m^2

If the sound power level entering the duct is L_{wd}, the sound power radiated through the duct surface is

$$L_{wr} = L_{wd} - Brnoisereduction \tag{3. 20}$$

For example the value for a 10 m^3/s fan might be 100 dB at 63 Hz for L_{wd}. Sound transmission loss might be 20 dB, with a breakout noise reduction of 12 dB for a 4.6 m duct. The sound power radiated from the section is 88 dB.

Table 3.18 TL_{out} versus Frequency for Various Rectangular Ducts in dB [15]

Duct size		**Octave band center frequency, Hz**							
mm	**Thickness, mm**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
300x300	0.7 (24 ga)	21	24	27	30	33	36	41	45
300x600	0.7 (24 ga)	19	22	25	28	31	35	41	45
300x1200	0.85 (22 ga)	19	22	25	28	31	37	43	45
600x600	0.85 (22 ga)	20	23	26	29	32	37	43	45
600x1200	1.0 (20 ga)	20	23	26	29	31	39	45	45
1200x1200	1.3 (18 ga)	21	24	27	30	35	41	45	45
1200x2400	1.3 (18 ga)	19	22	25	29	35	41	45	45

Note: The data are from tests on 6 m long ducts, but the TL values are for ducts of the cross section shown regardless of length.

3.2.2.2. Duct-borne Cross-talk

Duct-borne cross-talk is sound transmitted between rooms by way of the duct system. It is controlled by using duct linings, splitters, prefabricated duct attenuators, and limiting straight duct routings. Generally, the requisite duct attenuation to prevent cross-talk is about 5 dB greater than the transmission loss (TL) of the intervening architectural construction, but if ceiling heights, room sound absorbing treatments, or background noise levels are radically different between source and receiving rooms, the needed duct attenuation may be even greater [6].

3.2.2.3. Transmission Loss Through Duct Walls

Breakout is the sound associated with fan or airflow noise inside a duct radiates through the duct walls into the surrounding area. Breakout can be a problem if it is not adequately attenuated before the duct runs over an occupied space. Noise can also be transmitted into the duct in one space and radiated from the duct to another space. Sound that is transmitted into a duct from the surrounding area is called break-in [5].

3.2.2.3.1. Transmission Loss of Rectangular Ducts

3.2.2.3.1.1. Breakout Transmission Loss

The breakout transmission loss TL_{out} of duct walls is defined as:

$$TL_{out} = 10\log\left[(W_i A_0)/(A_i W_r)\right] \text{ dB} \tag{3. 21}$$

where;

W_i : sound power in the duct in dB

W_r : sound power radiated from the walls in dB

$A_i = \text{ab}$

$A_0 = 2\text{L(a+b)}$

A_i : cross-sectional area of inside of the duct in m^2

A_0 : surface area of outside sound-radiating surface of the duct in m^2

a: larger dimension of duct cross section in m

b: smaller dimension of duct cross section in m

L: duct length in m

To obtain W_r, Equation (3. 21) may be written as follows;

$$L_{W_r} = L_{W_i} + 10\log(A_0 / A_i) - TL_{out} \tag{3. 22}$$

where;

$$L_{W_r} = 10\log(W_r \times 10^{12})$$
$$L_{W_i} = 10\log(W_i \times 10^{12})$$

Table 3.18 lists values for TL_{out} in Equation (3. 22) for selected duct sizes.

Since the sound power within the duct slowly decreases along the length of the duct, the actual TL_{out} as defined above increases slightly as the length increases for any type of duct. This effect does not apply to the prediction method and is usually minimal.

The transmission loss curve can be divided into two regions, one where the plane mode transmission within the duct predominates, and another where multi-mode transmission is dominant. The limiting frequency between these two regions is given by:

$$f_L = 613000/(ab)^{0.5} \qquad (3.\ 23)$$

where;

a and b in meters.

For $f < f_L$, the plane mode breakout TL may be calculated from the formula;

$$TL_{out} = 10\log\left[fq^2/(a+b)\right] + 17 \text{ dB} \qquad (3.\ 24)$$

where;

q: mass/unit area of the duct walls in kg/m^2

f: frequency in Hz

a, b: duct cross section dimensions in mm

The minimum possible value of TL_{out} occurs where $W_r = W_i$ and a lower limit is thus imposed on TL_{out}:

$$TL_{out}(\min) = 10\log\left[1000L(1/a + 1/b)\right] \qquad (3.\ 25)$$

where;

L is in meters and a, b are in millimeters.

When $f \geq f_L$, the magnitude TL may be found from the formula;

$$TL_{out} = 20\log(qf) - 45 \text{ dB} \qquad (3.\ 26)$$

where again;

q is in kg/m^2 and f is in Hz.

This method is based on a mass law impedance for the duct walls and is valid only for:

$$f = 144000 t^{0.5} / a \tag{3. 27}$$

where;

t: duct wall thickness in mm
a: larger inverse wall dimension in mm

This frequency limit applies only to galvanized steel ducts. Below this frequency, the *actual* TL will exhibit damped resonances, which produce progressively more pronounced maxima and minima in the TL curve as the frequency falls. The present method still predicts the overall behavior, however, and generally gives conservative predictions at low frequencies. The effects of wave coincidence are not explicitly included in the prediction scheme (aside from the 45 dB limit on TL_{out}), but these would occur, for ordinary ductwork, at frequencies above the range of interest.

Duct fittings, such as elbows, do not materially affect the TL but should be include in the effective radiating surface area when the actual sound power radiated from a length of ductwork is being calculated.

3.2.2.3.1.2. Break-in Transmission Loss

The incident sound power from the surrounding space is W_i, and sound power W_t travels out along the duct in *both* directions. The break-in TL can be defined as

$$TL_{in} = 10\log(W_i / 2W_t) \tag{3. 28}$$

which is the logarithmic ratio of the total incident sound power W_i to the total transmitted power $2W_t$. To obtain sound power, W_t transmitted down a duct in a specified direction, Equation (3. 28) must be written as follows:

$$L_{W_t} = L_{W_i} - TL_{in} - 3 \text{ dB} \tag{3. 29}$$

This definition is more in line with the common definition of the TL of a partition than that of the breakout TL. An equivalent definition cannot be used in the latter case because there is no well-defined incident (as opposed to reflected) sound power. The incident sound power level L_{W_i} may be related to sound pressure level in the reverberant field L_{prev} as follows:

$$L_{W_i} = L_{prev} + 10\log A_0 - 6 \text{ dB} \tag{3. 30}$$

where;

W_i is in watts and A_0 is in square meters.

The break-in TL may be calculated in terms of TL_{out} (since it is difficult to calculate directly) by one of two formulas - the first applies at frequencies below the cut-off frequency f_L, for the lowest acoustic cross-mode in the duct, and the second, at frequencies above f_L. For rectangular ducts, this cut-off frequency is given by;

$$f_L = 172000 / a \tag{3. 31}$$

where;

a the larger duct dimension is in millimeters.

For $f \le f_L$, TL_{in} equals the larger of the following:

$$TL_{out} = 10\log(a/b) + 20\log(f / f_L) - 4 \text{ dB} \tag{3. 32}$$

or

$$10\log[1000L(1/a + 1/b)] \tag{3. 33}$$

where;

L is in meters and a, b in millimeters.

For $f > f_L$, TL_{in} is given by;

$$TL_{in} = TL_{out} - 3 \text{ dB} \tag{3. 34}$$

Table 3.19 shows some values of TL_{in}, corresponding to the ducts listed in Table 3.18.

Table 3.19 TL_{in} versus Frequency for Various Rectangular Ducts in dB [5]

Duct size		Octave band center frequency, Hz							
mm	**Thickness, mm**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
300x300	0.7 (24 ga)	16	16	16	25	30	33	38	42
300x600	0.7 (24 ga)	15	15	17	25	28	32	38	42
300x1200	0.85 (22 ga)	14	14	22	25	28	34	40	42
600x600	0.85 (22 ga)	13	13	21	26	29	34	40	42
600x1200	1.0 (20 ga)	12	15	23	26	28	36	42	42
1200x1200	1.3 (18 ga)	10	19	24	27	32	38	42	42
1200x2400	1.3 (18 ga)	11	19	22	26	32	38	42	42

Note: The data are from tests on 6 m long ducts, but the TL values are for ducts of the cross section shown regardless of length.

3.2.2.3.2. Transmission Loss of Circular Ducts

3.2.2.3.2.1. Breakout Transmission Loss

Simple prediction methods for the TL of circular ducts can disagree with actual measurements by 20 to 30 dB at low frequencies, so only general comments and data on selected ducts are given hereby.

Equation (3. 22) applies either for circular ducts. The equations for A_0 and A_i for round ducts are;

$$A_0 = L\pi d \quad (3.\ 35)$$

$$A_i = \pi d^2 / 4 \quad (3.\ 36)$$

where;

d: duct diameter in m
L: length of duct sound radiating surface in m

TL_{out} values for circular ducts are given in Table 3.20.

Table 3.20 Experimentally Measured TL_{out} versus Frequency for Circular Ducts [5]

			TL_{out}, dB						
Diameter	**Length**		**Octave band center frequency, Hz**						
mm	**m**	**gage**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
Long seam ducts									
205	4.6	26	>45	(53)	55	52	44	35	34
355	4.6	24	>50	60	54	36	34	31	25
560	4.6	22	>47	53	37	33	33	27	25
815	4.6	22	(51)	46	26	26	24	22	38
Spiral wound ducts									
205	3.0	26	>48	>64	>75	72	56	56	46
355	3.0	26	>43	>53	55	33	34	35	25
660	3.0	24	>45	50	26	26	25	22	36
660	3.0	16	>48	53	36	32	32	28	41
815	3.0	22	>43	42	28	25	26	24	40

Note: In cases where background sound swamped the sound radiated from the duct walls, a lower limit on TL_{out} is indicated by a > sign. Parentheses indicate measurements in which background sound has produced a greater uncertainty than usual.

An ideal circular duct has a high TL at low frequencies because uniform internal sound pressure fluctuations cause a uniform "breathing" in the pipe walls that presents high impedance to the sound waves. Small departures from the circularity in the ducts cause higher structural modes to be excited in the duct walls, even with plane wave transmission within the duct. Spiral wound ducts are more nearly circular than long seam ducts and may have a significantly higher TL at low frequencies. However, both types of duct exhibit similar behavior at high frequencies.

Low frequency sound and vibration energy below 50 Hz can be important because it relates to motors running to shaft speeds of 29 Hz (1750 rpm) and fans running at shaft speeds of 15 to 25 Hz in typical HVAC equipment.

Although the TL curve is complex, it is highest at "low" frequencies. Its maximum value (discounting the very low frequency rise) is between 50 and 300 Hz, depending on the duct size. Here, the TL can be 30 dB or more higher than that of a rectangular duct. The internal sound power levels in this region also tend to be highest. In this respect, circular ducts are far superior to either rectangular or flat oval ducts.

Duct fittings, such as elbows, appear to reduce the TL to some extent, although this effect appears to be most severe at low frequencies.

Compared with Table 3.20, Table 3.18 shows that round ductwork is much more effective in containing low-frequency noise than rectangular ductwork, regardless of whether the noise is flow-generated or due to acoustic excitation. In tight spaces, multiple round ducts in parallel can be used where space is not available for a single round duct of equivalent cross-sectional area. Flat-oval ductwork falls between round and rectangular, and its breakout noise reduction depends on the area of the flat surface.

Providing good airflow conditions at or near the fan is the most effective method of avoiding possible rumble conditions.

Noise and turbulence inside a duct cause the walls to vibrate and radiate noise to surrounding spaces. Rectangular ducts are more susceptible to this than are round ducts, and the problem is a function of duct size and air velocity. To prevent excessive noise radiation from duct walls, all fittings should be smooth and designed to avoid abrupt changes of direction or velocity. Whenever possible, medium and high velocity ducts and terminal boxes should be located in non-critical areas (e.g., above corridors) [5].

3.2.2.3.2.2. Break-in Transmission Loss

The definition of Equation (3. 28) still applies for the break-in TL of circular ducts, and Equation (3. 30) can still be used to relate W_i to the reverberant field sound pressure level, if the correct value of A_0 is inserted.

In the case of circular ducts, the cut-off frequency for the lowest acoustic cross-mode is given by;

$$f_L = 201400 / d \tag{3. 37}$$

where;

d is the duct diameter in millimeters, and the break-in TL can be found, as before, in terms of TL_{out}.

where $f \leq f_L$, is given by

$$TL_{in} = \text{the larger of } \begin{cases} TL_{out} + 20\log\left(f / f_1\right) - 4 \\ 10\log(2L / d \end{cases} \tag{3. 38}$$

where;

L and d are in meters;

for $f > f_L$, TL_{in} is given by Equation (3. 34)

Table 3.21 gives the TL_{in} values for the ducts listed in Table 3.20.

Table 3.21 Experimentally Measured TL_{in} versus Frequency for Circular Ducts [5]

			TL_{in}, dB						
Diameter	**Length**		**Octave band center frequency, Hz**						
mm	**m**	**gage**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
Long seam ducts									
205	4.6	26	>17	(31)	39	42	41	32	31
355	4.6	24	>27	43	43	31	31	28	22
560	4.6	22	>28	40	30	30	30	24	22
815	4.6	22	(35)	36	23	23	21	19	35
Spiral wound ducts									
205	3.0	26	>20	>42	>59	>62	53	43	26
355	3.0	26	>20	>36	44	28	31	32	22
660	3.0	24	>27	38	20	23	22	19	33
660	3.0	16	>30	>41	30	29	29	25	38
815	3.0	22	>27	32	25	22	23	21	37

Note: In cases where background sound swamped the sound radiated from the duct walls, a lower limit on TL_{in} is indicated by a > sign. Parentheses indicate measurements in which background sound has produced a greater uncertainty than usual.

3.2.2.3.3. Transmission Loss of Flat-oval Ducts

3.2.2.3.3.1. Breakout Transmission Loss

Vibration measurements indicate that at low to mid frequencies, most of the acoustic radiation emanates from the flat duct sides; at high frequencies, the duct radiates uniformly. The flat sides vibrate in much the same way as the walls of rectangular ducts at low frequencies, with the curved end acting as springs. Resonant behavior, probably related to the ring frequency phenomenon of circular ducts, prevails at high frequencies. Available data show that the minimum in the TL curve is at the ring frequency of the equivalent circular duct, the diameter of which is equal to the smaller dimension of the flat oval duct.

Although flat oval ducts combine undesirable features of rectangular and circular ducts, they have one advantage – only the flat sides of the duct radiate acoustical energy, primarily at low to mid frequencies. This means that these ducts tend to have a higher TL than rectangular ducts in the low frequency region, resulting in a difference of 8 to 10 dB in some cases. However, circular ducts are superior to flat oval and rectangular ducts at low frequencies.

The breakout TL is defined by Equation (3. 21) but A_0 and A_i are now given by the expressions;

$$A_0 = L\left[2(a-b)+\pi b\right] \tag{3. 39}$$

$$A_i = b(a-b)+\frac{\pi b^2}{4} \tag{3. 40}$$

where;

a: length of larger major axis in m

b: length of minor duct axis in m

L: length of duct sound radiating surface in m

The duct perimeter P is given by;

$$P = 2(a-b)+\pi b \quad (3.\ 41)$$

and the fraction σ of P taken up by the flat duct sides is given by;

$$\sigma = 1/[1+\pi b/2(a-b)] \quad (3.\ 42)$$

To estimate TL_{out} for flat oval ducts at low to mid frequencies, draw a baseline of TL_{out} (min),

where;

$$TL_{out}(\min) = 10\log(A_0 / A_i) \quad (3.\ 43)$$

where;

A_0 and A_i (in the same units) are given by Equations (3. 44) and Equation (3. 45). Next plot a curve of

$$TL_{out} = 10\log(q^2 f / \sigma^2 P) + 20 \text{ dB} \quad (3.\ 44)$$

where;

q is the mass/unit area of the duct walls in kg/m2 and f the frequency in Hz from the baseline up to a frequency of

$$f_L = 206/b \quad (3.\ 45)$$

where;

b is the minor axis of the duct cross section in meters. The frequency f_L is the upper limit of applicability of Equation (3. 44), and is equal to one-eight of the ring frequency of the equivalent circular duct. Finally, erase that portion of the baseline to the right of the point where it meets this curve.

The comments on the low frequency TL_{out} of rectangular ducts should also apply to flat oval ducts; low frequency resonance effects may be expected, but Equation (3. 43) and Equation (3. 44) should still predict the overall behavior or give conservative predictions. The effects of fittings can be expected to be small at low frequencies; but, as in the case of rectangular ducts, they should be included in calculating the surface area of the duct.

Table 3.22 gives some values of TL_{out} for flat oval ducts of various sizes. The upper frequency limit on TL_{out} is imposed by Equation (3. 45).

Table 3.22 TL_{out} versus Frequency for Flat Oval Ducts [5]

Duct size		**TL_{out}, dB Octave band center frequency, Hz**							
mmxmm	**Gage**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
305x150	24	31	34	37	40	43	-	-	-
610x150	24	24	27	30	33	36	-	-	-
610x305	24	28	31	34	37	-	-	-	-
1220x305	22	23	26	29	32	-	-	-	-
1220x610	22	27	30	33	-	-	-	-	-
2440x610	20	22	25	28	-	-	-	-	-
2440x1220	18	28	31	-	-	-	-	-	-

Note: The data are for duct lengths of 6.1 m, but the values may be used for the cross-section shown regardless of length.

Equation (3. 22) assumes no interior sound attenuation along the length of the duct sound radiating surface. Thus, it is valid only for unlined ducts. It is generally valid for duct lengths up to 9 m. L_{w_r} must always be equal to or less than L_{w_i}.

For most applications including rectangular, circular and flat-oval ducts, the sound pressure level in an occupied space as a result of duct sound breakout can be obtained from [5]

$$L_p = L_{w_r} - 10\log[\pi r L] + 10 \qquad (3.\ 46)$$

where;

L_p : sound pressure level at a specified point in the space in dB

L_{w_r} : sound power level of sound radiated from outside surface of duct walls given by Equation (3. 22) in dB

r: distance between duct and position at which L_p is being calculated in m

L: length of duct sound radiating surface in m

3.2.2.3.3.2. Break-in Transmission Loss

The break-in TL of flat oval ducts is defined by Equation (3. 28); again, Equation (3. 30) relates the reverberant field sound pressure level to W_i.

While there are no exact solutions for the cut-off frequency for the lowest acoustic cross mode in flat oval ducts, an approximate solution exists as follows:

$$f_l = \frac{172}{(a-b)\left[1+\pi b/2(a-b)\right]^{0.5}} \qquad (3.\ 47)$$

where;

a and b are in meters.

Equation (3. 47) is valid where $a/b \geq 2$; for $a/b < 2$, and deteriorates progressively as a/b approaches unity. Again, TL_{out} may be found in terms of TL_{in}, as follows;

where ;

$f \leq f_l$, TL_{in} is given by;

$$TL_{in} = \text{the larger of } \begin{cases} TL_{out} + 10\log(f^2 A_i) - 109 \\ 10\log(PL/A) + 27 \end{cases} \text{ dB} \qquad (3.\ 48)$$

where;

A_i is in square meters and is given by Equation (3. 40), P is in meters and is given by Equation (3. 39) and L is in meters.

If $f > f_l$, TL_{in} is given by Equation (3. 34).

Table 3.23 gives some values of TL_{in} for flat oval ducts of various sizes [6].

Table 3.23 TL_{in} versus Frequency for Flat Oval Ducts [5]

Duct size		**TL_{in}, dB Octave band center frequency, Hz**							
mmxmm	**Gage**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
305x150	24	18	18	22	31	40	-	-	-
610x150	24	17	17	18	30	33	-	-	-
610x305	24	15	16	25	34	-	-	-	-
1220x305	22	14	14	26	29	-	-	-	-
1220x610	22	12	21	30	-	-	-	-	-
2440x610	20	11	22	25	-	-	-	-	-
2440x1220	18	19	28	-	-	-	-	-	-

Note: The data are for duct lengths of 6.1 m, but the values may be used for the cross-section shown regardless of length.

Equation (3. 22) assumes no interior sound attenuation along the length of the duct sound radiating surface. Thus, it is valid only for unlined ducts. It is generally valid for duct lengths up to 9 m. L_{w_r} must always be equal to or less than L_{w_i}.

3.2.3. SOUND TRANSMISSION THROUGH CEILINGS

When terminal units, fan-coil units, air handling units, ducts, or return air openings to mechanical equipment rooms are located in a ceiling plenum above an occupied room, sound transmission through the ceiling system can be high enough to cause excessive noise levels in that room. There are no standard test procedures associated with measuring the direct transmission of sound through ceilings from sources close to the ceiling. This problem seems further complicated by the presence of light fixtures, diffusers, grilles, speakers, and so forth, which might be expected to reduce the transmission loss of the ceiling. For ceiling panels supported in a T-bar grid system, the leakage between the panels and the grid is the major transmission path.

To estimate the sound levels in a room associated with sound transmission through the ceiling, the sound power levels in the ceiling plenums must be attenuated by factors that account for the transmission loss of the ceiling and the plenum [5].

The procedure is:

1. Obtain the octave band radiated sound power levels of the device.

2. Subtract the environmental correction given in Table 3.24.

3. Calculate the surface area of the bottom panel of the source closest to the ceiling tiles.

4. From Table 3.25 find the adjustment to be subtracted from the sound power values at three frequencies given there.

5. Select the Ceiling/Plenum attenuations from Table 3.26, according to the ceiling type in use.

6. Subtract the three sets of values, taking account of sign where necessary, from the sound power values. The result is the average sound pressure level in the room.

7. For practical purposes, the sound field in the room may be assumed to uniform up to distances of about 5 m from the source.

Table 3.24 Environmental Correction to be Subtracted from Device Sound Power [5]

Environmental Correction, dB Octave band center frequency, Hz							
63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
4	2	1	0	0	0	0	0

Table 3.25 Compensation Factors for Source Area Effect [5]

Area Range, m^2 Octave band center frequency, Hz			
63	**125**	**250**	**Adjustment, dB**
Less than 0.26	Less than 0.22		-3
0.28 to 0.29	0.24 to 0.46	Less than 0.23	-2
0.51 to 0.72	0.49 to 0.71	0.27 to 0.63	-1
0.74 to 0.94	0.73 to 0.95	0.67 to 1.03	0
0.97 to 1.17	0.98 to 1.20	1.07 to 1.43	1
1.19 to 1.40	1.22 to 1.44	1.47 to 1.83	2
1.42 to 1.63	1.46 to1.68	1.87 to 2.23	3
1.65 to 1.85	1.71 to 1.93		4
1.88 to 2.08	1.95 to 2.17		5
2.10 to 2.31			6

Note: Find correct area in each frequency column and read adjustment from last column on right.

Table 3.26 Ceiling/Plenum/Room Attenuations in dB for Generic Ceilings with T-Bar Suspensions [5]

			Octave Band Frequency, Hz							
Tile Type	**Approx. Mass Per Unit Area, kg/m^2**	**Tile Thickness, mm**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Mineral fiber	4.9	16	13	16	18	20	26	31	36	0
Mineral fiber	2.4	16	13	15	17	19	25	30	33	0
Glass fiber	4.9	16	13	16	15	17	17	18	19	0
Glass fiber	2.9	50	14	17	18	21	25	29	35	0
Glass fiber with TL backing	2.9	50	14	17	18	22	27	32	39	0
Gypsum board tiles	8.8	13	14	16	18	18	21	22	22	0
Solid gypsum board ceiling	8.8	13	18	21	25	25	27	27	28	0
Solid gypsum board ceiling	11.2	16	20	23	27	27	29	29	30	0
Double layer of gypsum board	18.1	25	24	27	31	31	33	33	34	0
Double layer of gypsum board	22.0	32	26	29	33	33	35	35	36	0
Mineral fiber tiles, concealed spline mount	2.4 to 4.9	16	20	23	21	24	29	33	34	0

3.2.4. RETURN AIR SYSTEMS NOISE

The fan return air system provides a sound path (through the ducts or through an unducted ceiling plenum) between a fan and occupied spaces, typically with an opening into the mechanical equipment room. This condition can result in high sound levels in adjacent spaces. The high sound levels are caused by the close proximity of the fan and other sound sources in the mechanical equipment room and the low system attenuation between the mechanical equipment room and the adjacent spaces.

Sound propagates in the opposite direction to the airflow in a return air system, which results in a negligible effect on sound attenuation since the speed of sound (approximately 340 m/s) is much greater than typical return air velocities.

Noise in ducted return air systems is controlled by the fan intake sound power level. Unducted plenum return air is impacted by the sound power levels of the fan intake and casing radiated noise components. In certain installations, sound from other equipment located in the mechanical equipment room may also radiate through the wall opening and into adjacent spaces. Good design yields room return sound levels that are approximately 5 dB below the corresponding supply air system sound levels, so that it does not add significantly to the overall room sound pressure level.

Procedures for predicting the room sound pressure level in ducted return air systems can make use of Equation (3. 49) [6].

$$L_{pt}(1.5) = L_{ws} - 5\log X - 28\log h + 1.13\log N - 3\log f + 17 \text{ dB} \tag{3. 49}$$

where;

L_{pt} : average sound pressure level in a plane 1.5 m above the floor in dB re $20\mu Pa$

L_{ws} : sound power level of a single outlet in the array (i.e., the combined sound power level of that delivered by the distribution duct and that generated at the air terminal) in dB re 10^{-12} W

X: ratio of the floor area served by each outlet to the square of the ceiling height (X=1 if the area served equals h^2)

N: number of ceiling outlets in the room (N should be at least 4)

f : octave band center frequency in Hz

h: ceiling height in m

Unducted plenum return systems can use the following Equation (3. 50) [6], with adjustments made to the sound power level value to account for plenum losses.

$$L_p = L_w - 5\log V - 3\log f - 10\log r + 12 \text{ dB} \tag{3. 50}$$

where;

L_p : room sound pressure level at the chosen reference point in dB re $20\mu Pa$

L_w : source sound power level in dB re 10^{-12} W

V: room volume in m^3

f: octave band center frequency in Hz

r: distance from the source to the reference point in m

The composite transmission loss of the mechanical equipment room wall above the ceiling line and the loss associated with the ceiling plenum needs to be included. These values can be deducted from the sound power level to arrive at an adjusted sound power level at the ceiling return air opening [6]. The composite transmission loss of the wall can be calculated by the following steps;

1. Determine the wall sound transmission loss from Table 3.26.

2. Define the opening as a percentage of the wall area above the ceiling line:

$$\frac{Areaofopening}{WallArea} \times 100 \tag{3. 51}$$

3. Calculate the composite transmission loss TL_c of the wall element with an opening using Equation (3. 52)

$$TL_c = TL - 10\log\left[1 - S_2 / S_1 + (10^{TL/10}) S_2 / S_1\right] \tag{3. 52}$$

where;

TL: transmission loss of the wall in dB
S_1 : area of the wall in m^2
S_2 : area of the opening in m^2

3.2.5. RECEIVER SOUND ROOM CORRECTION

The sound pressure level at a given location in a room due to a particular sound source is a function of the sound power level and sound radiation characteristics of the sound source, the acoustic properties of the room (surface treatments, furnishings, etc.), the room volume, and the distance between the sound source and the point of observation. Two types of observation are typically encountered in HVAC applications: point source and line source. Typical point sources are sound radiated from grilles, registers, and diffusers; air-valve and fan powered air terminal units and fan coil units located in ceiling plenums; and return-air openings. Line sources are usually associated with sound breakout from air ducts and long slot diffusers.

For a point source in an enclosed space, classical diffuse-field theory predicts that as the distance between the source and point of observation is increased, the sound pressure level initially decreases at the rate of 6 dB Per doubling of distance. At some point, the reverberant sound field begins to dominate and the sound pressure level remains at a constant level.

Schultz (1985) and Thompson (1981) found that diffuse-field theory does not apply in real word rooms with furniture or other sound-scattering objects. Instead, the sound pressure levels decrease at the rate of around 3 dB per every doubling of distance between the sound source and the point of observation. Generally, a reverberant sound field does not exist in small rooms (room volume less than 420 m^3). In large rooms (room volume greater than 420 m^3), reverberant field usually exists, but usually at distances from the sound sources that are significantly greater than those predicted by diffuse-field theory.

3.2.5.1. Point Sound Sources

Most normally furnished rooms with regular proportions have acoustic characteristics that range from average to medium dead. These usually include carpeted rooms that have sound absorptive ceilings. If a normally furnished room has a room volume less than 420 m^3 and the sound source is a single point source, the sound pressure levels associated with the sound source can be obtained from

$$L_p = L_w + A - B \tag{3. 53}$$

where;

L_p : sound pressure level at specified distance from sound source in dB

L_w : sound power level of sound source in dB

Values for A and B are given in Table 3.27 and Table 3.28

Table 3.27 Values for A in Equation (3. 53) [5]

Room Volume m^3	Value for A, dB Octave band center frequency, Hz						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
42	4	3	2	1	0	-1	-2
71	3	2	1	0	-1	-2	-3
113	2	1	0	-1	-2	-3	-4
170	1	0	-1	-2	-3	-4	-5
283	0	-1	-2	-3	-4	-5	-6
425	-1	-2	-3	-4	-5	-6	-7

Table 3.28 Values for B in Equation (3. 53) [5]

Distance from sound source, m	Value for B, dB
0.9	5
1.2	6
1.5	7
1.8	8
2.4	9
3.0	10
4.0	11
4.9	12
6.1	13

If a normally furnished room has a room volume greater than 420 m^3 and the sound source is a single point source, the sound pressure levels associated with the sound source can be obtained from

$$L_p = L_w - C - 5 \tag{3. 54}$$

Values for C are given in Table 3.29.

Table 3.29 Values for C in Equation (3. 54) [5]

Distance from sound source, m	Value for C, dB Octave band center frequency, Hz						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
0.9	5	5	6	6	6	7	10
1.2	6	7	7	7	8	9	12
1.5	7	8	8	8	9	11	14
1.8	8	9	9	9	10	12	16
2.4	9	10	10	11	12	14	18
3.0	10	11	12	12	13	16	20
4.0	11	12	13	13	15	18	22
4.9	12	13	14	15	16	19	24
6.1	13	15	15	16	17	20	26
7.6	14	16	16	17	19	22	28
9.8	15	17	17	18	20	23	30

Equation (3. 54) can be used for room volumes of up to 4250 m^3. The accuracy of Equations (3. 53) and Equation (3. 54) is typically within 2 to 5 dB.

Equation (3. 50) can be used to estimate the sound pressure level at a chosen distance from a sound source in normal rooms, as a function of the source sound power level, room size, and frequency. Predictions should be accurate to ±2 dB.

Equation (3. 50) applies directly to a single sound source in the room. When the room has more than one source, the total sound pressure level at the reference point is obtained by adding (on an energy basis) the contribution of each source, using the corresponding L_w and r for each source [5].

3.2.5.2. Distributed Array of Ceiling Sound Sources

In many office buildings, air supply outlets are located flush with the ceiling of the conditioned space and constitute an array of distributed sound sources in the ceiling. The geometric pattern depends on the floor area served by each outlet, the ceiling height, and the thermal load distribution. In the interior zones of a building where the thermal load requirements are essentially uniform, the air delivery per outlet is usually the same throughout the space; thus, the sound sources tend to have nominally equal sound power levels.

For a distributed array of ceiling sound sources (air outlets) of nominally equal sound power, the room sound pressure levels tend to be uniform in the horizontal plane parallel to the ceiling. Although the sound pressure levels will decrease with distance from the ceiling along a vertical axis, the sound pressure levels along any selected horizontal plane are nominally constant. The calculation for a distributed ceiling array can be greatly simplified by using Equation (3. 55). For this case, it is desirable to use a reference plane of 1.5 m above the floor (the average distance between seated and standing head height). Thus, $L_{p(1.5)}$ is obtained from the following equation

$$L_{p(1.5)} = L_{w(s)} - D \qquad (3.\ 55)$$

where;

$L_{p(1.5)}$: sound pressure level 1.5 m above floor in dB

$L_{w(s)}$: sound pressure level of single diffuser in array in dB

Values for D are given in Table 3.30.

Table 3.30 Values for D in Equation (3.55) [5]

Floor area per Diffuser, m^2	**Value for D, dB Octave Band center frequency, Hz**						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
Ceiling height 2.4 to 2.7 m							
9.3 to 14	2	3	4	5	6	7	8
18.5 to 23	3	4	5	6	7	8	9
Ceiling height 3.0 to 3.7 m							
14 to 18.5	4	5	6	7	8	9	10
23 to 28	5	6	7	8	9	10	11
Ceiling height 4.3 to 4.9 m							
23 to 28	7	8	9	10	11	12	13
32.5 to 37	8	9	10	11	12	13	14

The total sound pressure level L_{pt} at a given reference point which results from the combination of individual sources in the room can be determined by using Equation (3. 50) to compute the L_p due to each source, and then summing these on an energy basis. However, Equation (3. 49) should be used for the frequently encountered situation in which the number and spacing of diffusers is influenced by the room geometric proportions and results in an approximately regular ceiling array.

Equation (3. 49) may be used to estimate the resulting room sound pressure level L_{pt} in a plane 1.5 m above the floor (standing head-eight) due to a distributed ceiling array of nominally similar diffusers. For diffuser spacings on the order of the ceiling height, the variation in sound pressure level at any point within the reference plane above the floor should be about 1 dB, provided that there are at least four diffusers in the ceiling array. Equation (3. 49) may also be used for an array of linear diffusers, by taking the source sound power level as that due to a single section and the number of sources as the number of sections in the array.

CHAPTER 4

NOISE CRITERIA FOR HVAC SYSTEMS

HVAC related sound is often the major source of background noise in indoor spaces. Whether an occupant considers the background noise to be acceptable or not generally depends on two factors. First is the **perceived loudness** of the noise relative to that of normal activities; if it is clearly noticeable then it is likely to be distracting and cause complaint. Second is the **quality** of the background noise; noise perceived as a rumble, roar, or hiss may result in complaints of annoyance and stress. The spectrum is then said to be unbalanced.

The acoustical design must ensure that HVAC noise is of sufficiently low level and unobtrusive quality so as not to interfere with the occupancy requirements of the space use. Consequently, methods of rating HVAC related noise should assess both the **relative loudness** and **quality** of the background noise.

4.1. SOUND RATING METHODS

Currently, several methods are used to rate indoor sound. They include the tangent noise criteria (Noise Criteria-NC method), Room Criteria-RC method, traditional A-weighted sound pressure level (dBA) and, the new RC Mark II method. Each sound rating method was developed based on data for specific applications; hence not all are suitable for the rating of HVAC related noise in the variety of applications encountered. The preferred sound rating methods generally comprise two distinct parts; a family of criterion curves (specifying sound levels by octave bands), and a companion procedure for rating the calculated or measured sound data relative to the criterion curves.

Ideally, HVAC related background noise should have the following characteristics;

- Balanced contributions from all parts of sound spectrum with no predominant bands of noise.
- No audible tones such as a hum or whine
- No fluctuations in level such as throbbing or pulsing

Table 4.2 summarizes the essential differences, advantages, and disadvantages of the rating methods that characterize HVAC related background noise [5].

4.1.1. Noise Criteria (NC) Method

Method is a single number rating that is somewhat sensitive to the relative loudness and speech interference properties of a given spectrum. The method consists of a family of criterion curves extending from 63 to 8000 Hz, and a **tangency rating procedure**. The criterion curves define the limits of octave band spectra that must not be exceeded to meet occupant acceptance in certain spaces. The rating is expressed as NC followed by a number (e.g., NC 40). The curves are arbitrarily numbered for easy reference. The Noise Criteria (NC) rating of a room is the number of the lowest curve that is not exceeded by any of the noise in the room. To determine an NC rating, the noise levels at 8 different octave band frequencies are measured and compared to the Noise Crtiteria curves.

The NC method is sensitive to level and has the disadvantage that the tangency method to determine the rating does not require that the noise spectrum approximate the shape of the NC curves. Thus, many different sounding noises can have the same numeric rating, but rank differently on the basis of subjective sound quality.

With the advent of VAV systems, excessive low frequency noise below the 63 Hz octave band became a serious problem, and the NC rating method does not address this. In many HVAC systems that do not produce excessive low frequency noise, the NC rating correlates relatively well with occupant satisfaction, if sound quality is not a significant concern [5].

4.1.1.1. Procedure for Determining NC Rating

The curve chosen for a given application depends on the type of space use and its sensitivity to the level of background noise. In principle the chosen NC curve defines the recommended octave-band limits of an acceptable background noise spectrum for a particular type of space use. However, when interpreted strictly as a limit on the level not to be exceeded in any octave band, experience has shown that noise problems are not avoided unless the shape of the actual spectrum approximates that of the chosen NC curve over three to four contiguous octave bands; when the shape of the actual spectrum approximates that of the NC curve in only one or two octave bands, the quality of the background noise may be objectionably rumbly or hissy [6].

However, in practice, both in assessment of the noise control design and in rating system noise in the field, the predicted (or measured) noise spectrum is not assigned an NC rating based on the highest NC curve tangent to the spectrum, regardless of where in the frequency range this occurs. This practice, while simple, has created both confusion and frustration in many field situations where a noise complaint has occurred. While an octave-band measurement of the noise may show that the specified NC limit has not been exceeded, the noise may sound unpleasant.

Based on the foregoing discussion, the NC tangent contour convention should not be used for field qualification of HVAC system background noise. A better method of specification requires that the spectrum shape of the background noise approximates that of an NC curve over at least three contiguous octave bands, without exceeding the limits defined by the specified NC curve.

4.1.2. RC Room Criteria Curve

The RC method consists of a family criterion curves and a rating procedure. This rating procedure [5] assesses background noise in spaces, both on the basis of its effect on speech communication and on subjective sound quality. The rating is expressed as RC followed by a number to show the level of noise and a letter to indicate the quality, for example RC 35 (N) where N denotes neutral.

The shape of the RC curve is a close approximation to a well-balanced, bland-sounding spectrum. It provides guidance whenever the space requirements dictate that a certain level of background sound be maintained for masking or other purposes. Generally, it is desirable to approximate the shape of the curve within ± 2 dB over the entire frequency range to achieve an optimum balance in sound quality. If the low frequency levels (31.5 to 250 Hz) exceed the design curve by as much as 5 dB, the sound is likely to be rumbly; exceeding the design curve by 3 dB at high frequencies (2000 to 4000 Hz) causes the sound to be hissy.

The shape of these curves differs from that of the NC curves at both low and high frequencies. While RC ratings may be an excellent tool for evaluating all sound in a space, they are not practical as a means of rating air terminals. When a terminal is rated against RC requirements, a low numerical value, with an “R” rating usually results. The numerical value that results (the average of the 500, 1000 and 2000 frequency bands) is typically so low that it has no impact on sound quality in the space, and the resultant “R” rating, gives no discrimination between units. RC is not therefore recommended, or practical, as a means of single number rating an air terminal. For diffusers, they usually result in a value identical to the determined NC value.

The RC procedure for noise rating corrects several of the shortcomings of the A-weighted sound level and NC rating methods, because the shape of the noise spectrum is taken into account in the assessment of the sound quality. In addition, the frequency range of evaluation extends down to the 16 Hz octave-band, thus addressing problems associated with extensive low-frequency noise.

4.1.2.1. Procedure for Determining RC Rating

The procedure [15] for determining the RC rating of an octave-band noise spectrum provides valuable information for use in estimating the likely acceptability of a given system design. Four steps are required in the procedure;

1. The first step is to plot the spectrum to be rated, and then calculate the arithmetic average of the octave-band levels in the 500, 1000, and 2000 Hz octave-bands. This average value becomes the numerical part of the RC rating which is important in addressing the speech communication or acoustical privacy requirements of the application, which are affected by the sound pressure levels in this frequency region.

2. The second step is to plot a reference curve that has a slope of –5 dB/octave from 16 Hz to 4000 Hz, which passes through the 1000 Hz octave band at the average value determined in the first step. This reference curve represents the optimum shape of a "neutral-sounding" spectrum having the same degree of speech communication or acoustical privacy as the spectrum being rated.

3. The third step is to plot the limits above the reference curve that cannot be exceeded by the noise spectrum being rated, in order to be classified as a neutral-sounding, subjectively inoffensive sound. The limits are +5 dB, for the 16 Hz through 500 Hz octave-bands, and +3 dB, for the 1000 Hz through 4000 Hz octave-bands.

4. The final step is to note any deviations in the noise spectrum that exceed the level of the reference curve. If the deviations do not exceed 5 dB in the octave-bands from 16 Hz to 500 Hz, nor 3 dB in the octave-bands from 1000 Hz to 4000 Hz, the spectrum is classified as "neutral", and the letter descriptor, (N), is appended to the numerical RC rating obtained in step one. However, if the deviations exceed 5 dB in the lower frequency range, the spectrum is classified as "rumbly" and assigned the letter descriptor "R". Conversely, if the deviations are in excess of 3 dB in the upper frequency range, the spectrum is classified as "hissy" and assigned the letter descriptor "H".

With regard to achieving occupant satisfaction, it is obviously desirable to obtain an "N" rating in the assessment of sound quality. Should the spectrum receive an "R" or "H" rating, a potential for occupancy complaints exists. As a general rule, rumble and hiss complaints are likely if the levels of the spectrum exceed the reference curve by more than 5 dB or 3 dB, respectively [15].

4.1.3. A-Weighted Sound Level

The A-weighted sound level is a single number measure of the relative loudness, of noise that is used extensively in outdoor environmental noise standards. The rating is expressed as a number followed by dBA, for example, 40 dBA. Maximum allowable background noise levels in Turkish Noise Legislation i.e., Gürültü Kontrol Yönetmeliği, are specified in terms of A-weighted sound levels.

A-weighted sound levels can be measured with simple sound level meters. The ratings correlate well with human judgments of relative loudness but take no account of spectral balance or sound quality. Thus many different sounding spectra can result in the same numeric value but have quite different subjective qualities.

4.1.3.1. Procedure for Determining dBA Rating

Corrections for the A-weighting network are listed in Table 4.1 [34].

Table 4.1 A-weighting Network Corrections in dB [34]

A-weighted correction values, dB **Octave band center frequency, Hz**							
63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
-26.2	-16.1	-8.6	-3.2	0	1.2	1.0	-1.1

Once the spectra are corrected by arithmetic addition of the correction values to the spectra, i.e., band pressure levels under search, the linear level is calculated using;

$$L_{A_t} = 10\log_{10}\sum_{i=1}^{8} 10^{L_{A_i}/10} \tag{4. 1}$$

where L_{A_i} is termed A-weighted band levels dropping pressure off the phrase since they are no longer associated with physical pressures.

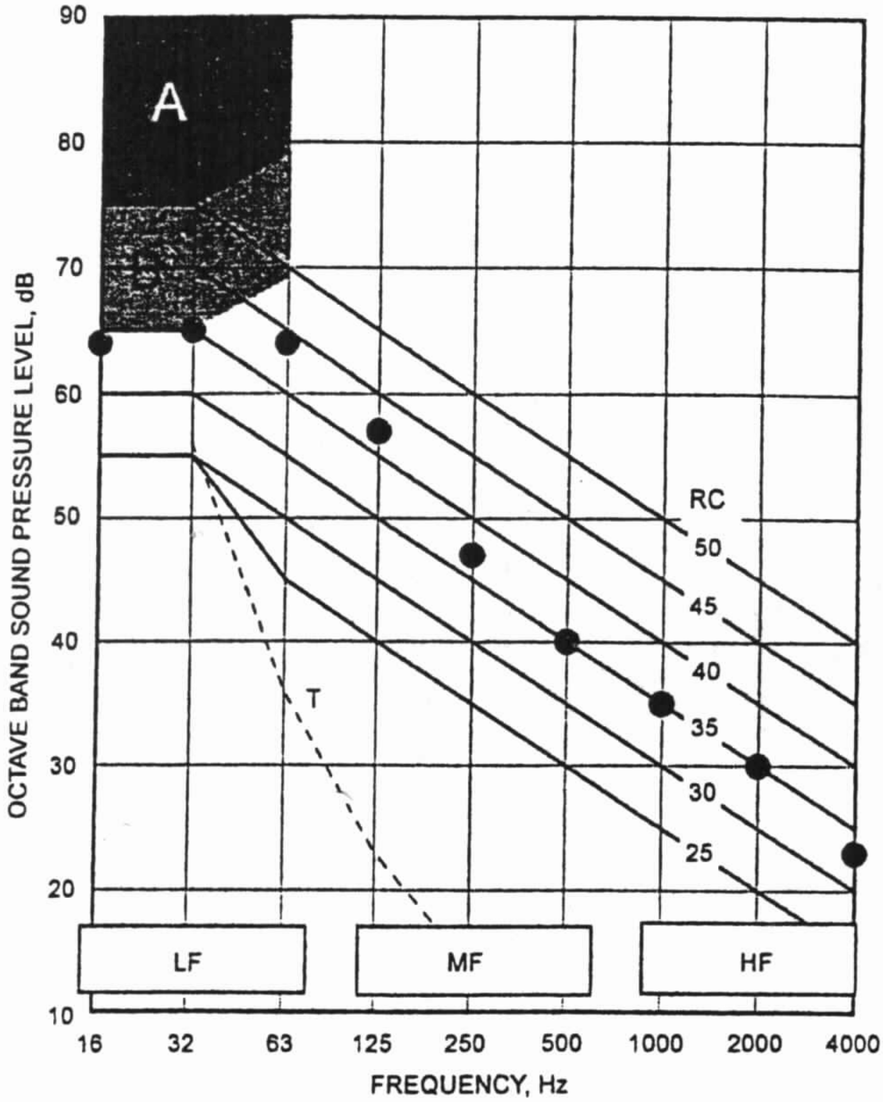
4.1.4. RC Mark II Room Criteria Method

Based on experience and the findings from ASHRAE sponsored research, the RC method was revised to the RC Mark II method [35]. Like its predecessor, the RC Mark II method is intended for rating the sound performance of an HVAC system as a whole. The method can also be used as a diagnostic tool for analyzing noise problems in the field. The RC Mark II is more complicated to use than the RC method.

The RC Mark II method has three parts; firstly, a family of criterion curves as shown in Figure 4.1, secondly, a procedure for determining the RC numerical rating and the noise spectral balance (quality), and finally a procedure for estimation of occupant satisfaction when the spectrum does not have the shape of an RC curve (Quality Assessment Index) [36].

The rating is expressed as RC followed by a number and a letter, for example, RC 35 (N). The number is the arithmetic average rounded to the nearest integer of the sound pressure levels in the 500, 1000, and 2000 Hz octave bands (the principal speech frequency region). The letter identifies the perceived character of the sound: (N) for neutral, (LF) for low frequency rumble, (MF) for mid frequency roar, and (HF) for high frequency hiss. There are also subcategories of the low frequency descriptor: (LFB), denoting a moderate but perceptible degrees of sound induced ceiling/wall vibration, and (LFA), denoting a noticeable degree of sound induced vibration.

Each reference curve in Figure 4.1 identifies the shape of an neutral, bland-sounding spectrum, indexed to a curve number corresponding to the sound level in the 1000 Hz octave band. Regions A and B denote levels at which sound can induce vibration in light wall and ceiling construction that can potentially cause rattles in light fixtures, furniture, etc. Curve T is the octave band threshold of hearing.



Noise levels in region B are likely to generate perceptible vibration in light walls and ceilings. Rattles are a slight possibility in light fixtures, doors, windows, etc. Noise levels in region A have a high probability of generating easily perceptible noise induced vibration in light walls and ceilings. Audible rattling in light fixtures, doors, windows, etc. is likely. Regions LF, MF, and HF are explained in the text. The solid dots are sound pressure levels for Example 5.

Figure 4.1 Room Criterion Curves, Mark II [5]

Table 4.2 Comparison of Sound Rating Methods [5]

Method	Overview	Considers speech interference	Evaluate Sound Quality	Components currently rated by method
NC	Can rate components. No quality assessment. Does not evaluate low frequency rumble.	Yes	No	Air terminals Diffusers
RC	Used to evaluate systems. Should not be used to evaluate components. Can be used to evaluate sound quality. Provides some diagnostic capability.	Yes	Yes	
dBA	Can be determined using sound level meter. No quality assessment. Frequently used for outdoor noise ordinances.	Yes	No	Cooling towers Water chillers Condensing units
NCB	Can rate components. Some quality assessment.	Yes	Yes	
RC Mark II	Evaluates sound quality. Provides improved diagnostic capabilities.	Yes	Yes	

4.1.4.1. Procedure for Determining RC Mark II Rating

1. Determine the appropriate RC reference curve by obtaining the arithmetic average of the sound levels in the 500, 1000, 2000 Hz octave bands. The RC reference curve is chosen to be that which has the same value at 1000 Hz as the calculated average value.

2. Assign a subjective quality by calculating the quality assessment index (QAI) . The QAI is a measure of the deviation of the shape of the spectrum under evaluation to the shape of the RC reference curve. The procedure requires calculation of **energy-averaged** spectral deviations from the RC reference curve in each of three frequency groups: low frequency, (LF) (16-63 Hz), medium frequency MF (125-500 Hz), an high frequency HF (1000-4000 Hz). (A simple arithmetic average of these deviations is usually adequate for evaluating HVAC sounds.) The procedures for the LF, MF and HF regions are given by Equation (4. 2), Equation (4. 3) and Equation (4. 4), respectively.

$$L_{lf} = 10\log\left[\left(10^{0.1\Delta L_{16}} + 10^{0.1\Delta L_{31.5}} + 10^{0.1\Delta L_{63}}\right)/3\right] \tag{4. 2}$$

$$L_{mf} = 10\log\left[\left(10^{0.1\Delta L_{125}} + 10^{0.1\Delta L_{250}} + 10^{0.1\Delta L_{500}}\right)/3\right] \tag{4. 3}$$

$$L_{hf} = 10\log\left[\left(10^{0.1\Delta L_{1000}} + 10^{0.1\Delta L_{2000}} + 10^{0.1\Delta L_{4000}}\right)/3\right] \tag{4. 4}$$

where the ΔL terms are the differences between the spectrum being evaluated and the RC reference curve in each frequency band. In this way, three specific spectral deviation factors, expressed in dB with either positive or negative values, are associated with the spectrum being rated. QAI is the range in dB between the highest and lowest values of the spectral deviation factors.

If QAI $\leq 5\,\text{dB}$, the spectrum is assigned a neutral (N) rating. If QAI exceeds 5 dB, the sound quality descriptor of the RC rating is the letter designation of the frequency of the deviation factor having the highest positive value.

4.1.4.1.1. Estimating Occupant Satisfaction Using QAI

The quality assessment index is useful in estimating the probable reaction of an occupant when the system does not produce optimum sound quality. The basis for estimating occupant satisfaction is that changes in sound level less than 5 dB do not cause subjects to change their ranking of sounds of similar spectral content. However, level changes greater than 5 dB do significantly affect subjective judgments. A QAI of 5 dB or less corresponds to a generally acceptable condition, provided that the perceived level of the sound is in a range consistent with the given type of space occupancy as recommended in Table 4.4. An exception to this rule occurs when the sound pressure levels in the 16 Hz or 31 Hz octave bands exceed 65 Hz. In such cases, the potential for acoustically induced vibration in typical light office construction should be considered. If the levels in these bands exceed 75 dB, a significant problem with induced vibration is indicated.

A QAI that exceeds 5 dB, but is less than or equal to 10 dB, represents a marginal situation in which occupant acceptance is questionable. However, a QAI greater than 10 dB will likely be objectionable to the average occupant. Table 4.3 lists sound quality descriptors and QAI values and relates them to probable occupant reaction to the noise.

Table 4.3 Definition of Sound-Quality Descriptor and Quality-Assessment Index (QAI) to aid in Interpreting RC Mark II Ratings of HVAC Related Sound [5]

Sound-quality descriptor	**Description of subjective perception**	**Magnitude of QAI**	**Probable Occupant Evaluation, Assuming level of specified criterion is not exceeded**
(N) Neutral Band	Balanced sound spectrum, no single frequency range dominant.	$QAI \le 5dB, L_{16}, L_{31} \le 65$ $QAI \le 5dB, L_{16}, L_{31} > 65$	Acceptable Marginal
(LF) Rumble	Low frequency range dominant (16 to 63 Hz).	$5dB < QAI \le 10dB$ $QAI > 10dB$	Marginal Objectionable
(LFVB) Rumble, with moderately perceptible room surface vibration	Low frequency range dominant (16 to 63 Hz).	$QAI \le 5dB, 65 < L_{16}, L_{31} < 75$ $5dB < QAI \le 10dB$ $QAI > 10dB$	Marginal Marginal Objectionable

Table 4.3 (cont.)

Sound-quality descriptor	**Description of subjective perception**	**Magnitude of QAI**	**Probable Occupant Evaluation, Assuming level of specified criterion is not exceeded**
(LFVA) Rumble, with moderately perceptible room surface vibration	Low frequency range dominant (16 to 63 Hz).	$QAI \leq 5dB, L_{16}, L_{31} > 75$ $5dB < QAI \leq 10dB$ $QAI > 10dB$	Marginal Marginal Objectionable
(MF) Roar	Mid frequency range dominant (125 to 500 Hz).	$5dB < QAI \leq 10dB$ $QAI > 10dB$	Marginal Objectionable
(HF) Hiss	High frequency range dominant (1000 to 4000 Hz).	$5dB < QAI \leq 10dB$ $QAI > 10dB$	Marginal Objectionable

Undoubtedly situations occur in the assessment of HVAC related noise where the numerical part of the RC rating is less than the specified maximum for the space use, but the sound quality descriptor is other than the desirable (N). For example, a maximum of RC 40 (N) is specified, but the actual noise environment turns out to be RC 35 (MF). There is not much evidence available in this area to decide which spectrum is preferable.

Even at moderate levels, if the dominant portion of the background noise occurs in the low frequency region, Perrson-Waye [37] observed that some people experience a sense of oppressiveness or depression in the environment. In such situations, the basis for complaint may result from an exposure to that environment for several hours, and may not be noticeable during a short exposure period.

4.1.5. Guideline Criteria for HVAC Related Background Sound in Rooms

Table 4.4 and Table 4.5 list design guidelines for HVAC related background sound appropriate to various occupancies. Perceived loudness and task interference are factored into the numerical part of the RC rating. Sound quality assumes the preferred design target is a neutral-sounding (N) spectrum, although some spectrum in balance is likely tolerable in certain limits.

If these levels are used as a basis, for contractual requirements, the following additional information must be provided [5].

1. What sound levels will be measured (specify L_{eq} or L_{max} levels, etc. in each octave frequency band).

2. Where and how the sound levels will be measured (specify space average over a defined area or specific points, etc.).

3. What instruments will be used to make the sound measurements (specify ANSI Type 1 or Type 2 sound level meters with octave band filters).

4. How the instrument used for the sound measurements will be calibrated.

5. How the sound measurement results will be interpreted.

Unless the above five points are clearly stipulated, the specified sound criteria may be unenforceable.

When applying the level specified in Table 4.4 as a basis for design, sound from non-HVAC sound sources, such as traffic or office equipment, may be the lower limit for sound levels in a space.

Table 4.4 Design Guidelines for HVAC Related Background Sound in Rooms [5]

Room Types	**RC(N);** $QAI \leq 5dB$ **Criterion in dB**
Residences, apartments, condominiums	25-35
Hotels/Motels	
Individual rooms or suites	25-35
Meeting/banquet rooms	25-35
Corridors, lobbies	35-45
Service/support areas	35-45
Office buildings	
Executive and private offices	25-35
Conference rooms	25-35
Tele-conference rooms	25 (max)
Open-plan offices	30-40
Corridors and lobbies	40-45
Hospitals and clinics	
Private rooms	25-35
Wards	30-40
Operating rooms	25-35
Corridors and public areas	30-40
Performing art spaces	
Drama theatres	25 (max)
Concert and recital halls	below RC 30
Music teaching studio	25 (max)
Music practice rooms	35 (max)
Laboratories (with fume hoods)	
Testing/research, minimal speech communication	45-55
Research, extensive telephone use,	
Speech communication	40-50
Group teaching	35-45

Table 4.4 (cont.)

Room Types	**RC(N);** $QAI \leq 5dB$ **Criterion, in dB**
Church, mosque, synagogue	
General assembly	25-35
with critical music programs	
Schools	
Classrooms up to 70 m2	40 (max)
Classrooms over 70 m2	35 (max)
Large lecture rooms, without speech amplification	35 (max)
Libraries	30-40
Courtrooms	
Unamplified speech	25-35
Amplified speech	30-40
Indoor stadiums, gymnasiums	
Gymnasiums and natatoriums	40-50
Large seating capacity spaces	
with speech amplification	45-55

Table 4.5 Design Guidelines for Background Sound in Rooms According to the Turkish Legislation [38]

Room Types	**Acceptable Sound Pressure Levels Leq (dBA)**
Private Spaces	
Theatres	25
Conference rooms	30
Hotel rooms	30
Hotel restaurants	35
Hospitals	
Hospitals	35
Residences	
Bedrooms (Urban)	35
Living rooms (Countryside)	40
Living rooms (Suburb)	45
Living rooms (Urban)	60
Kitchens, Bathrooms	70
Schools	
Classrooms, laboratories	45
Gymnasiums, service/support areas	60
Office Buildings	
Executive and private offices	50
Open-plan offices	60
Industrial Buildings	
Factories (Small Scale)	70
Factories (Large Scale)	80

CHAPTER 5

IMPLEMENTATION OF THE SOFTWARE

In this section; introduction of the software that is written within the scope of the study, general software structure, and implementation of HVAC elements used in the software are explained giving different user-interface samples.

5.1. INTRODUCTION OF THE SOFTWARE

This software is developed to predict the sound level passing through the HVAC system and reaching building occupant's ears. The name of the program is chosen as "**HVAC Noise Prediction (HNP)**" so that it reflects the purpose of the program. HNP projects equipment sound power data through the surrounding (e.g., ductwork, ceiling) to estimate the sound level that will be heard. Industry-standard calculations published by ASHRAE's 1991 Algorithms, 1999 Sound and Vibration Control for HVAC Acoustics handbook, and ARI's Standard 885 for estimating occupied space sound levels in the application of air terminals and air outlets are the basis for this estimate. The programming language is Visual Basic 6.0.

In HNP, user can model the conditions of an HVAC system by choosing specific equipment and building component criteria. HNP analyzes the sound path and calculate the total effect for an enclosed space. User can continuously adjust the elements used in the program and compare the results effortlessly. HNP even outputs presentation quality graphs of NC or RC levels.

HNP provides the user a windows-based format means that easy - to - use windows environment and double-click functionality.

When multiple paths are involved (i.e., duct breakout, return system radiated), HNP not only determines the overall sound level at the receiver, but also how much of that sound each path contributes. İt defines the descriptions of equipment and building components that generate, attenuate, reduce or regenerate sound. From the user's input, HNP models the sound level that will be perceived by the building's occupants. It quickly compares the results using presentation quality graphs. HNP has a built-in graph function that plots the analysis results on an NC or RC chart.

The input of the program is the description of sound paths from HVAC equipment and systems through ductwork, and ceilings. The output of the program is the analysis results that can be viewed on screen either as as series of detailed tables or as plots on an NC or RC chart with HNP's built-in graphing function.

Evaluating the total effect of sound in an enclosed space requires many mathematical equations and necessity of looking up many tables. Solving those equations and looking up many tables manually takes hours of precious design time and is prone to error. HNP streamlines this analysis task with easy-to-use menus and dialog boxes that help the user create pictorial diagrams of sound paths. As path elements are added or deleted, HNP dynamically recalculates the resulting sound power levels; and when multiple paths are involved, HNP not only determines the overall sound level at the receiver, but also how much of that sound each path contributes. Moreover; as paths are added or deleted to the sums, in order to determine the total contribution of the paths on sums, HNP even dynamically calculates the resulting sound power levels.

5.2. GENERAL SOFTWARE STRUCTURE

HNP uses the “Source-Path-Receiver” model to estimate sound levels. In this model, the “source” is the sound-generating device, the “receiver” is typically the environment where a person hears that sound, and the “path” comprises eveything that affects the sound as it travels from the source to the receiver. The term elements collectively describe the source, path and receiver components.

HNP models each user-selected-element individually. For proper analysis, the order in which the user selected these elements should reflect the direction that sound travels from the source to the receiver.

What’s heard at the receiver location is the sum of all sound traveling to that location. Depending on the application, there may be several sources of sound, and the sound from each source may travel to the receiver along one or more paths. Regardless of the number of sound sources and paths, HNP analyzes each path individually; then calculates the overall result using the equations and tables as explained in 5.1.

HNP’s graphical user interface helps to learn how to predict noise via HNP. A new acoustical analysis can be started when the user clicks on HNP’s file menu and selects the **New** command. The main HNP screen now contains three subsidiary windows as seen in Figure 5.1.

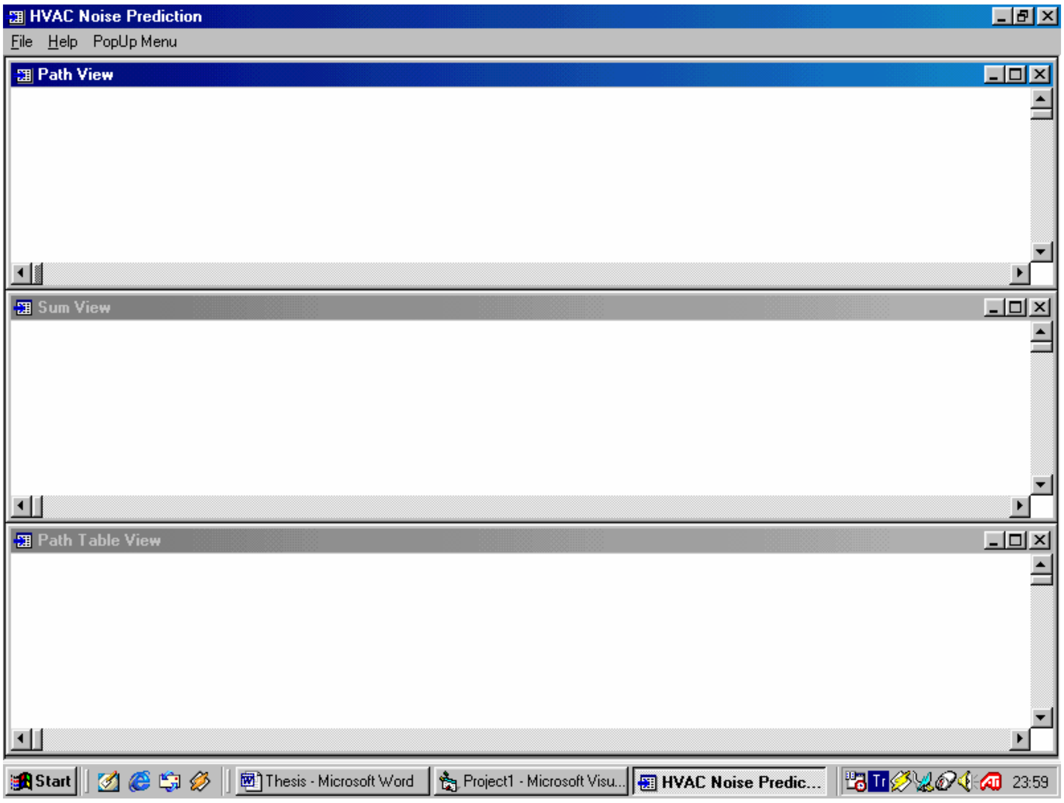


Figure 5.1 Main HNP Screen with "Path View", "Sum View" and "Path Table View" Windows

The **Path View** window is where the user will select icons representing the the source, path components and receiver from the **Path View PopUp Menu**, as seen in Figure 5.2, and arrange them to depict each noise path.

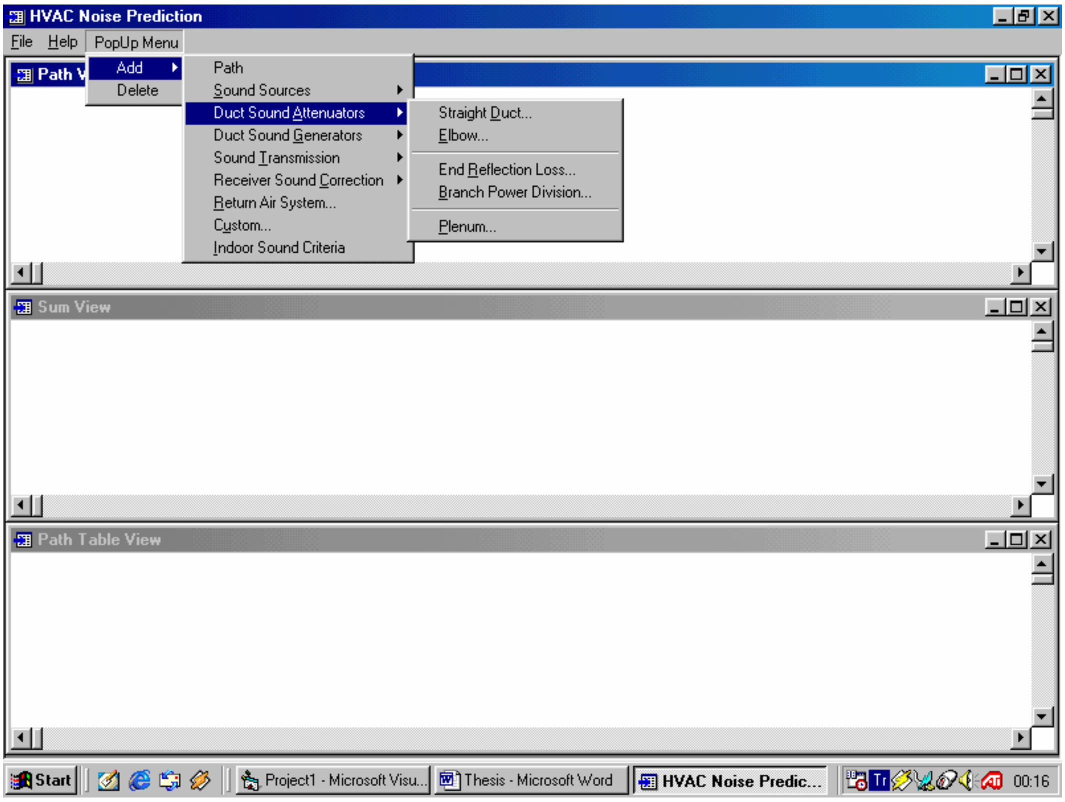


Figure 5.2 Main HNP Screen with Partially Extracted "Path View" PopUp Menu Window

Other HNP menus available when this window is active are: **File**, **Help** and **PopUp Menu**.

After the user has constructed the path (s), he/she will use the **Sum View** window to add them together. That is simply a matter of creating a "Sum header" there using the **Sum View PopUp Menu**, as seen in Figure 5.3; then "dragging" each path from the Path View window and dropping on the newly created Sum header.

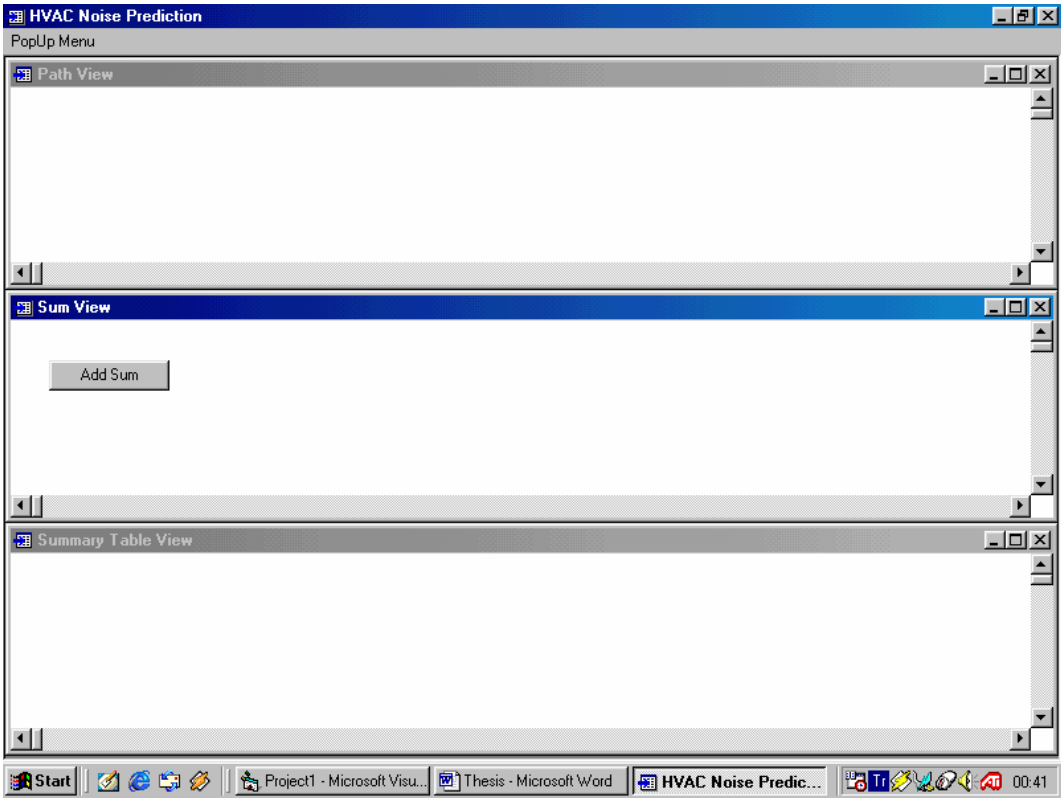


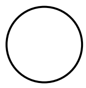
Figure 5.3 Main HNP Screen with Partially Extracted "Sum View" PopUp Menu

Other HNP menu available when this window is active is: **PopUp Menu**.

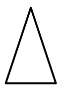
HNP automatically calculates the acoustical performance of the active path as the user adds, modifies or removes elements in Path View or Sum View, and displays the results in Path Table View or Summary Table View windows. When Path View is activated, Path Table View is seen, and when Sum View is activated, Summary Table View is seen on the screen, as seen in Figure 5.2 and Figure 5.3 respectively. This feature enables the user to track the results of both Path View and Sum View, whatever he/she prefers.

The user has the opportunity to display the resulting RC and NC curves for any of the path or the sum created by adding "**Indoor Sound Criteria**" to the end of each path or the sum chosen to be displayed.

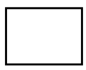
The essence of the HVAC Noise Prediction Program lies under Path View PopUp Menu, where the user select the source, path and receiver entities. Figure 5.4 "maps" the commands provided in that menu. To add an entity, user just chooses the appropriate PopUp Menu command and completes the entries in the corresponding dialog box. The added elements via Path View PopUp Menu are seen in the Path View by a symbol. The shape of the circumferential box declares the type of the element as source, path or receiver. The shapes and the corresponding meanings are stated below.



sound source



sound attenuator



receiver correction

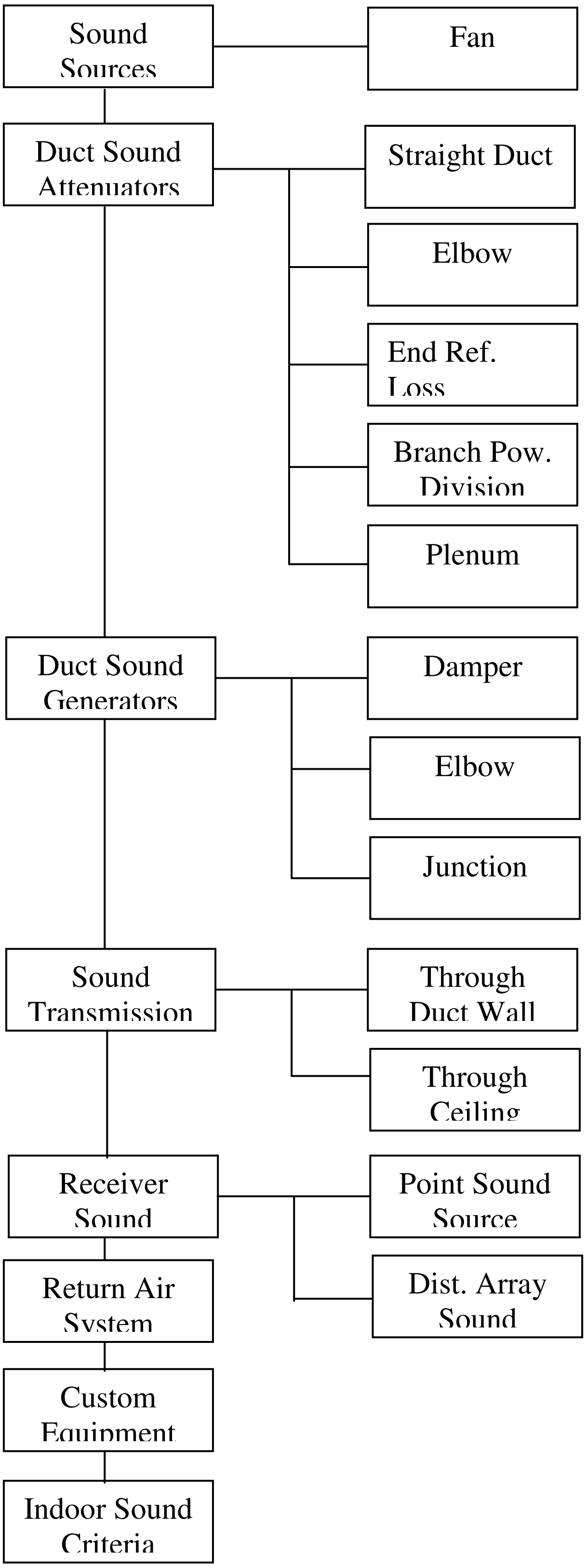


Figure 5.4 HNP's PopUp Menu Structure

5.3. IMPLEMENTATION OF HVAC ELEMENTS IN THE SOFTWARE

The simple scenario given in this section helps to introduce the implementation of HVAC elements in the software and gives an idea about the processing of the HNP. Indeed, detailed case studies are elaborated in Chapter 6. This section is only a demonstration of implementation of HVAC elements in the HNP software.

Scenario: A forward-curved fan is selected to supply 4.2 m^3/s at 70 Pa. Sized for efficient operation, it has 24 blades and operates at 1170 rpm. What is the fan's estimated sound power level at the fan discharge and what is the rating of this level? [6]

In this simple scenario, sound emanating from the fan (the "source") travels along only one path to someone (the "receiver"). The path consists of just two elements: the sound source itself and the indoor sound criteria for the rating of the sound reached the receiver.

To model this scenario, after starting the program, the user;

1. With the Path View window active, right clicks on the form, chooses **Add**, then **Path**. Following, the program adds a "Path" icon to the Path View.

2. Right clicks again, but now on the Path icon, chooses **Add**, then **Sound Sources** and, finally **Fan**, (as seen in Figure 5.5).

3. In the resulting dialog box, chooses "Centrifugal-Forward Curved" as the fan type, and inputs "4.2 m^3/s", "70 Pa", "24" and, "1170 rpm" as "fan volume flow rate", "fan pressure", "number of

impeller blades", and "rpm of the fan", respectively, (as seen in Figure 5.6).

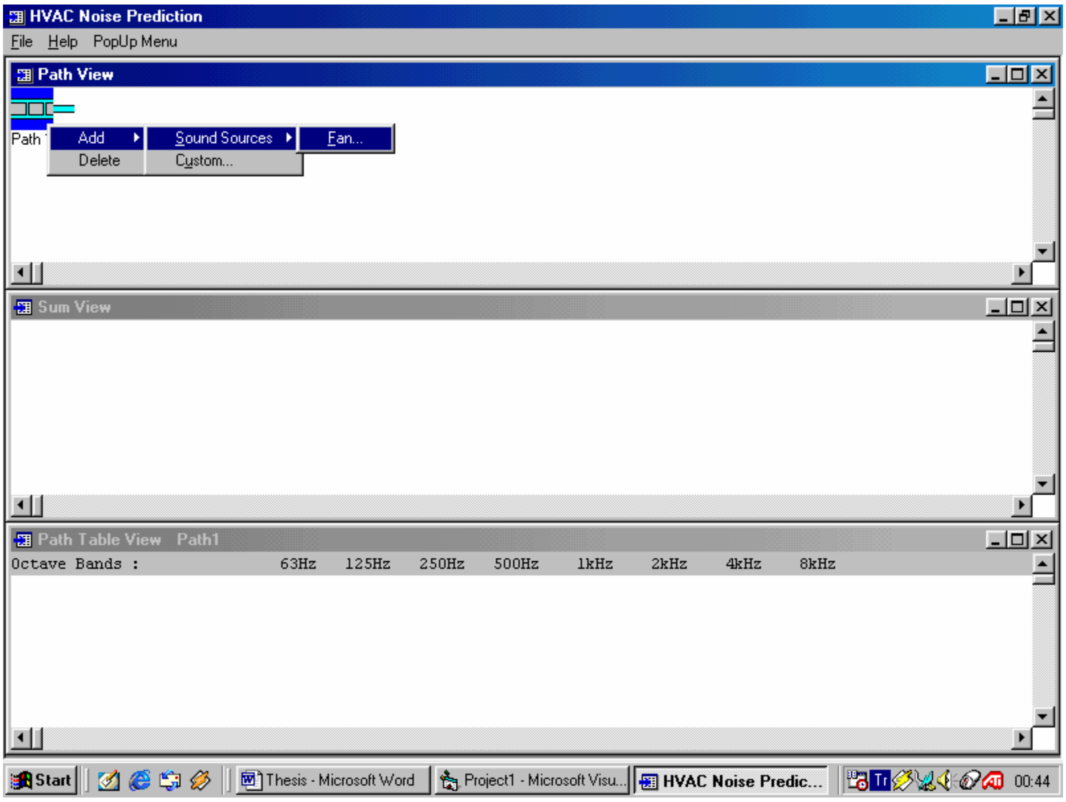


Figure 5.5 A Snap Shot from the HNP Program during the Addition of Fan Element to a Path

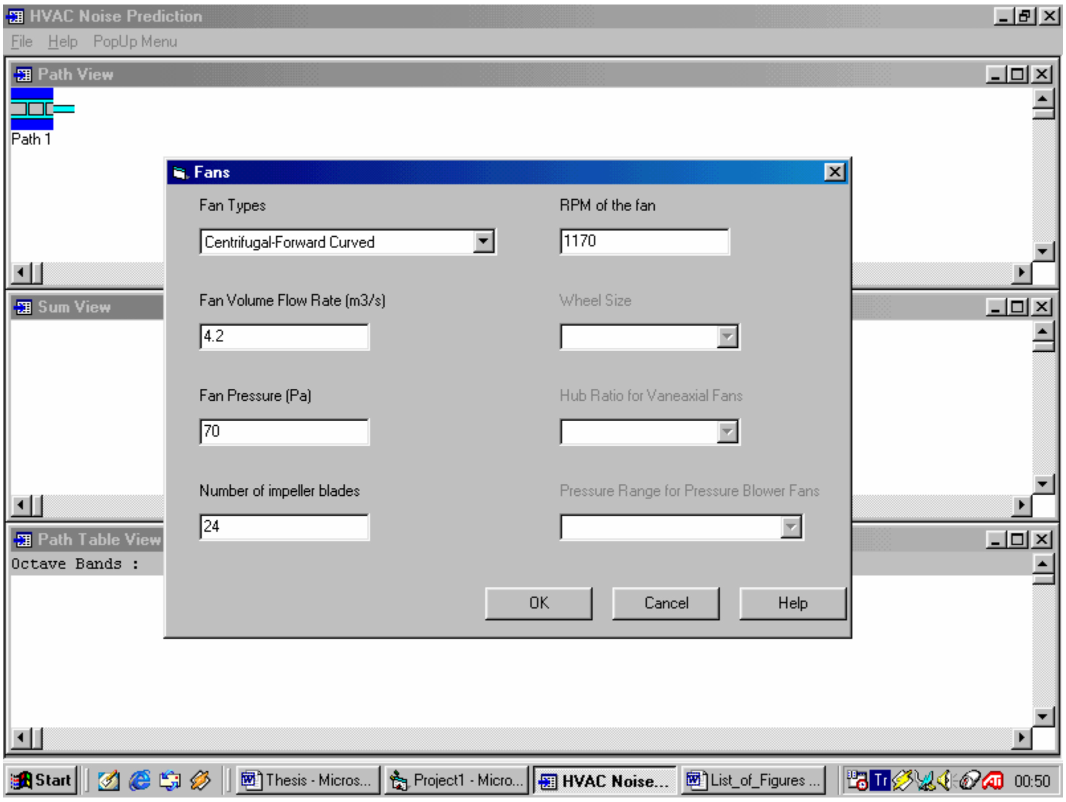


Figure 5.6 A Snap Shot from the HNP Program while Filling the Necessary Fields in the Dialog Box for Fan Element Calculation

When the dialog box closes, HNP automatically adds a fan icon and related header of it (Fan) to the path already created. HNP also updates the Path Table View to show the addition of this element and its impact on the resulting sound level.

4. Clicks on the PopUp Menu and chooses this time **Indoor Sound Criteria**. HNP puts the related icon to the end of the path. It should be noted that, Indoor Sound Criteria should always be put as the last element at the end in order to complete the related Path or Sum.

Following the addition of Indoor Sound Criteria, HNP automatically evaluates the resulting noise rating according to four different noise rating procedure comprising of **NC Rating**, **RC Rating**, **dBA Rating**, and **RC MarkII Rating**, respectively. HNP shows the results of all these ratings on the Path Table View as seen in Figure 5.7.

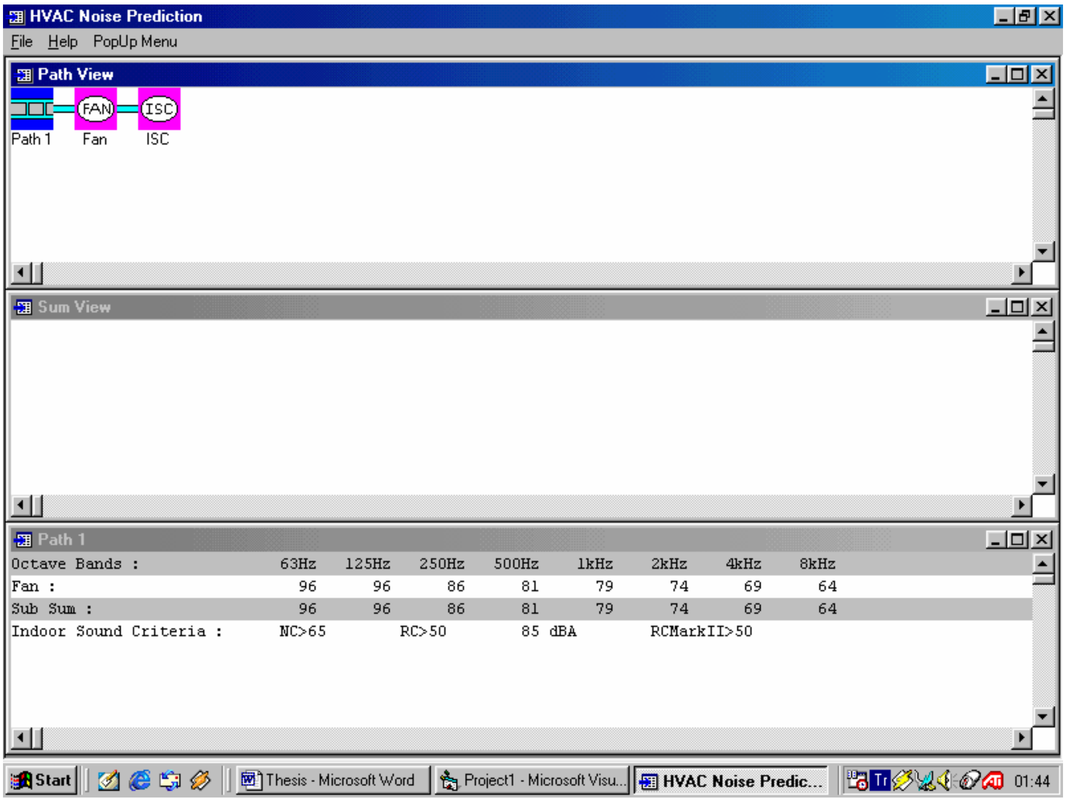


Figure 5.7 A Snap Shot from the HNP Program with a Completed Path Showing the Final Rating Results

Initial HNP analysis is now complete. When the combined effect of multiple paths are investigated, a sum object is created with its icon and header (Sum 1), and all paths to be observed are dragged one by one to this sum icon to see the combined effect of all these paths on the receiver. For instance, just for demonstration purposes, a sum is created and our pre-calculated path is dragged over this sum, and the resulting HNP screen is shown at Figure 5.8. HNP also updates the Summary Table View to show the addition of this path and its impact on the resulting sound level. Again, Indoor Sound Criteria is put as the last element to the end of the sum in order to rate the final sound according to the procedures stated above. HNP shows the results of all these ratings on the Summary Table View as seen in Figure 5.8.

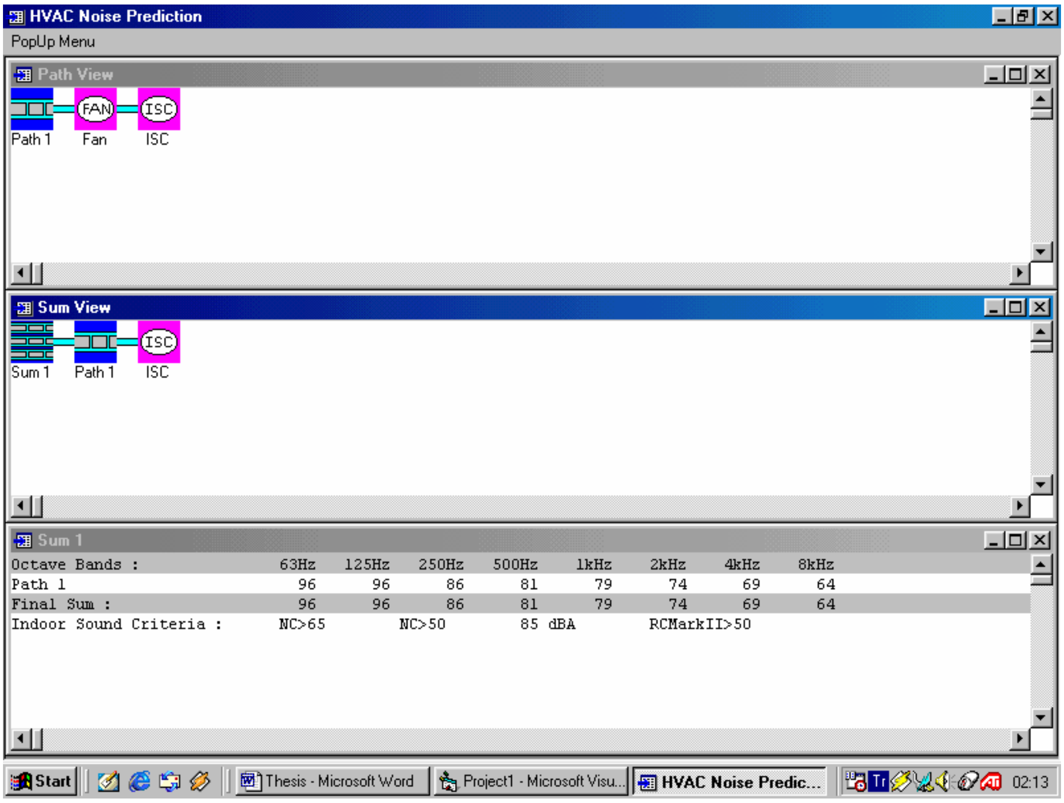


Figure 5.8 A Snap Shot from the HNP Program with a Completed Sum Showing the Final Rating Results

Evaluating The Results: Reviewing the data occured in Path Table View and Summary Table View (since one path is added to the sum, the ratings are the same with the path exists at the Path View), it can be said that HNP sums the overall sound pressure by octave bands in the range of 63 to 8000 Hz, and calculates the overall NC, RC, dBA and RC MarkII values. In this scenario, the path analysis predicts an overall rating of NC>65, RC>50, 85 dBA, and RC MarkII>50 which means considerably high rating values. However, a fan is generally not used in a HVAC system by itself in real life, but accompanied with another attenuator elements in the system. So, it is expected to have lower rating values in a real HVAC system to be compatible with the standards. On the other hand, the resulting noise level of the fan can be decreased by lowering the fan volume flow rate, for instance. HNP gives user the opportunity to enter the new flow rate value and quickly see the recent change in the resultant noise level. Hence, the user every time has the chance to optimize HVAC system by changing the parameters via dialog boxes and to try reaching the desired design criteria easily in terms of overall expected noise levels of the HVAC system.

CHAPTER 6

CASE STUDIES

In addition to the knowledge about the HNP program given in Chapter 5, three different cases are elaborated in this section gradually from a simple case to more complicated and true-to-life one so as to demonstrate the capabilities and structure of the program in different manners. The scenarios are chosen so that different rating values in total are obtained like “neutral”, “hissy” and “rumble” tonal spectrums.

6.1. CASE 1

Scenario: The Variable Air Volume (VAV) box situated above a conference room produces sound that reaches the occupants below. That sound moves along two paths: it travels with supply air through the diffuser along the discharge-airborne path, and it passes directly from the VAV box through the ceiling along the casing-radiated path. To determine the overall sound level in the conference room, each path is modeled individually; then the results are summed [39].

First, discharge-airborne path is modeled. It consists of the discharge sound power of the VAV box (the “source”), 1.82 m of lined round duct, a diffuser and the room correction factor (the “receiver”). As stated in the preceding chapter, each element must be added to the path separately and in the direction of sound travel.

HNP doesn't include sound data for VAV boxes. To model that element of the path, Custom entry is used. The following cataloged unit data is typed in Custom dialog box.

Custom – Generator in dB:

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
65	65	61	58	56	48	42	42

The next element is the duct. The following data is entered in the appropriate sections of "Straight Duct" dialog box of "Duct Sound Attenuators" sub-menu of "PopUp" main-menu.

Duct:

Duct Type: Circular/Lined

Inside Diameter: 200 mm

Duct Length: 1.82 m

Lining Thickness: 50 mm

The following element is the diffuser element to add the path. Since manufacturer's specific data is used generally as the diffuser sound data, HNP doesn't include sound data for diffusers. To model that element of the path, Custom entry is used. The following cataloged unit data is typed in Custom dialog box.

Custom – Generator in dB:

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
31	36	39	40	39	36	30	30

Path is continued adding the room or "receiver" correction factor. The following data is entered in the appropriate sections of "Distributed Array of Ceiling Sound Sources" dialog box of "Receiver Sound Correction" sub-menu of "PopUp" main-menu.

Distributed Array of Ceiling Sound Sources:

Ceiling Height: 2.44 m

Number of evenly spaced outlets in the room: 2

Ratio of the floor area served by each outlet to the square of the ceiling height: $13.93/(2.44^2)=2.34$

Finally, the path is completed adding "Indoor Sound Criteria". Now, Path Table View displays the sum and NC, RC, dBA and, RC MarkII ratings for this path, as well as the octave band data for each of its elements, as seen in Figure 6.1.

As the secondary path, casing-radiated path is modeled. It consists of three elements: the radiated sound power of the VAV box (the "source"), the attenuating effect of the ceiling and the room correction factor (the "receiver").

The sound data related to the VAV casing radiated sound is again entered via Custom dialog box with the parameters given below.

Custom – Generator in dB:

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
54	54	49	45	39	33	27	27

Then, the second element at the second path is declared using "Through Ceiling" dialog box of "Sound Transmission" sub-menu of "PopUp" main-menu with the following parameters.

Transmission Ceiling:

Ceiling Tile Type: Mineral Fiber (16 mm, 4.9 kg/m^2)

Surface area of the buttom panel of the source closest to the ceiling tiles: 0.9 m^2

The third element in the path is the room or "receiver" correction factor. The following data is entered in the appropriate sections of "Point Sound Source" dialog box of "Receiver Sound Correction" sub-menu of "PopUp" main-menu.

Point Sound Source:

Room Volume: 2.44*4.58*6.1=68.168 m^3

Shortest distance from noise source to the receiver: 1.5 m

And, as usual, the path is completed adding "Indoor Sound Criteria". At the end, the discharge-airborne and casing-radiated path results are combined to determine the overall sound level in the conference room. This operation completes the HNP analysis for this scenario. The sum of Paths 1 and 2 is seen in the Summary Table View window activating the Sum View window by double-clicking on the Sum 1 icon, as seen in Figure 6.1.

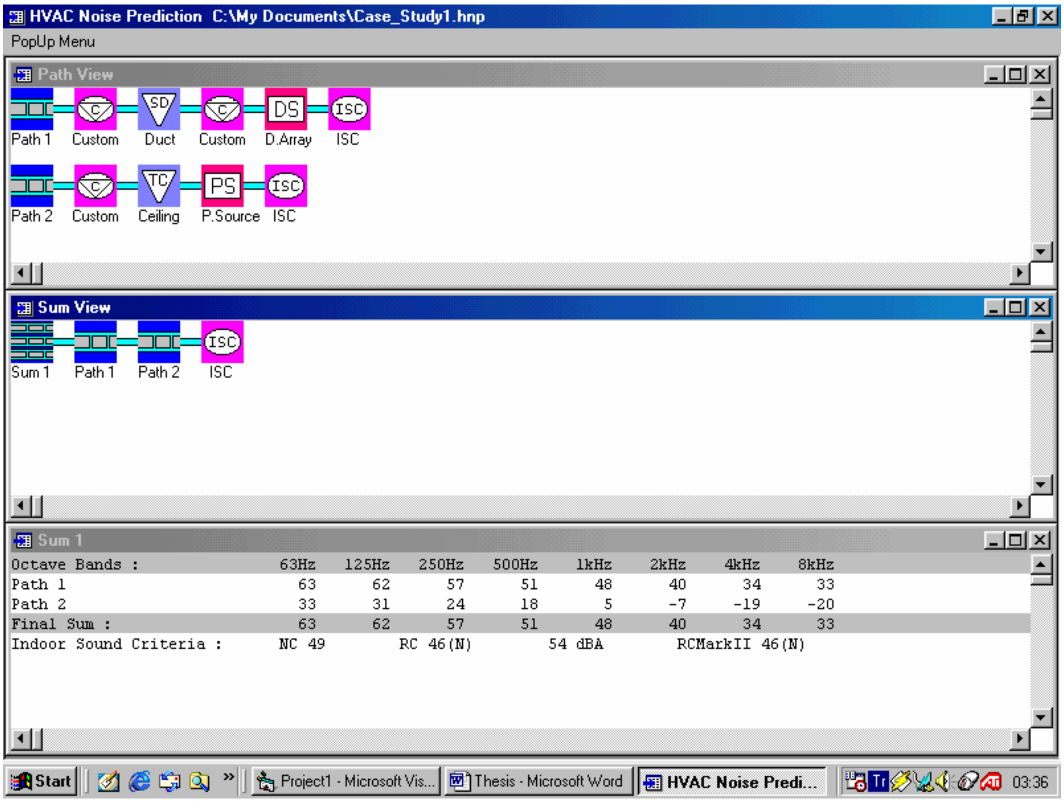


Figure 6.1 A Snap Shot from the HNP Program with a Completed Sum Showing the Case_Study1 Rating Results

Figure 6.2 and Figure 6.3 show the results of Path 1 and Path 2 with their rating results, respectively.

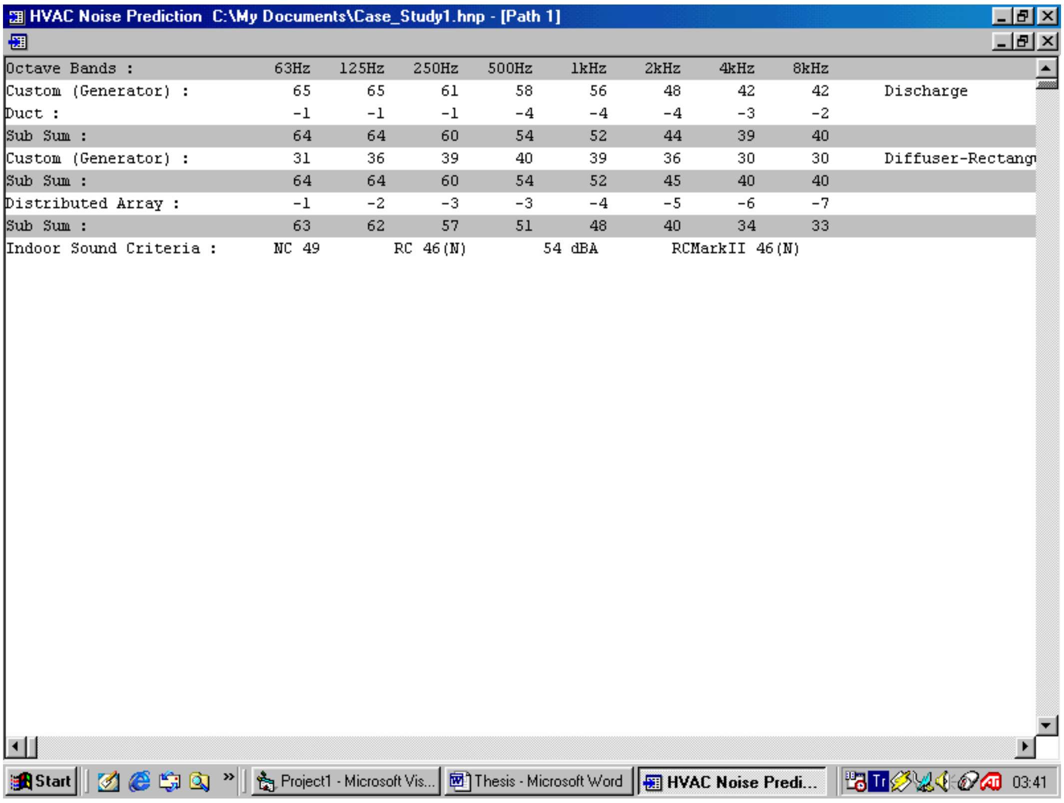
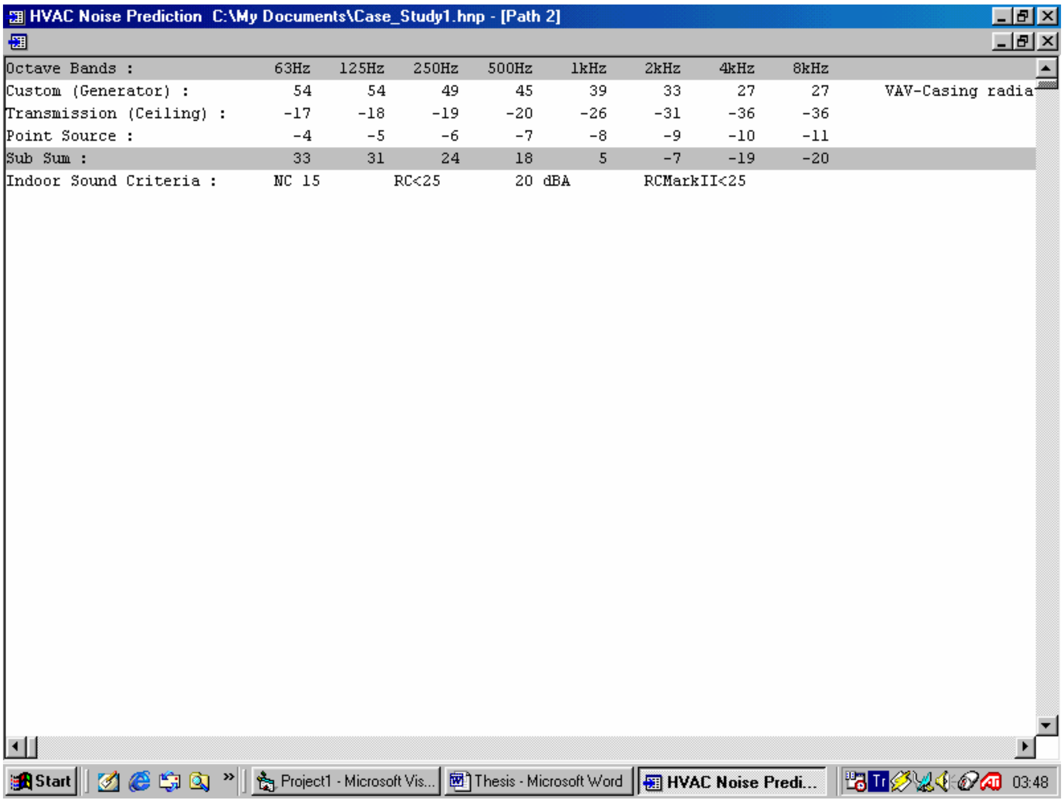


Figure 6.2 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study1, Path 1 Rating Results



HVAC Noise Prediction C:\My Documents\Case_Study1.hnp - [Path 2]

Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	54	54	49	45	39	33	27	27	VAV-Casing radia
Transmission (Ceiling) :	-17	-18	-19	-20	-26	-31	-36	-36	
Point Source :	-4	-5	-6	-7	-8	-9	-10	-11	
Sub Sum :	33	31	24	18	5	-7	-19	-20	
Indoor Sound Criteria :	NC 15		RC<25		20 dBA		RCMarkII<25		

Figure 6.3 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study1, Path 2 Rating Results

As it is seen from the results, it can be said that the discharge sound power of the VAV box has the greatest influence on the conference room's overall sound level. If the obtained results are compared with the permissible RC criterion stated in Table 4.4, it will be seen that although the tonal spectrum seems like neutral, RC 46(N), the overall rating of two paths (RC 46) is too higher than the permissible conference room criteria which is max RC 35. Consequently, there should be HVAC design changes in the system, either changing the VAV box with a quiter one, this is the possible one, or applying additional insulation measures on the duct work.

6.2. CASE 2

Scenario: A fan powered box/induction unit produces sound that reaches the occupants below. That sound moves along six paths:

1. Radiated and Induction Inlet
2. Duct Breakout
3. Distribution Duct Breakout
4. Flexible Duct Breakout
5. Discharge
6. Outlet Generated Sound

To determine the overall sound level in the occupant space, each path is modeled individually; then the results are summed. The acoustic model of the system is given in Figure 6.4 with specific dimensions for the scenario [15]. Table 6.1 provides a detailed list of six sound paths.

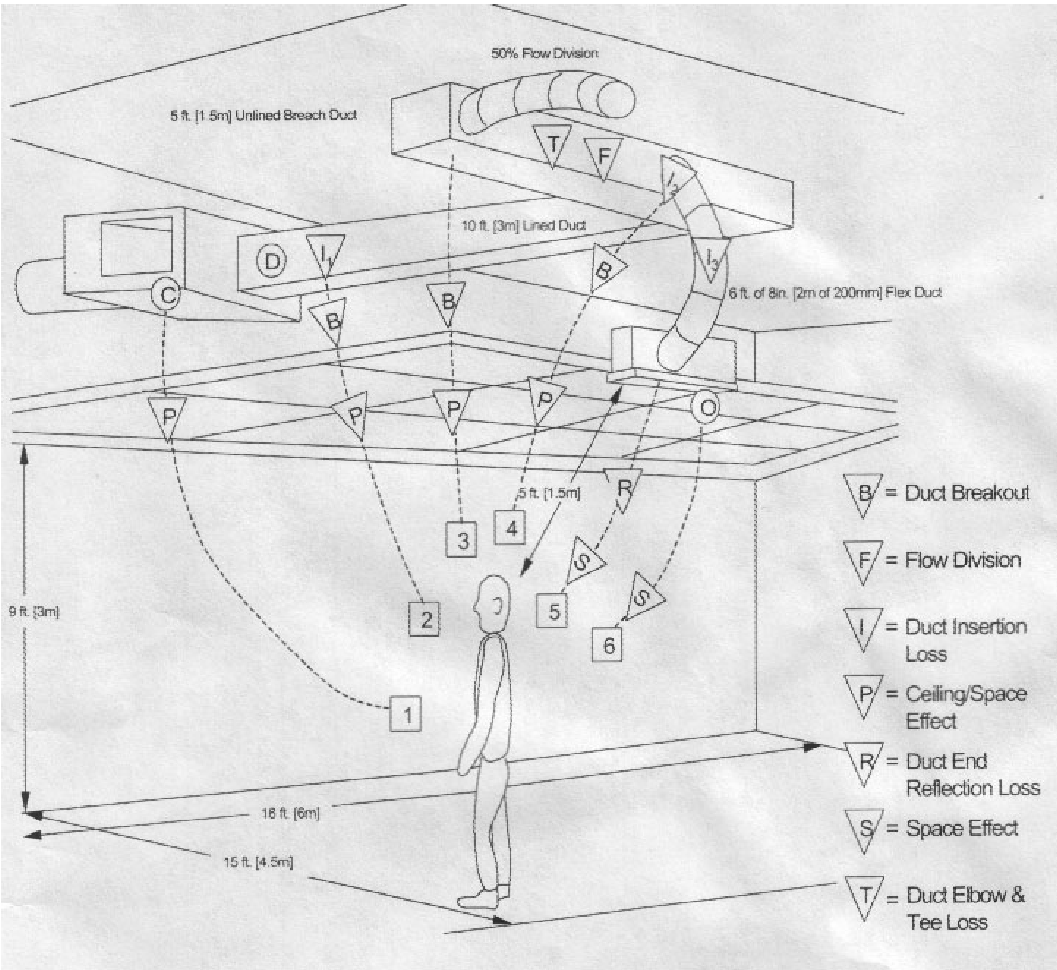


Figure 6.4 Fan-Powered Terminal or Induction Terminal-Case_Study2 Calculation Acoustic Model [15]

Table 6.1 Detailed Path List of the System Elaborated in Case_Study2 [15]

Path Number	Path Name	Path Attenuation Calculation
1	Radiation and Induction Inlet	Ceiling Effect
2	Duct Breakout Sound	- Duct Insertion Loss - Duct Breakout Transmission Loss - Ceiling Effect
3	Distribution Duct Breakout	- Duct Insertion Loss - Duct Elbow Loss - Branch Power Division - Duct Insertion Loss - Duct Breakout Transmission Loss - Ceiling Effect
4	Flexible Duct Breakout	- Duct Insertion Loss - Duct Elbow Loss - Branch Power Division - Duct Insertion Loss - Duct Insertion Loss - Duct Breakout Transmission Loss - Ceiling Effect
5	Discharge Sound	- Duct Insertion Loss - Duct Elbow Loss - Branch Power Division - Duct Insertion Loss - Duct Breakout Transmission Loss - End Reflection Factor - Space Effect
6	Outlet Generated Sound	Space Effect

The following data is entered in the appropriate sections of the related dialog boxes as demonstrated in Chapter 5 and similar to Case Study 1. All data related to the each of the elements of each six paths and steps of the calculation are stated below respectively.

Path 1:

Custom – Generator in dB: Radiated and induction inlet

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
64	64	60	57	58	55	52	52

Transmission Through Ceiling:

Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)

Surface area of the bottom panel of the source closest to the ceiling tiles: 0.9 m^2

Radiated path sound pressure level at receiver location and the rating of this path is given in Figure 6.5.

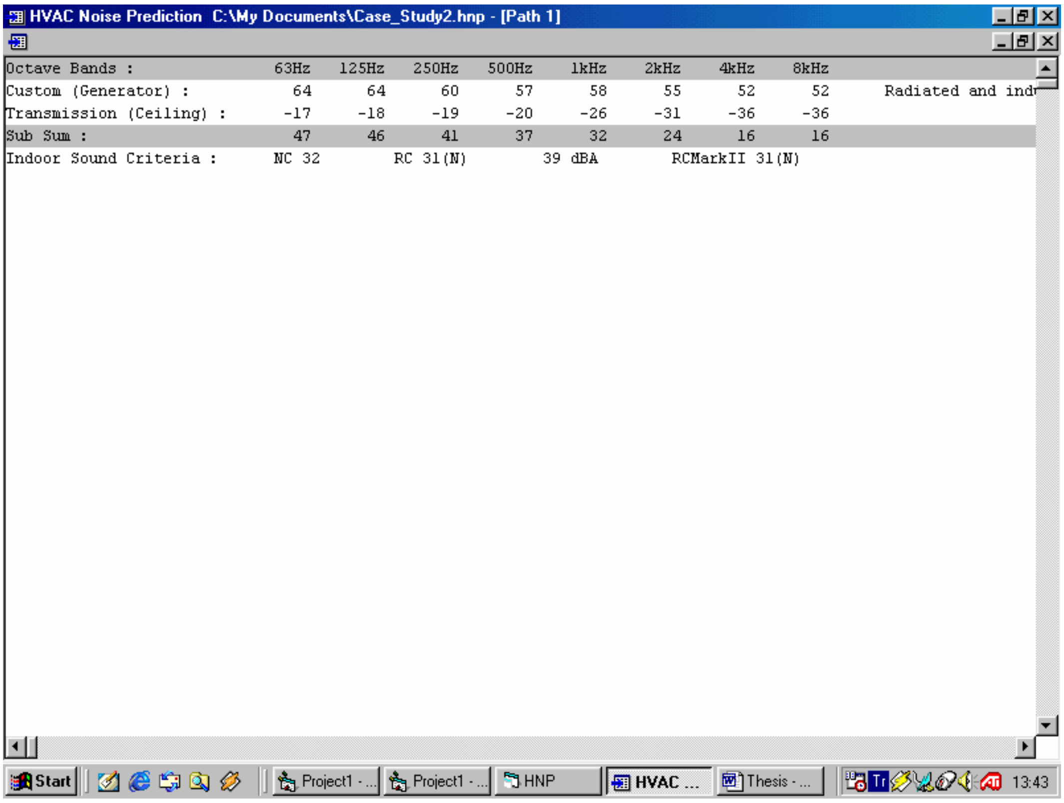


Figure 6.5 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 1 Rating Results

Path 2:

Custom – Generator in dB: Terminal discharge

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
66	66	65	62	62	62	60	60

Duct:

Duct Type: Rectangular/Lined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 1.5 m

Lining Thickness: 25 mm

External Insulation: No

Transmission Through Duct Wall:

Sound Attenuator: Chosen

Transmission Type: Breakout

Duct Type: Rectangular

Height: 300 mm

Width: 300 mm

Duct Length: 1.5 m

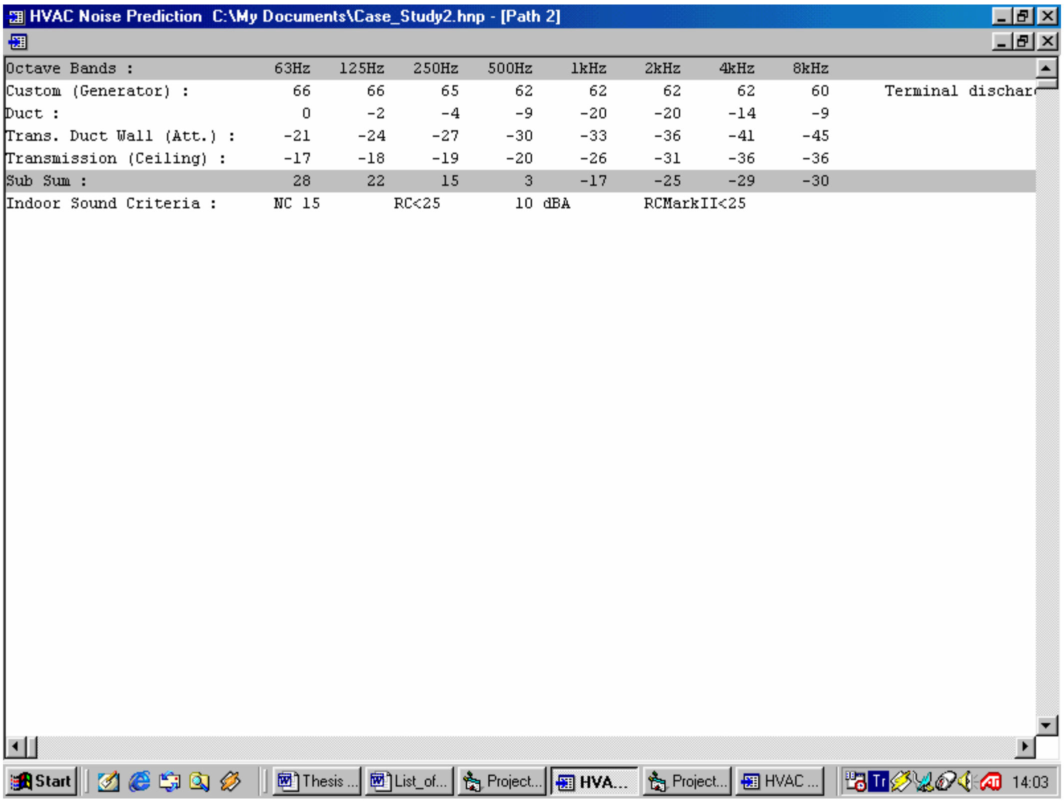
Mass/Unit Area of duct walls: 5 kg/m^2

Transmission Through Ceiling:

Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)

Surface area of the bottom panel of the source closest to the ceiling tiles: 0.9 m^2

Duct breakout path sound pressure level at receiver location and the rating of this path is given in Figure 6.6.



HVAC Noise Prediction C:\My Documents\Case_Study2.hnp - [Path 2]

Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	66	66	65	62	62	62	62	60	Terminal dischar
Duct :	0	-2	-4	-9	-20	-20	-14	-9	
Trans. Duct Wall (Att.) :	-21	-24	-27	-30	-33	-36	-41	-45	
Transmission (Ceiling) :	-17	-18	-19	-20	-26	-31	-36	-36	
Sub Sum :	28	22	15	3	-17	-25	-29	-30	
Indoor Sound Criteria :	NC 15		RC<25		10 dBA		RCMarkII<25		

Figure 6.6 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 2 Rating Results

Path 3:

Custom – Generator in dB: Terminal discharge

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
66	66	65	62	62	62	60	60

Duct:

Duct Type: Rectangular/Lined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 3 m

Lining Thickness: 25 mm

External Insulation: No

Elbow:

Elbow Type: Square/Unlined

Turning Vane: without

Elbow Width: 300 mm

Branch Power Division:

Branch Cross Sectional Area: pi*(300)^2/4 =70685.83 mm^2

Total Cross-Sectional Area Leaving The Take-off: 2*70685.83=141371.66 mm^2

Note: Since the branch power division is given as %50 power split in the scenario, branch cross sectional area is multiplied by two in order to find total cross-sectional area leaving the take-off.

Duct:

Duct Type: Rectangular/Unlined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 1.5 m

External Insulation: No

Transmission Through Duct Wall:

Sound Attenuator: Chosen

Transmission Type: Breakout

Duct Type: Rectangular

Height: 300 mm

Width: 300 mm

Duct Length: 3 m

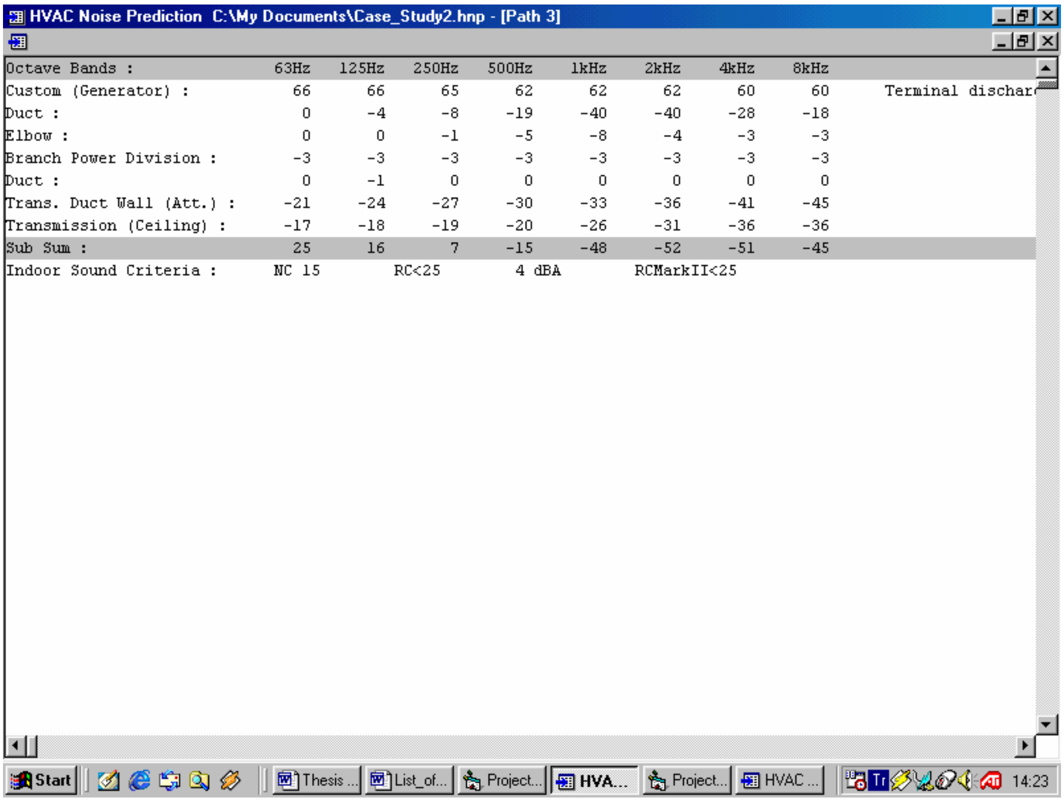
Mass/Unit Area of duct walls: 5 kg/m^2

Transmission Through Ceiling:

Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)

Surface area of the bottom panel of the source closest to the ceiling tiles: 0.9 m^2

Distribution duct breakout sound pressure level at receiver location and the rating of this path is given in Figure 6.7.



Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	66	66	65	62	62	62	60	60	Terminal dischar
Duct :	0	-4	-8	-19	-40	-40	-28	-18	
Elbow :	0	0	-1	-5	-8	-4	-3	-3	
Branch Power Division :	-3	-3	-3	-3	-3	-3	-3	-3	
Duct :	0	-1	0	0	0	0	0	0	
Trans. Duct Wall (Att.) :	-21	-24	-27	-30	-33	-36	-41	-45	
Transmission (Ceiling) :	-17	-18	-19	-20	-26	-31	-36	-36	
Sub Sum :	25	16	7	-15	-48	-52	-51	-45	
Indoor Sound Criteria :	NC 15	RC<25		4 dBA		RCMarkII<25			

Figure 6.7 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 3 Rating Results

Path 4:

Custom – Generator in dB: Terminal discharge

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
66	66	65	62	62	62	60	60

Duct:

Duct Type: Rectangular/Lined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 3 m

Lining Thickness: 25 mm

External Insulation: No

Elbow:

Elbow Type: Square/Unlined

Turning Vane: without

Elbow Width: 300 mm

Branch Power Division:

Branch Cross Sectional Area: pi*(300)^2/4 =70685.83 mm^2

Total Cross-Sectional Area Leaving The Take-off: 2*70685.83=141371.66 mm^2

Note: Since the branch power division is given as %50 power split in the scenario, branch cross sectional area is multiplied by two in order to find total cross-sectional area leaving the take-off.

Duct:

Duct Type: Rectangular/Unlined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 1.5 m

External Insulation: No

Duct:

Duct Type: Flexible/Lined

Inside Diameter: 200 mm

Duct Length: 0.9 m

Lining Thickness: 25 mm

External Insulation: No

Transmission Through Duct Wall:

Sound Attenuator: Chosen

Transmission Type: Breakout

Duct Type: Flexible

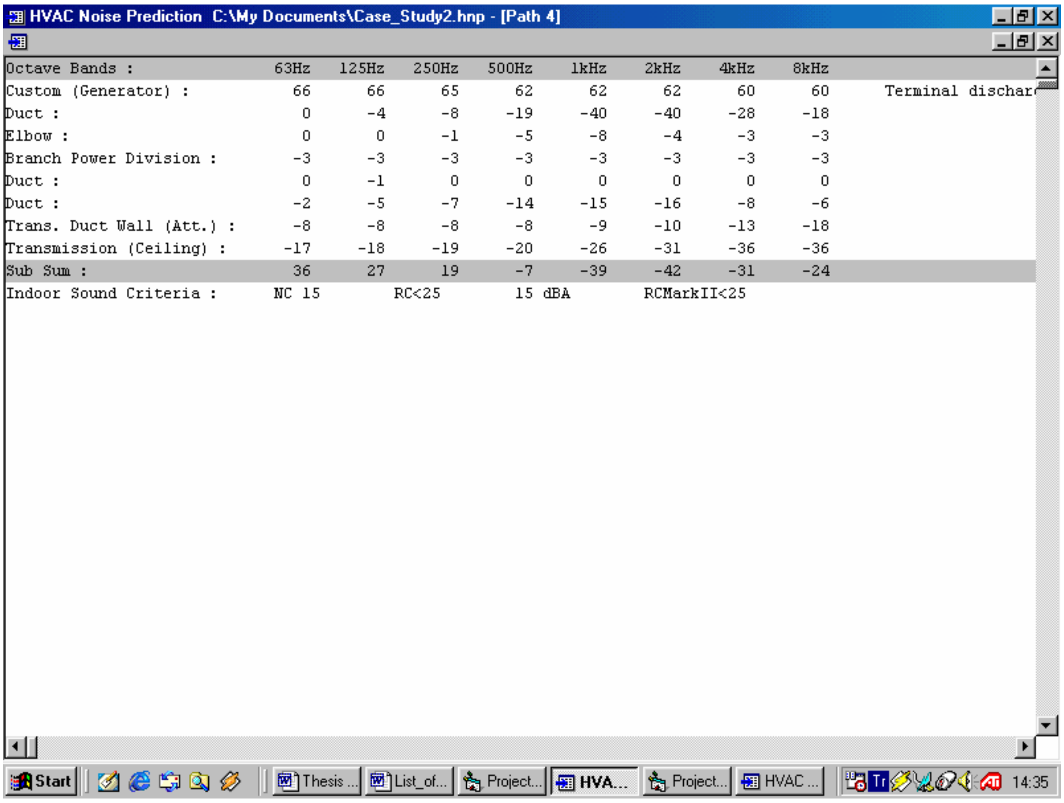
Diameter: 200 mm

Transmission Through Ceiling:

Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)

Surface area of the bottom panel of the source closest to the ceiling tiles: 0.9 m^2

Flexible duct breakout path sound pressure level at receiver location and the rating of this path is given in Figure 6.8.



Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	66	66	65	62	62	62	60	60	Terminal dischar
Duct :	0	-4	-8	-19	-40	-40	-28	-18	
Elbow :	0	0	-1	-5	-8	-4	-3	-3	
Branch Power Division :	-3	-3	-3	-3	-3	-3	-3	-3	
Duct :	0	-1	0	0	0	0	0	0	
Duct :	-2	-5	-7	-14	-15	-16	-8	-6	
Trans. Duct Wall (Att.) :	-8	-8	-8	-8	-9	-10	-13	-18	
Transmission (Ceiling) :	-17	-18	-19	-20	-26	-31	-36	-36	
Sub Sum :	36	27	19	-7	-39	-42	-31	-24	
Indoor Sound Criteria :	NC 15		RC<25		15 dBA		RCMarkII<25		

Figure 6.8 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 4 Rating Results

Path 5:

Custom – Generator in dB: Terminal discharge

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
66	66	65	62	62	62	60	60

Duct:

Duct Type: Rectangular/Lined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 3 m

Lining Thickness: 25 mm

External Insulation: No

Elbow:

Elbow Type: Square/Unlined

Turning Vane: without

Elbow Width: 300 mm

Branch Power Division:

Branch Cross Sectional Area: pi*(300)^2/4 =70685.83 mm^2

Total Cross-Sectional Area Leaving The Take-off: 2*70685.83=141371.66 mm^2

Note: Since the branch power division is given as %50 power split in the scenario, branch cross sectional area is multiplied by two in order to find total cross-sectional area leaving the take-off.

Duct:

Duct Type: Rectangular/Unlined

Inside Height: 300 mm

Inside Width: 300 mm

Duct Length: 1.5 m

External Insulation: No

Duct:

Duct Type: Flexible/Lined

Inside Diameter: 200 mm

Duct Length: 1.5 m

Lining Thickness: 25 mm

External Insulation: No

End Reflection Loss:

Duct Type: Round

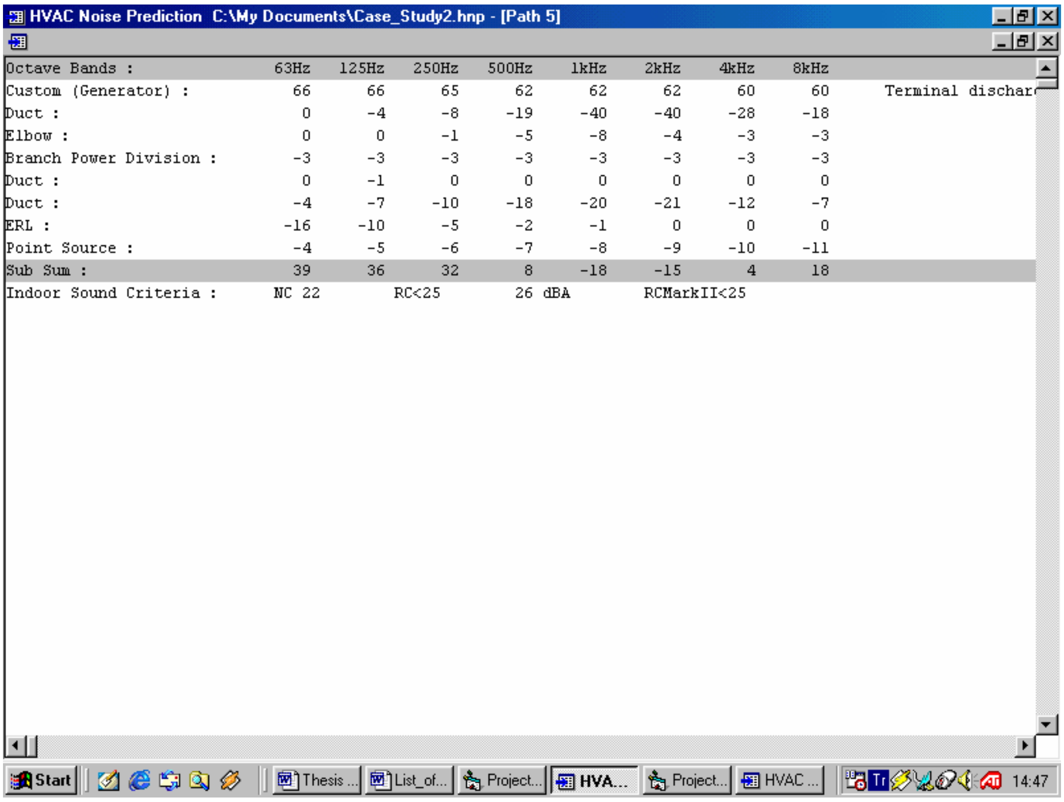
Duct Diameter: 200 mm

Receiver Correction: Point Sound Source

Room Volume: 66.7 m^3

Shortest Distance from Noise Source to Receiver: 1.5 m

Discharge sound pressure level at receiver location and the rating of this path is given in Figure 6.9.


HVAC Noise Prediction C:\My Documents\Case_Study2.hnp - [Path 5]

Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	66	66	65	62	62	62	60	60	Terminal dischar
Duct :	0	-4	-8	-19	-40	-40	-28	-18	
Elbow :	0	0	-1	-5	-8	-4	-3	-3	
Branch Power Division :	-3	-3	-3	-3	-3	-3	-3	-3	
Duct :	0	-1	0	0	0	0	0	0	
Duct :	-4	-7	-10	-18	-20	-21	-12	-7	
ERL :	-16	-10	-5	-2	-1	0	0	0	
Point Source :	-4	-5	-6	-7	-8	-9	-10	-11	
Sub Sum :	39	36	32	8	-18	-15	4	18	
Indoor Sound Criteria :	NC 22	RC<25		26 dBA		RCMarkII<25			

Figure 6.9 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 5 Rating Results

Path 6:

Custom – Generator in dB: Outlet Generated

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
40	40	43	46	46	44	42	42

Receiver Correction: Point Sound Source

Room Volume: 66.7 m^3

Shortest Distance from Noise Source to Receiver: 1.5 m

Outlet generated sound pressure level at receiver location and the rating of this path is given in Figure 6.10.

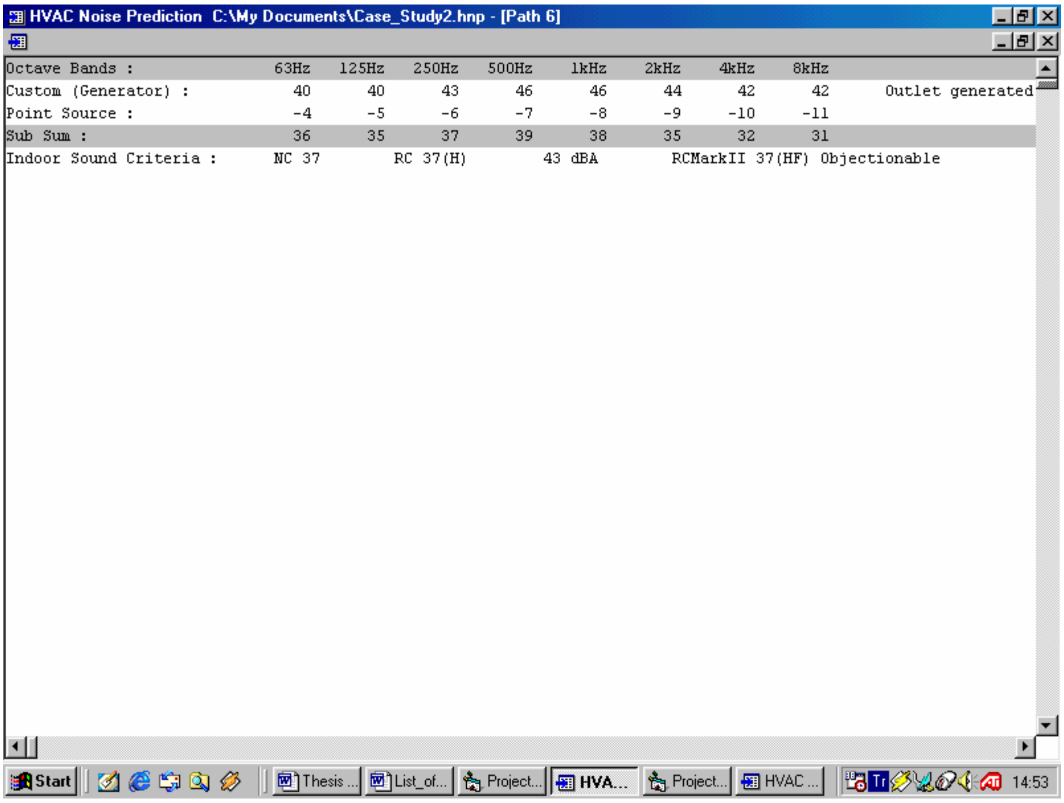


Figure 6.10 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study2, Path 6 Rating Results

At the end, the contributions of the six individual paths are combined to obtain the total sound pressure level at the receiver location. This operation completes the HNP analysis for this scenario. The sum of Paths 1, 2, 3, 4, 5 and 6 is seen in the Summary Table View window activating the Sum View window by double-clicking on the Sum 1 icon, as seen in Figure 6.11. On the other hand, the full view of the Path View of this scenario is given in Figure 6.12 to provide the whole path structure.

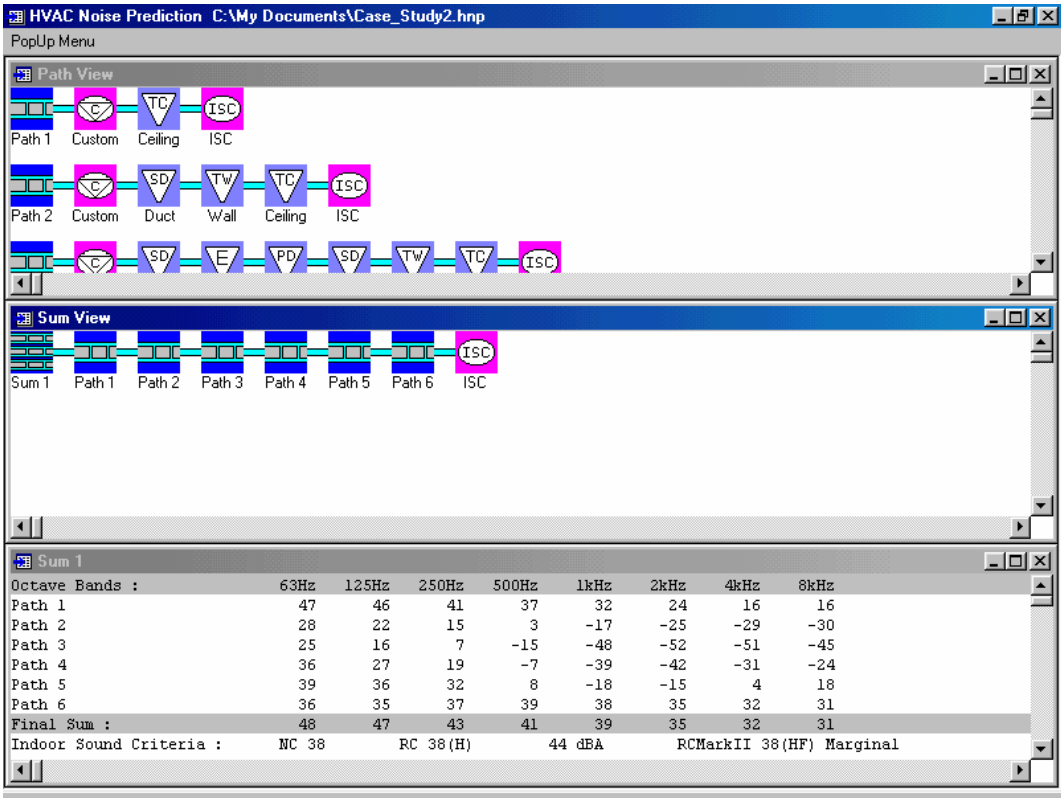


Figure 6.11 A Snap Shot from the HNP Program with a Completed Sum Showing the Case_Study2 Rating Results

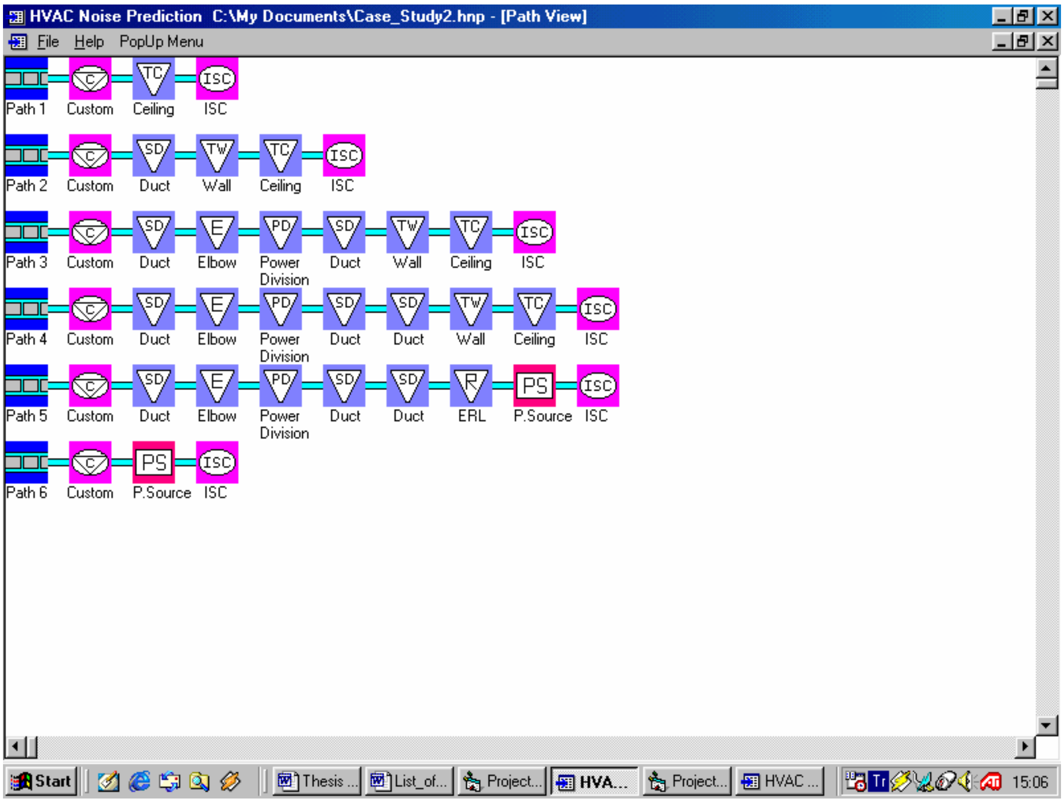


Figure 6.12 A Snap Shot from the HNP Program with a Full View of the Path View of Case_Study2

It can be seen from the results that the critical paths are Casing Radiated (Path 1), Discharge (Path 5) and Outlet Generated (Path 6). The ratings of the overall sound pressure level of the six paths are NC 38, RC 38 (H), 44 dBA and RC MarkII 38 (HF) according to the four different rating procedures. RC procedure points out that the tonal spectrum of the resultant sound behaves hissy characteristic and RC MarkII method supports this conclusion showing that there is a marjinal hissy spectrum at high frequency region, i.e. (1000-4000 Hz). When the values stated in Table 4.4 are compared to the ratings of Case_Study2, it is seen that the obtained overall sound level is permitted for open-plan offices, corridors, lobbies, wards, and public areas, for instance.

6.3. CASE 3

Scenario: Air is supplied to the HVAC system in this case by the rooftop unit shown in Figure 6.13. The receiver room is directly below the unit. The room has the following dimensions: length= 6100 mm, width=6100 mm; and height=2750 mm. This scenario assumes the roof penetrations for the supply and return air ducts are well sealed and there are no other roof penetrations. The supply side of the rooftop unit is ducted to a Variable Air Volume (VAV) terminal control unit that serves the room in question. A return air grille conducts air to a common ceiling return air plenum. The return air is then directed to the rooftop unit through a short rectangular return air duct [5].

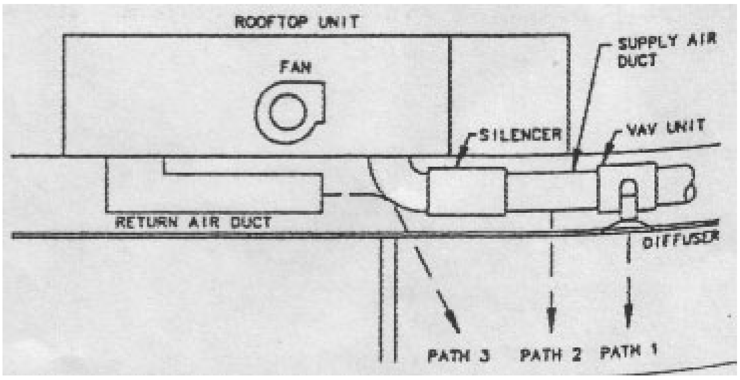


Figure 6.13 Sound Paths Layout for Case_Study3 [5]

The following three sound paths are examined:

Path 1: Fan airborne supply air sound that enters the room from the supply air system through the ceiling diffuser.

Path 2: Fan airborne supply air sound that breaks out through the wall of the main supply air duct into the plenum space above the room.

Path 3: Fan airborne return air sound that enters the room from the inlet of the return air duct.

The sound power levels associated with the supply air and return air sides of the fan in the rooftop unit are specified by the manufacturer as follows:

Sound Power Levels in dB	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
Rooftop supply air = 3.3 m^3/s at 620 Pa	92	86	80	78	78	74	71	71
Rooftop return air = 3.3 m^3/s at 620 Pa	82	79	73	69	69	67	59	59

Paths 1 and 2 are associated with the supply air side of the system. The main duct is a 560 mm diameter, 0.551 mm, unlined, round sheet metal duct. The flow volume in the main duct is 3.3 m^3/s. The silencer after the radiused elbow is a 560 mm diameter by 1.12 m long, high pressure, circular silencer.

The branch junction that occurs 2.44 m from the silencer is a 45^0 wye. The branch duct between the main duct and the VAV control unit is a250 mm diameter, unlined, round sheet metal duct. The flow volume in the branch duct is 0.37 m^3/s.

The straight section of duct between the VAV control unit and the diffuser is a 250 mm diameter, unlined round sheet metal duct. The diffuser is 380 mm by 380 mm square. Typical distance between the diffuser and a listener in the room is assumed as 1.5 m.

With regard to the duct breakout sound associated with the main duct, the length of the duct that runs over the room is 6.1 m. The ceiling of the room is comprised of 6.1 m by 1.22 m by 16 mm lay-in-ceiling tiles that have a surface density of 2.9 to 3.4 kg/m^2. The ceiling has integrated lighting and diffusers.

Path 3 is associated with the return air side of the system. The rectangular return air duct is lined with 50 mm thick, 48 kg/m^3 density fiberglass duct liner. For the return air path, typical distance between the inlet of the return air duct and a listener is assumed as 3 m.

Figure 6.14 and Figure 6.15 shows a layout of the part of the supply air system and return air system respectively that is associated with the receiver room.

The following data is entered in the appropriate sections of the related dialog boxes as demonstrated in Chapter 5 and similar to Case Studies 1 and 2. All data related to the each of the elements of each three paths and steps of the calculation are stated below respectively. In this case, descriptive numbers are given to each element, immediately after the name of each element, in the calculation in accordance with the Figure 6.14 and Figure 6.15 to provide easiness to trace the calculation steps of the case study.

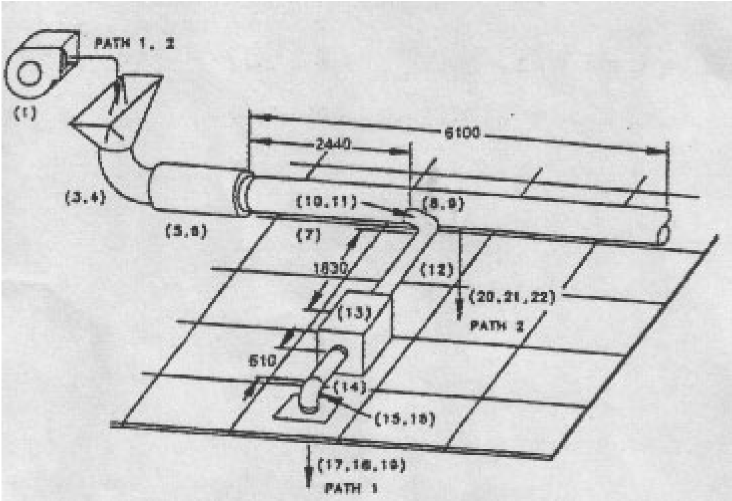


Figure 6.14 Supply Air Portion Layout for Case_Study3 [5]

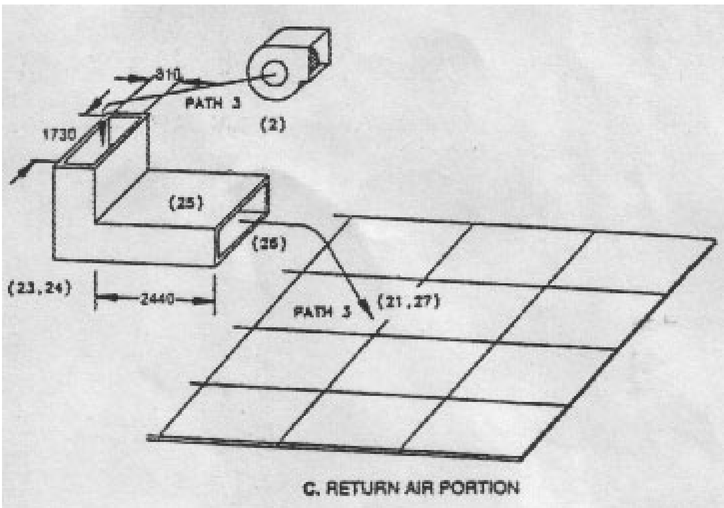


Figure 6.15 Return Air Portion Layout for Case_Study3 [5]

Path 1:

Custom - Generator (1) in dB: Fan-Supply Air, 3.3 m^3/s, 620 Pa

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
92	86	80	78	78	74	71	71

Elbow (3):

Elbow Type: Round

Elbow Width: 560 mm

Elbow-Generator (4): 90^0 bend w/o turning vanes, 320 mm radius

Turning Vane: without

Pressure Drop Across Elbow: 620 Pa

Volume Flow Rate: 3.3 m^3/s

Cross Sectional Area of the Elbow: $pi*(0.56)^2/4 = 0.2463\ m^2$

Height of the Elbow: 0.56 m

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	3	5	10	21	42	84	167	334
K (Char. Spectrum) in dB	0	-6	-14	-21	-29	-35	-35	-35
Rounding Correction in dB	0	0	0	0	0	0	0	0

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 320/560 = 0.57

Custom - Attenuator (5) in dB: 560 mm dia. by 1.12 m high pressure silencer

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
4	7	19	31	38	38	27	27

Custom - Generator (6) in dB: Regenerated noise from above silencer

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
68	79	69	60	59	59	55	55

Duct (7):

Duct Type: Circular/Unlined

Inside Diameter: 560 mm

Duct Length: 2.44 m

External Insulation: No

Branch Power Division (10):

Branch Cross Sectional Area: $pi*(250)^2/4$ =49087.38 mm^2

Total Cross-Sectional Area Leaving The Take-off:

$pi*(250)^2/4 + pi*(560)^2/4 = 295388.2493$

Junction (11): Duct 90^0 branch take-off, 50 mm radius

Volume Flow Rate in Main Duct: 3.3 m^3/s

Volume Flow Rate in Branch Duct: 3.3 m^3/s

Cross-sectional Area of the Main Duct: pi*$(0.56)^2/4$ = 0.2463

Cross-sectional Area of the Branch Duct: pi*$(0.25)^2/4$ = 0.04908

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	2	4	8	17	33	66	133	265
K (Char. Spectrum) in dB	15	11	4	-3	-9	-18	-29	-35
Rounding Correction in dB	0	0	0	0	0	0	0	0
Turbulence Correction in dB	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 50/250 = 0.2

Note 3: HNP automatically calculates Velocity Factor value, and the user enters the Upstream Turbulence Correction iaw the Velocity Factor value. In this case, Velocity Factor value is $M=U_M/U_B = 2$.

Duct (12):

Duct Type: Circular/Unlined

Inside Diameter: 250 mm

Duct Length: 1.83 m

External Insulation: No

Custom - Attenuator (13) in dB: Terminal volume register unit

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
0	-5	-10	-15	-15	-15	-15	-15

Duct (14):

Duct Type: Circular/Unlined

Inside Diameter: 250 mm

Duct Length: 0.61 m

External Insulation: No

Elbow (15):

Elbow Type: Round

Elbow Width: 250 mm

Elbow-Generator (16): 90^0 bend w/o turning vanes, 50 mm radius

Turning Vane: without

Pressure Drop Across Elbow: 620 Pa

Volume Flow Rate: 0.37 m^3/s

Cross Sectional Area of the Elbow: $pi*(0.25)^2/4 = 0.04908$ m^2

Height of the Elbow: 0.25 m

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	2	4	8	17	33	66	133	265
K (Char. Spectrum) in dB	3	5	-11	-19	-26	-34	-35	-35
Rounding Correction in dB	0	0	0	0	0	0	0	0

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 50/250 = 0.2

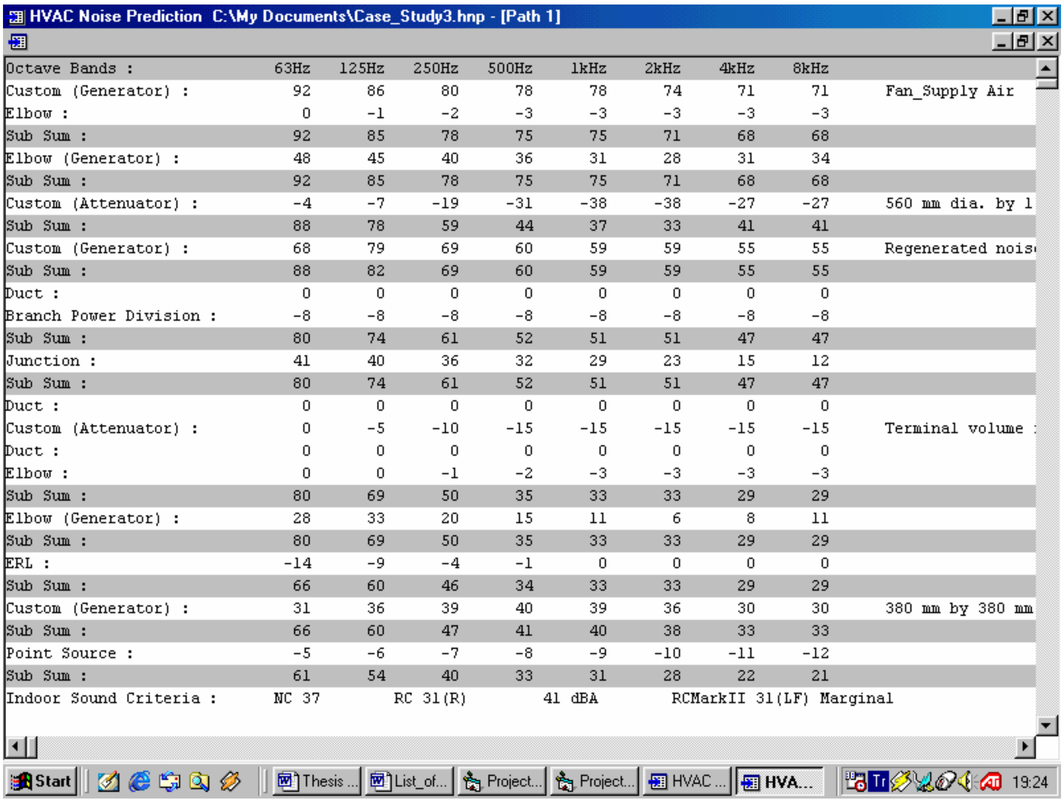
End Reflection Loss (17):
Duct Type: Round
Duct Diameter: 250 mm

Custom - Generator (18) in dB: 380 mm by 380 mm rectangular diffuser

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
31	36	39	40	39	36	30	30

Receiver Correction (19): Point Sound Source
Room Volume: 6.1*6.1*2.75 = 102.3275 m^3
Shortest Distance from Noise Source to Receiver: 1.5 m

Path 1 generated sound pressure level at receiver location and the rating of this path is given in Figure 6.16.



Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	92	86	80	78	78	74	71	71	Fan_Supply Air
Elbow :	0	-1	-2	-3	-3	-3	-3	-3	
Sub Sum :	92	85	78	75	75	71	68	68	
Elbow (Generator) :	48	45	40	36	31	28	31	34	
Sub Sum :	92	85	78	75	75	71	68	68	
Custom (Attenuator) :	-4	-7	-19	-31	-38	-38	-27	-27	560 mm dia. by 1
Sub Sum :	88	78	59	44	37	33	41	41	
Custom (Generator) :	68	79	69	60	59	59	55	55	Regenerated nois
Sub Sum :	88	82	69	60	59	59	55	55	
Duct :	0	0	0	0	0	0	0	0	
Branch Power Division :	-8	-8	-8	-8	-8	-8	-8	-8	
Sub Sum :	80	74	61	52	51	51	47	47	
Junction :	41	40	36	32	29	23	15	12	
Sub Sum :	80	74	61	52	51	51	47	47	
Duct :	0	0	0	0	0	0	0	0	
Custom (Attenuator) :	0	-5	-10	-15	-15	-15	-15	-15	Terminal volume :
Duct :	0	0	0	0	0	0	0	0	
Elbow :	0	0	-1	-2	-3	-3	-3	-3	
Sub Sum :	80	69	50	35	33	33	29	29	
Elbow (Generator) :	28	33	20	15	11	6	8	11	
Sub Sum :	80	69	50	35	33	33	29	29	
ERL :	-14	-9	-4	-1	0	0	0	0	
Sub Sum :	66	60	46	34	33	33	29	29	
Custom (Generator) :	31	36	39	40	39	36	30	30	380 mm by 380 mm
Sub Sum :	66	60	47	41	40	38	33	33	
Point Source :	-5	-6	-7	-8	-9	-10	-11	-12	
Sub Sum :	61	54	40	33	31	28	22	21	

Indoor Sound Criteria : NC 37 RC 31(R) 41 dBA RCMarkII 31(LF) Marginal

Figure 6.16 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study3, Path 1 Rating Results

Path 2:

Custom - Generator (1) in dB: Fan-Supply Air, 3.3 m^3/s, 620 Pa

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
92	86	80	78	78	74	71	71

Elbow (3):

Elbow Type: Round

Elbow Width: 560 mm

Elbow-Generator (4): 90^0 bend w/o turning vanes, 320 mm radius

Turning Vane: without

Pressure Drop Across Elbow: 620 Pa

Volume Flow Rate: 3.3 m^3/s

Cross Sectional Area of the Elbow: $pi*(0.56)^2/4 = 0.2463\ m^2$

Height of the Elbow: 0.56 m

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	3	5	10	21	42	84	167	334
K (Char. Spectrum) in dB	0	-6	-14	-21	-29	-35	-35	-35
Rounding Correction in dB	0	0	0	0	0	0	0	0

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 320/560 = 0.57

Custom - Attenuator (5) in dB: 560 mm dia. by 1.12 m high pressure silencer

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
4	7	19	31	38	38	27	27

Custom - Generator (6) in dB: Regenerated noise from above silencer

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
68	79	69	60	59	59	55	55

Duct (7):

Duct Type: Circular/Unlined

Inside Diameter: 560 mm

Duct Length: 2.44 m

External Insulation: No

Branch Power Division (8):

Branch Cross Sectional Area: $pi*(560)^2/4 = 246300.864\ mm^2$

Total Cross-Sectional Area Leaving The Take-off:

$pi*(250)^2/4 + pi*(560)^2/4 = 295388.2493$

Junction (9): Duct 90^0 branch take-off, 50 mm radius

Volume Flow Rate in Main Duct: 3.3 m^3/s

Volume Flow Rate in Branch Duct: 3.3 m^3/s

Cross-sectional Area of the Main Duct: $pi*(0.56)^2/4 = 0.2463$

Cross-sectional Area of the Branch Duct: $pi*(0.56)^2/4 = 0.2463$

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	3	5	10	21	42	84	167	334
K (Char. Spectrum) in dB	1	-4	-13	-21	-29	-35	-35	-35
Rounding Correction in dB	2.8	2.5	2	1.8	1.5	0	0	0
Turbulence Correction in dB	0	0	0	0	0	0	0	0

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 50/560 = 0.089

Note 3: HNP automatically calculates Velocity Factor value, and the user enters the Upstream Turbulence Correction iaw the Velocity Factor value. In this case, Velocity Factor value is $M=U_M/U_B = 1$.

Transmission Through Duct Wall (20):

Sound Attenuator: Chosen

Transmission Type: Breakout

Duct Type: Circular

Diameter: 560 mm

Circular Duct Type = Long Seam

Transmission Through Ceiling (21):

Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)

Surface area of the bottom panel of the source closest to the ceiling tiles:

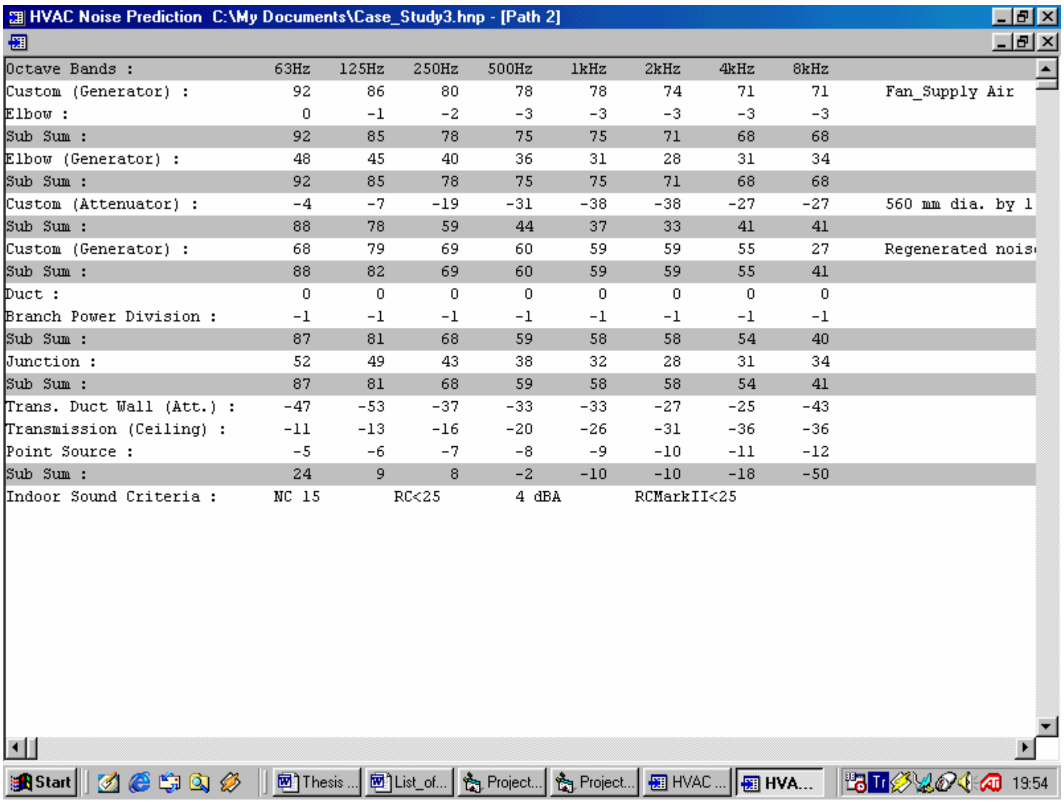
1.22*2*pi*(0.56/2) = 2.1463

Receiver Correction (22): Point Sound Source

Room Volume: 6.1*6.1*2.75 = 102.3275 m^3

Shortest Distance from Noise Source to Receiver: 1.5 m

Path 2 generated sound pressure level at receiver location and the rating of this path is given in Figure 6.17.



Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	92	86	80	78	78	74	71	71	Fan_Supply Air
Elbow :	0	-1	-2	-3	-3	-3	-3	-3	
Sub Sum :	92	85	78	75	75	71	68	68	
Elbow (Generator) :	48	45	40	36	31	28	31	34	
Sub Sum :	92	85	78	75	75	71	68	68	
Custom (Attenuator) :	-4	-7	-19	-31	-38	-38	-27	-27	560 mm dia. by 1
Sub Sum :	88	78	59	44	37	33	41	41	
Custom (Generator) :	68	79	69	60	59	59	55	27	Regenerated nois
Sub Sum :	88	82	69	60	59	59	55	41	
Duct :	0	0	0	0	0	0	0	0	
Branch Power Division :	-1	-1	-1	-1	-1	-1	-1	-1	
Sub Sum :	87	81	68	59	58	58	54	40	
Junction :	52	49	43	38	32	28	31	34	
Sub Sum :	87	81	68	59	58	58	54	41	
Trans. Duct Wall (Att.) :	-47	-53	-37	-33	-33	-27	-25	-43	
Transmission (Ceiling) :	-11	-13	-16	-20	-26	-31	-36	-36	
Point Source :	-5	-6	-7	-8	-9	-10	-11	-12	
Sub Sum :	24	9	8	-2	-10	-10	-18	-50	
Indoor Sound Criteria :	NC 15		RC<25	4 dBA		RCMarkII<25			

Figure 6.17 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study3, Path 2 Rating Results

Path 3:

Custom - Generator (2) in dB: Fan-Return Air, 3.3 m^3/s, 620 Pa

Octave band center frequency, Hz							
63	125	250	500	1000	2000	4000	8000
82	79	80	78	78	74	71	71

Elbow (23):

Elbow Type: Square/Lined

Turning Vane: without

Elbow Width: 810 mm

Elbow-Generator (24): 90^0 bend w/o turning vanes, 12.7 mm radius

Turning Vane: without

Pressure Drop Across Elbow: 620 Pa

Volume Flow Rate: 3.3 m^3/s

Cross Sectional Area of the Elbow: (0.81)^2 = 0.6561 m^2

Height of the Elbow: 0.81 m

	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
St (Strouhal Number)	10	20	40	81	161	322	644	1288
K (Char. Spectrum) in dB	-13	-20	-28	-35	-35	-35	-35	-35
Rounding Correction in dB	4.5	4	3.6	0	0	0	0	0

Note 1: HNP automatically calculates Strouhal Number values, and the user enters Characteristic Spectrum and Rounding Correction values according to these Strouhal Numbers.

Note 2: Rounding correction values are determined iaw the r/D_{Br} ratio. In this case, this ratio is 12.7/810 = 0.015

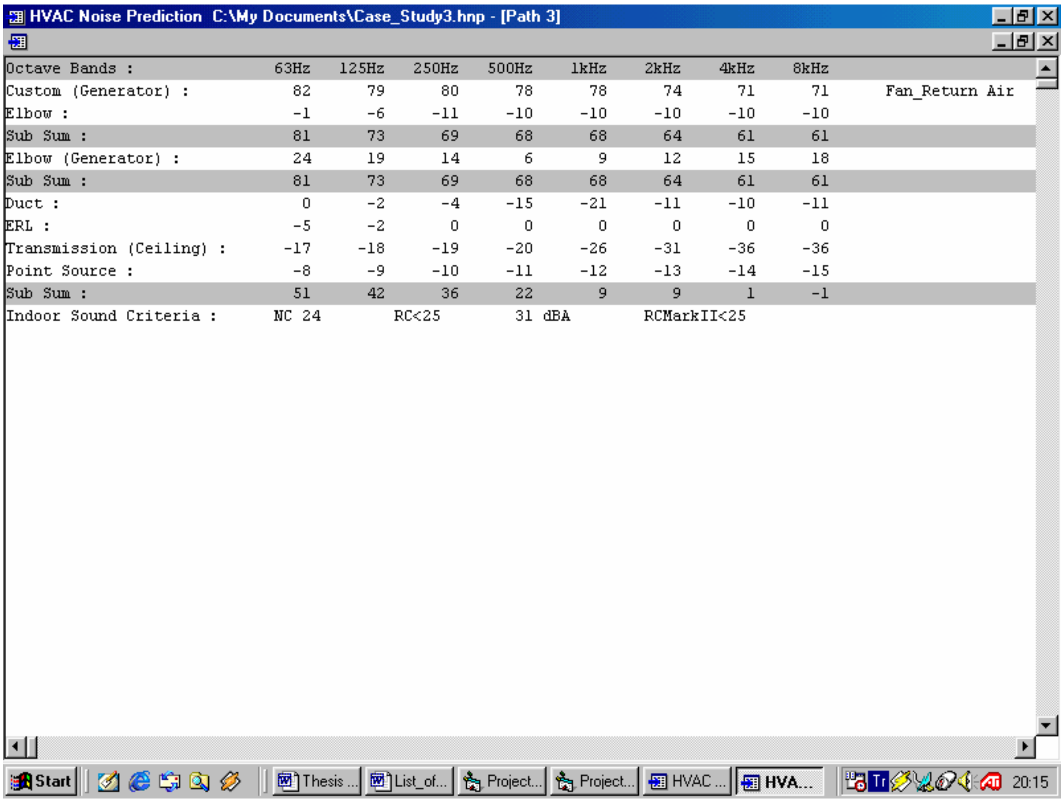
Duct (25):
Duct Type: Rectangular/Lined
Inside Height: 810 mm
Inside Width: 1730 mm
Duct Length: 2.44 m
Lining Thickness: 50 mm
External Insulation: No

End Reflection Loss (26):
Duct Type: Square/Rectangular
Mean Duct Width: 810 mm

Transmission Through Ceiling (21):
Ceiling Tile Type: Mineral fiber (16 mm, 4.9 kg/m^2)
Surface area of the bottom panel of the source closest to the ceiling tiles:
0.61*1.22 = 0.7442

Receiver Correction (27): Point Sound Source
Room Volume: 6.1*6.1*2.75 = 102.3275 m^3
Shortest Distance from Noise Source to Receiver: 3 m

Path 3 generated sound pressure level at receiver location and the rating of this path is given in Figure 6.18.



Octave Bands :	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz	
Custom (Generator) :	82	79	80	78	78	74	71	71	Fan_Return Air
Elbow :	-1	-6	-11	-10	-10	-10	-10	-10	
Sub Sum :	81	73	69	68	68	64	61	61	
Elbow (Generator) :	24	19	14	6	9	12	15	18	
Sub Sum :	81	73	69	68	68	64	61	61	
Duct :	0	-2	-4	-15	-21	-11	-10	-11	
ERL :	-5	-2	0	0	0	0	0	0	
Transmission (Ceiling) :	-17	-18	-19	-20	-26	-31	-36	-36	
Point Source :	-8	-9	-10	-11	-12	-13	-14	-15	
Sub Sum :	51	42	36	22	9	9	1	-1	
Indoor Sound Criteria :	NC 24		RC<25		31 dBA		RCMarkII<25		

Figure 6.18 A Snap Shot from the HNP Program with a Completed Path Showing the Case_Study3, Path 3 Rating Results

At the end, the total sound pressure levels in the receiver room from the three paths are obtained by logarithmically adding the individual sound pressure levels associated with each path. This operation completes the HNP analysis for this scenario. The sum of Paths 1, 2 and 3 is seen in the Summary Table View window activating the Sum View window by double-clicking on the Sum 1 icon, as seen in Figure 6.19. On the other hand, the full view of the Path View of this scenario is given in Figure 6.20 to provide the whole path structure.

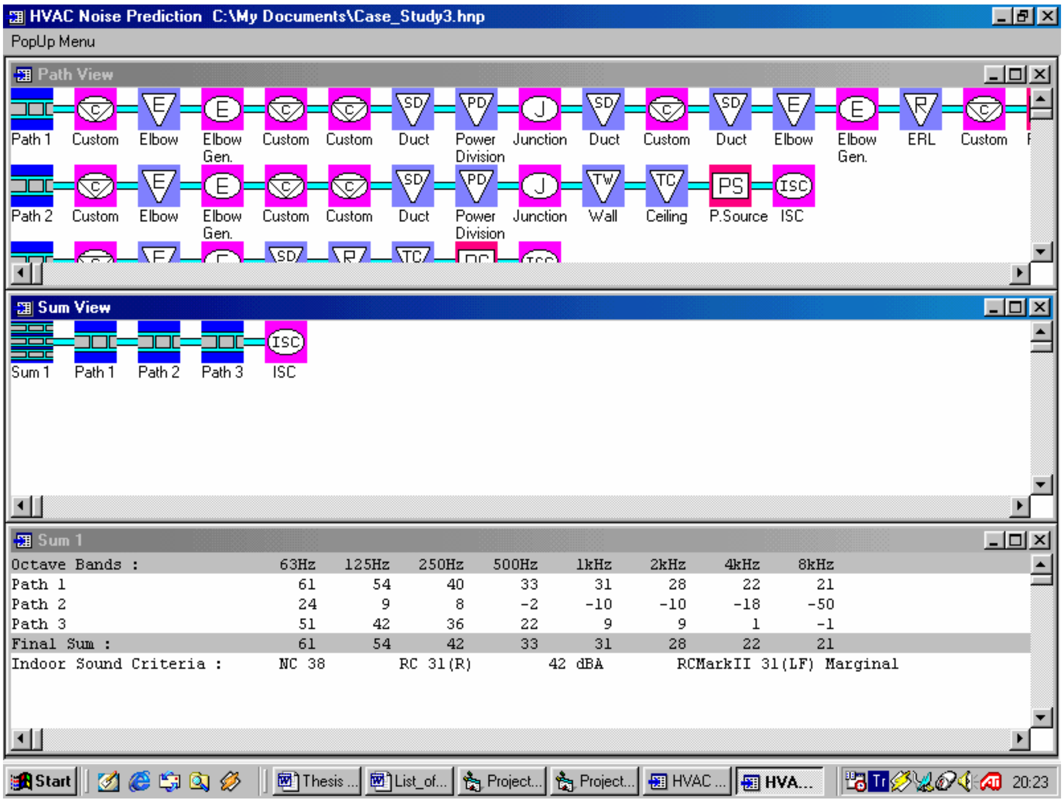


Figure 6.19 A Snap Shot from the HNP Program with a Completed Sum Showing the Case_Study3 Rating Results

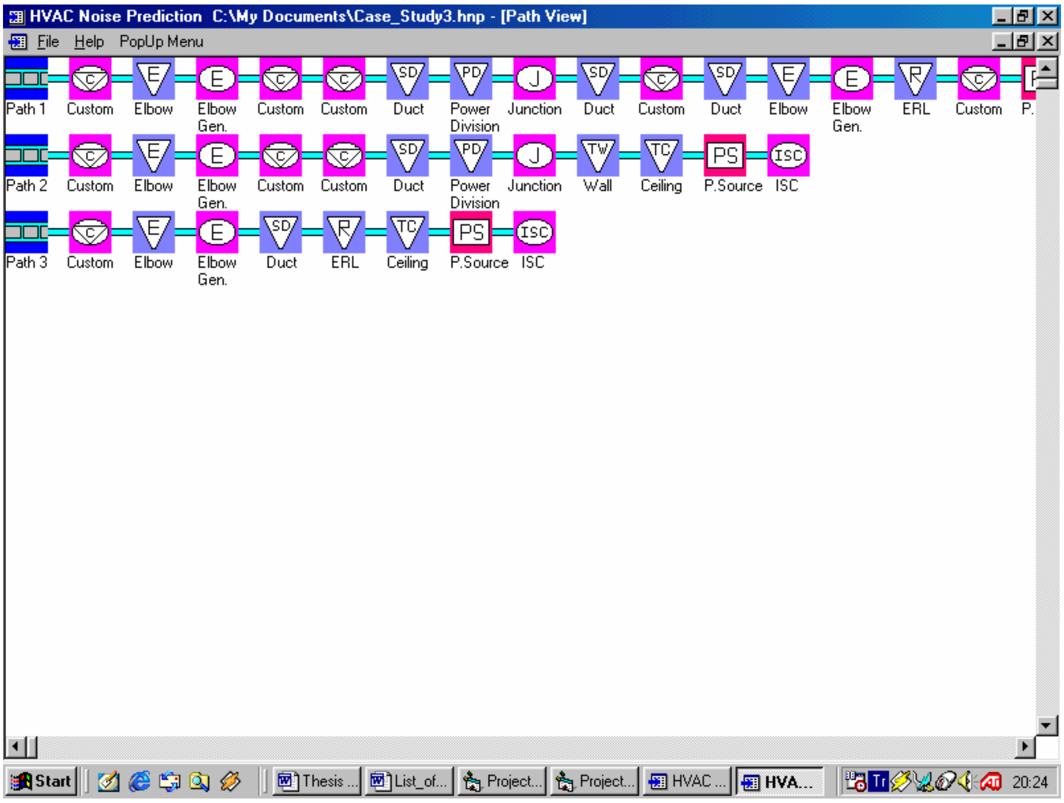


Figure 6.20 A Snap Shot from the HNP Program with a Full View of the Path View of Case_Study3

It can be seen from the results that the critical paths are Path 1 which comprised from fan airborne supply air sound that enters the room from the supply air system through the ceiling diffuser, and Path 3 comprised from fan airborne return air sound that enters the room from the inlet of the return air duct. The ratings of the overall sound pressure level of the three paths are NC 38, RC 31 (R), 42 dBA and RC MarkII 31 (LF) according to the four different rating procedures. RC procedure points out that the tonal spectrum of the resultant sound behaves rumble characteristic and RC MarkII method supports this conclusion showing that there is a marjinal rumble spectrum at low frequency region, i.e. (16-63 Hz). When the values stated in Table 4.4 are compared to the ratings of Case_Study3, it is seen that the obtained overall sound level is permitted for almost every kind of environment ranging from office buildings to libraries except art performing spaces.

CHAPTER 7

RESULTS AND DISCUSSIONS

A HVAC system is designed to provide the maximum comfort standards to the occupants whom it serves. This means providing required thermal comfort conditions to the environment where the occupants stay or work, as well as satisfactory acoustical conditions. Examination of the noise generated due to HVAC components necessitates elaborating lots of prediction formulae and handling several data either obtained from manufacturer or from the previous research studies. For this reason, it is intended to prepare a prediction program in a user-friendly manner to help tackle the acoustical calculations during the design phase of HVAC system. Since operating performance and the resultant acoustical level depending on operating conditions of each of the HVAC elements require to be elaborated together, making such a tool will ease the work of HVAC system designer.

For this purpose, several sources in technical literature have been investigated to compile necessary data in order to prepare such a prediction tool. ASHRAE and ARI standards were the primary sources for the calculation procedures needed to use in estimation algorithms and for the experimental data needed to embed in the program. The 1998, 1999 standards were heavily used in the algorithm of the program as far as 1991 standard that was accepted as the raw data for prediction algorithms. The acquired data in question obtained from this survey was exhibited in Chapter 3 of the study. In Chapter 3, a wide variety of sound generators and attenuators were introduced within the scope of the study and most of this data was used in prediction software. A “custom equipment” entry dialog box was included in the program to enable the user to define custom sound generators or attenuators for the elements where the user obtained its relevant data from its manufacturer.

Another primary concern in the process of HVAC system design is “to what extent the permissible sound levels are reached or whether those levels exceeded the reference levels for the functional space in question”. That is why; it was necessary to handle several indoor rating procedures to understand the system design in terms of acoustical concerns. The rating procedures used for this same purpose show some differences while evaluating the resultant sound level. These procedures were expressed in details in Chapter 4. However, it shall be stated that one of those rating procedures named as RC Mark II is newly introduced and it has the advantage of making sound quality assessment different from the other ones. And this rating procedure was added to the program as an important feature in addition to the conventional rating procedures.

It is important for such a program to provide flexibility to the user to quickly introduce a path or combination of paths to the system when the user needs to change one or more parameters and to try another possible solutions for system design purposes. HNP provides this feature to the user.

The features, structure and the capabilities of the program were demonstrated via three different existing scenarios in Chapter 6. These scenarios were chosen to validate the software HNP. At the end of the execution of the software three times, three different tonal spectra were obtained as neutral, rumble and hissy. RC and RC MarkII methods were consistent with each other looking at the rating values provided by such methods.

If the simulation work and the outcomes of the prediction software are examined, it can be said that the program HNP works properly and provides fruitful features for HVAC designers who wants to gain precious design time by bypassing lots of calculations. This also demonstrates the prepared software HNP accurately predicts resulting noise levels in rooms due to HVAC systems and it can be used to rate the quality aspect of the resulting noise.

As a future enhancement of the software HNP, optimization feature can be added to enable the program for making suggestions during the execution of the scenarios. Such suggestions may help the designer to select too alternatives, such as, adaptation of a silencer or lining the duct work.

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