SOUND POWER LEVEL PREDICTION IN DUCTED HEATING, VENTILATING AND AIR CONDITIONING SYSTEM

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ABSTRACT

SOUND POWER LEVEL PREDICTION IN DUCTED HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS

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The aim of this study is to develop user friendly software to predict and analyze noise levels in enclosed spaces due to Heating, Ventilating and Air Conditioning (HVAC) System. Sound sources and transmission mechanisms are investigated by the prediction formulae and data originated from research studies and standards. For the analysis portion of the software, through the implementation of HVAC elements, sound power levels at the ventilation system outlet are obtained. General software structure is explained by different user-interface samples in the thesis. A case study is presented to display the capability of the software prepared in C# programming language within the scope of the study.

Keywords: Heating, Ventilating and Air Conditioning (HVAC) System, HVAC noise, ventilation noise, noise prediction software.

ÖZ

KANALLI ISITMA, HAVALANDIRMA VE İKLİMLENDİRME SİSTEMLERİNDE SES GÜCÜ SEVİYESİ TAHMİNİ

Yayladere, Cemil

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Bu çalışmanın amacı; Isıtma, Havalandırma ve İklimlendirme Sistemlerinde ortaya çıkan gürültünün tahminine ve analizine yönelik, kullanıcı arayüzü kolay bir yazılım programı hazırlamaktır. Çeşitli araştırmalar sonucu elde edilmiş olan standart verileri ve formülleri kullanarak havalandırma kanallarında gürültü kaynakları ve ses yayılımları incelenmiş ve geliştirilen yazılıma entegre edilmiştir. Sözkonusu yazılımın genel yapısı ve yazılıma uygulanan farklı tipteki havalandırma kanal öğelerinin yazılımdaki uygulamaları kullanıcı arayüz örnekleri verilerek çalışmada anlatılmaktadır. C# programalama dili kullanılarak hazırlanan yazılımın yetkinliği örnek vaka çalışması kullanılarak tez kapsamında incelenmiştir.

Anahtar Kelimeler: Isıtma, Havalandırma ve İklimlendirme Sistemleri, gürültü tahmini, gürültü tahmin yazılımı, havalandırma gürültüsü.

To My Daughter

&

To My Love

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NOMENCLATURE

L_w	sound power level
K_F	spectral constant which depends on type of fan
Q_F	volume of air per time passing through the fan
Q_{REF}	reference volume
P_F	static pressure produced by the fan
P_{REF}	reference pressure
C_{EFF}	efficiency correction factor
C_{BFI}	blade frequency increment correction
f_{bp}	blade passage frequency
f_o	center frequency of octave band
K_j	characteristic spectrum of the junction or turn
U_B	velocity of branch duct
S_B	cross section area of the branch
Q_B	flow volume of the branch
D_B	equivalent diameter of the branch duct
C_B	correction term depends on type
Δr	correction term for the roughness of elbow
ΔT	correction term for upstream turbulence
R_D	rounding parameter
R	radius of the inside edge of the bend
S_t	Strouhal number
C	pressure loss coefficient
B	blockage factor
U_M	velocity of main duct
U_C	flow velocity in the constricted part of the duct
K_D	characteristic spectrum of the damper

S	cross section area of the duct
D_H	duct height
ΔP	pressure drop across the fittings
Q	flow volume
K_T	characteristic spectrum of the elbow with turning vanes
D_C	chord length of a typical vane
n	number of turning vane
P	perimeter of the duct
l	length of the duct
t	thickness of the lining
d	interior diameter of the duct
w	width of the duct
S_M	cross section area of the main duct
S_i	cross section area of the i^{th} branch
r	radius of the elbow
Q_i	directivity of the inlet
R	room constant of the plenum
S_i	sound inlet area of the plenum
S_o	sound outlet area of the plenum
S_p	interior surface area of the plenum
θ	angle between the inlet and the outlet
r	distance between the inlet and the outlet of the plenum
c_0	speed of sound in air
f	frequency
f_{co}	cut off frequency
a	larger dimension of the rectangular duct

ABBREVATIONS

HVAC	Heating, Ventilating and Air Conditioning
VAV	Variable-Air-Volume
ASHREA	American Society of Heating and Refrigeration Engineers
AHRI	Air Conditioning, Heating and Refrigeration Institute
FC	Forward Curve
NC	Noise Criterion

CHAPTER 1

INTRODUCTION

1.1 General Overview

In modern buildings, provision of thermal comfort and acceptable indoor air quality with reasonable installation is obtained by implementing Heating, Ventilating and Air Conditioning (HVAC) systems. However, noise problem may originate from the ventilation equipment itself and ductwork installed to circulate air in enclosed spaces. The solution of the overall problem requires systematic application of advanced acoustical engineering principles.

Noise control in HVAC systems should begin, at its best, in early stages of building design. The design according to requirements and modification for satisfactory noise control can be executed in initial construction phase. It is more economical to predict noise levels of building rather than measure, even if the building already exists. As a matter of fact, the more convenient way is to make reliable engineering estimates of the noise levels and applicable noise mitigation techniques in the HVAC system for specific components of the system.

The aim in ventilation quietening is not to eliminate all noise but to produce a balanced noise environment. For environmental considerations in enclosed spaces, the sound generated by mechanical equipment and sound transmission through ductwork has to be taken into consideration of the design of the HVAC system to enable establishment of acceptable indoor noise criteria in the buildings. Thus, the selection of the mechanical equipment and the design of spaces should be undertaken with an emphasis on noise control in occupied spaces of the buildings in which the HVAC equipment is in service.

Noise control in HVAC system design involves examining sound sources, transmission paths, and receivers. Sources are the components that generate noise when airflow passes through them. Paths are the routes by which the sound travels: through the air, over barriers, or along a ducted system. Receivers are usually the people who occupy a building or live in the nearby community. There are many possible paths for airborne and structure borne sound transmission between a sound source and receiver as shown in Figure 1-1.

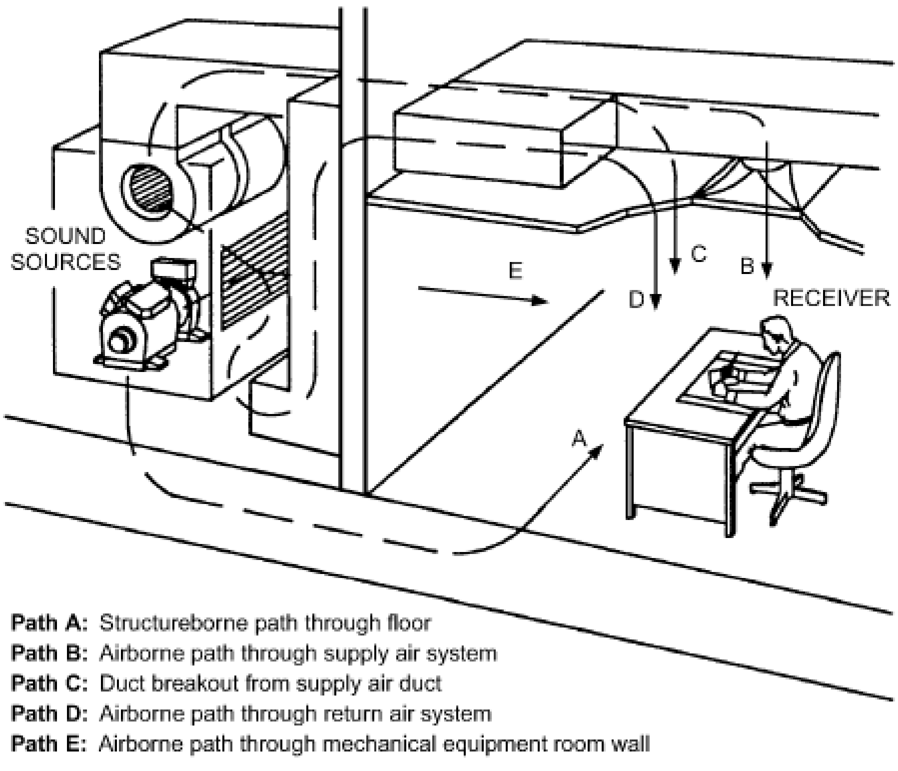


Figure 1-1 Typical Paths in HVAC Systems [1]

1.2 Scope and Objectives

Due to the necessity of a thorough noise analysis in the HVAC system design, it is intended to develop a user-friendly noise prediction program with simple interfaces 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 ventilation system noise levels.

While preparing the algorithm of the software, the source-path-receiver chain associates the components of the HVAC system. Sound generated from the source travels through path and it reaches to the receiver. The aim of the study is to investigate the airborne noise travelled from the source to the receiver.

As a consequence, the primary concern of the thesis study is to prepare prediction software to estimate the sound power levels at the outlet of the ventilation system in the occupied spaces. In this study, methods are given for predicting airborne noise path through ventilation system. An investigation of the sound propagation in HVAC elements is carried during the preparation of the software.

1.3 Outline of the Dissertation

The outline of the dissertation presented is given below:

In Chapter 2, information about publications, studies of researchers about HVAC system noise; several standards issued relating to the noise of HVAC components are given. Available similar HVAC noise prediction software programs are presented.

In Chapter 3, a theoretical background of the subject is explained. The HVAC elements noise generation and attenuation prediction equations and parameters from standards and research studies are given. The formulations and data demonstrated hereby form also the backbone of the algorithm of the prepared prediction software.

In Chapter 4, introduction of the software written within the scope of the study, its general structure, and implementation of HVAC elements employed are explained through presentation of different user-interface samples.

In Chapter 5, capability and structure of the software are demonstrated by a case study. The analysis is described including explanations and result screen is represented in this section.

In the final chapter, the predicted response values are discussed and the conclusion of the study is presented. Moreover, suggestions for possible future development, applicable to the program, can be found in the last chapter.

CHAPTER 2

LITERATURE SURVEY

2.1 Review of Literature

The control of background noise from HVAC systems in buildings has been a serious concern for many years. With the rise in the construction of air-conditioned buildings, the realization of excessive noise and vibration stemming from HVAC systems was occurred. However, in those times due to lack of information about the noise source elements and the path of the noise, making noise predictions for a design engineer was difficult. Thus, the design engineers tried to approach such noise problems by trial and error method, but they could not determine the acceptable values for the sound originated from HVAC systems.

In the late 1950's, development of HVAC noise control knowledge and technology took place and the first predictions for the people's subjective response to noise were made. While sound rating methods were studied within the years, researches and publications were issued on noise characteristics of HVAC system's components. Test data on HVAC system components were published by researchers or manufacturers and some of them are standardized.

From the aspect of research and publications on the topic of noise control design on HVAC systems, ASHRAE Handbooks presented remarkable studies with the basic sound and vibration principles on acoustic design guidelines and system design requirements from 1991 and 1999. Among those, ANSI/AMCA Standard 300 [4] provided the approved test conditions for sound emission by any type of fan performing at a given duty and presented sound attenuation characteristics associated with duct end reflection losses for ducts terminated into free space. On the other study, ASHRAE Standard 68/AMCA Standard 330 [5] determines the sound power radiated into duct on the supply and/or return side of the air handling equipment. Air Conditioning, Heating and Refrigeration Institute (AHRI) Standard 880 [6] specified the procedures to define, classify and test the rating requirement of Variable Air Volume (VAV) systems. ASHRAE Standard 70 [7] or AHRI Standard 890 [8] provided testing methods for air outlets and air inlets for distribution of air. AHRI Standard 575 [9] and AHRI Standard 370 [10] made possible to obtain factory sound data for indoor and outdoor air conditioning equipment, respectively. ASHRAE 1987 [11] handbook listed sound attenuation values for unlined round circular ducts. AHRI Standard 885 [12] provided a method for estimating sound pressure level in occupied spaces for the application of air terminals, air outlets and the low pressure ductwork. ASTM Standard E477 [13] outlined the test procedures to measure the acoustical insertion loss, airflow generated noise, and pressure drop as a function of airflow for dissipative and reactive silencers.

Studies on noise control of HVAC systems were conducted not only by ASHRAE, but also researchers at scientific and private institutions. Early work on HVAC systems was published by Reynolds and Bledsoe which covered HVAC system design and its algorithms in 1991 [14]. Schaffer [15] developed specific guidelines for acoustic design of HVAC systems. The study also pointed the construction phases related with HVAC systems and the identification of sound and vibration problems of HVAC systems was achieved (1991). The acoustical data of HVAC systems provided by the manufacturers and the implementation of the acoustical data were presented in a study carried by Ebbing and Blazier [16] in 1998. Another study conducted by Reynolds and Bevirt [20] emphasized on the techniques and requirements of the HVAC sound and vibration measurements. The measurement procedures and specifications taking into account of the construction installation were dealt in this study

.

From the aspect of transmission loss on HVAC systems, Beranek [18], Reynolds and Bledsoe [14], Reynolds and Bevirt [17], and Wells [19] conducted studies separately.

Several authors [20], [21], [22] examined the attenuation values of straight unlined rectangular sheet metal ducts. Moreover, tabulated attenuation values of rectangular sheet metal ducts with 25 mm and 50 mm acoustical linings and tabulated insertion loss values for dual-wall round sheet metal ducts for 25 mm and 50 mm can be found in the literature [21], [23], [24]. The insertion loss values for unlined and lined square elbows were suggested by Beranek [25]. In his study, La Ver [26] provided the principle of distribution of the sound power for the duct related with main feeder part of the duct and the branches containing the junctions of the element.

2.2 Review of Existing Software

Some noise prediction software were developed to reduce time consumption and design effort encountered in making a tremendous amount of noise prediction calculations in the HVAC system design. In the development of the noise prediction software the standardized studies and results of the previous research studies outlined above are used in a large scale in the preparation of such software.

Some of the major well known noise prediction programs, extensively used for the noise prediction studies of HVAC systems, are explained here briefly. Quick reviews are given for Trane Acoustics Program, Acoustic Analyzer, V-A Select Release 5.1, Applied Acoustics Program, HNP and Dynasonics AIM. Most of the programs described are commercial software from HVAC equipment's and components manufacturer or HVAC design consultant.

Trane Acoustics Program, which is Microsoft® Windows based tool, was developed with a claim to accurately model the sound level in occupied spaces [27]. ASHRAE's 1991 Algorithms for HVAC Acoustics handbook are the basis for estimating sound level and program includes Trane products sound data base. Program also analyses multiple paths and calculates the total effect for the enclosed space and contribution of each path (Figure 2-1). Analyzing sound paths can be performed by choosing specific equipment and building components that generate, attenuate, reduce or regenerate sound separately. Moreover, it enables a comparison of calculated sound levels with the desired indoor noise criteria.

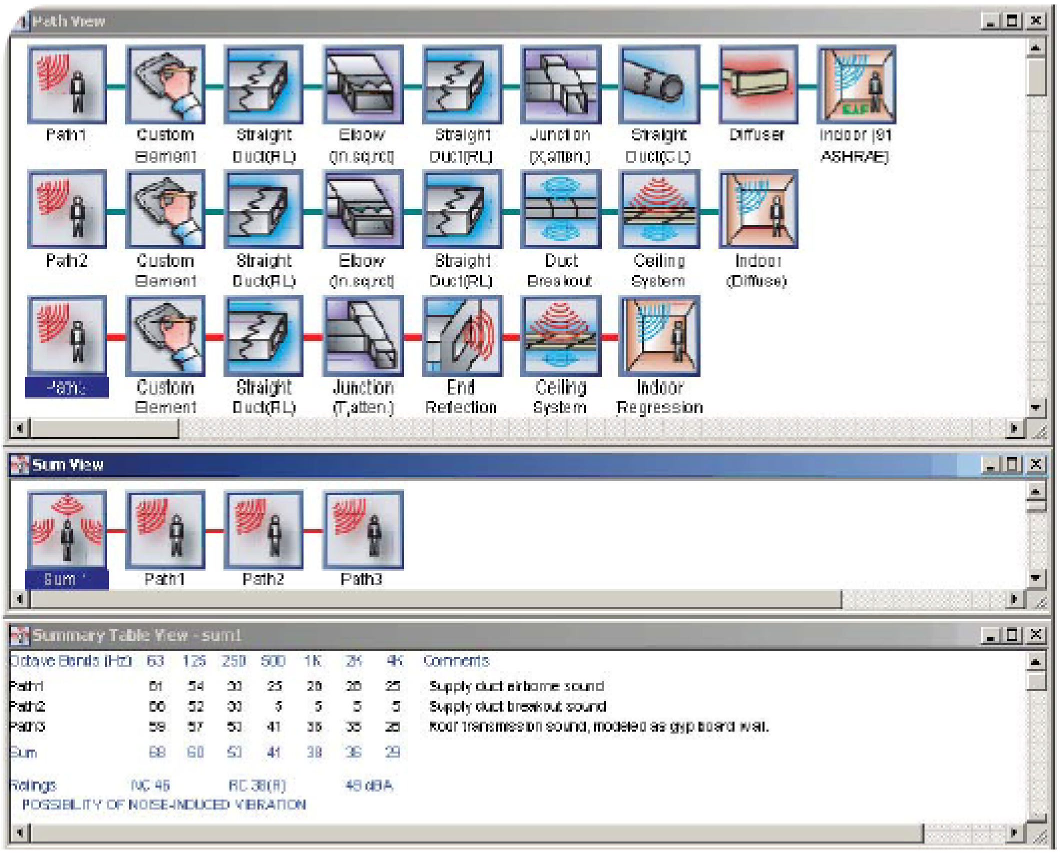


Figure 2-1 Trane Acoustic Program General Interface View

Acoustic Analyzer was developed to estimate the sound of HVAC in indoor and outdoor spaces. The software creates virtual scenario with the HVAC design using embedded design templates and offers 3 system analysis options, namely, outdoor, zoned comfort system and central system [28]. With the help of the program, barrier requirement for outdoor equipment, silencer location in an air handling system and HVAC equipment specification can be evaluated from the acoustical point of view.

The program allows the user to define one or more sound sources; the path sound energy travels to a receiver, and one or more sound attenuation or regeneration elements in the path of the sound.

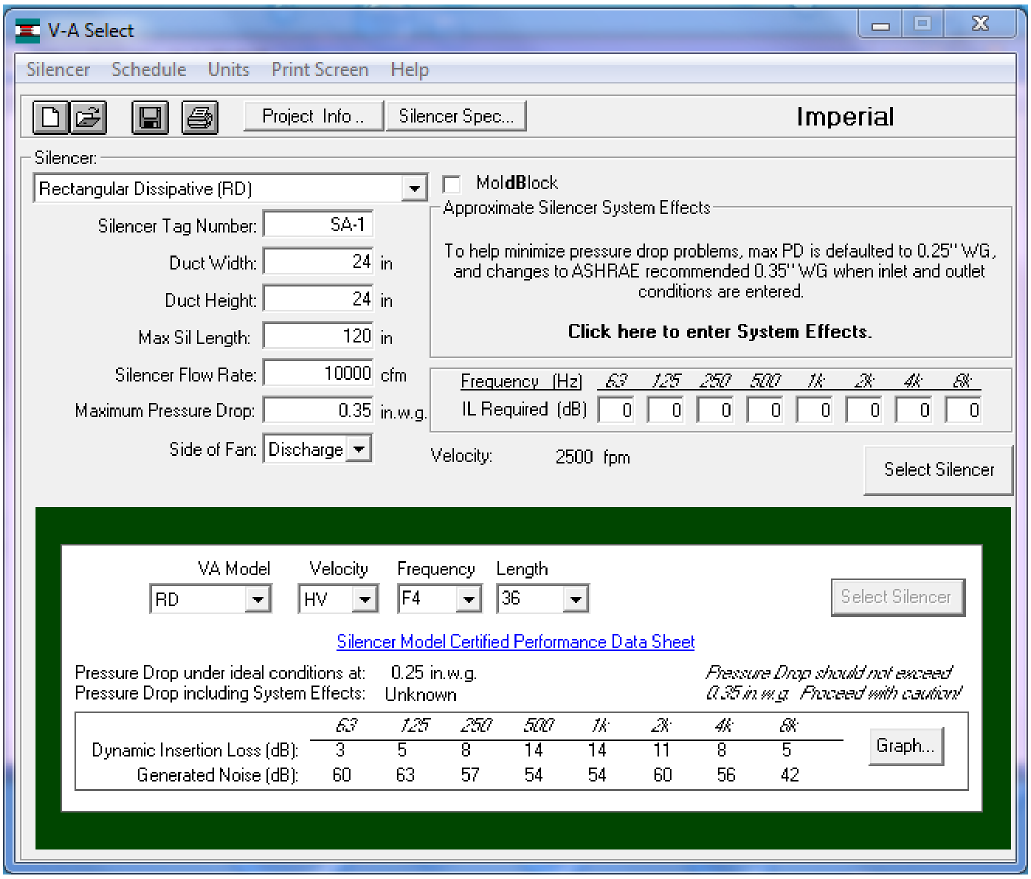


Figure 2-2 V-A Select Release 5.1 Sample Silencer Selection Interface View

V-A Select Release 5.1, developed as air-handling noise control software enables user-friendly acoustical analysis of HVAC duct systems [29]. It is mainly a silencer selection program, with additional module that calculates duct noise. Also noise level in an occupied space can be estimated through acoustic analysis of fan/duct system. The resultant noise level can then be compared to a desired or targeted sound criterion. Vibro-Acoustics, that is the developer of the software, has been working on noise control, vibration insulation and restraint systems. Moreover, with the extensive database of product testing, it is claimed that V-A Select Release 5.1 software provides cost-effective silencer selections to meet project requirements in its software (Figure 2-2). Selecting a silencer, to achieve specified insertion loss and pressure drop, is the distinctive part of the software.

In HNP [30], user can model the HVAC system by choosing equipment, ductwork 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. Program 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). The algorithms, in the establishment of the software, are based on ASHRAE and ARI standards.

Noise prediction software of Dynasonics the Acoustic Information Model (AIM) is designed to model noise influence in individual spaces in building through mechanical air conditioning systems [31]. The software utilizes equipment sound data of the HVAC system components, noise control accessories and architectural elements to predict background noise in the premises. The Dynasonics AIM software consists of multiple tools to perform specific functions. With the experience in acoustic products such as silencer, acoustic panels and louvers, AIM software has a good silencer selection module with integrated company’s silencer database.

Various software products were developed by different companies to serve for specific purposes in noise prediction applications. Thus every program gives different flexibilities to the users. For example; Vibro-Acoustics and Dynasonics Companies, which are experienced companies in HVAC acoustics, offer to their users effective silencer selection for their projects. Every company inserted its experience of specific HVAC product in to their respective software. In most noise prediction programs results are given for the overall path and contribution of HVAC elements cannot be distinguished. In some prediction programs each HVAC elements has to be modeled as a sound attenuation and/or generation element separately by the users.

Similar to this concept, the program developed in this study intends to enable the user to perform an acoustical analysis to estimate the resultant sound power levels in ventilation systems with providing a user-friendly interface for easy use of features.

All of the existing software prepared for the prediction of the noise levels in HVAC systems show both some similarities and differences with each other and with the program written within the scope of this study. It shall be noted that new programs for the estimation noise in HVAC system will definitely be developed in accordance with the developing technological innovation in HVAC systems by closely following new studies and updated standards.

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 generation mechanisms 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, misfittings of parts and components, bearing noise, brush noise, magnetic noise, and belt noise.

All buildings have fans of one sort or another for air circulation in HVAC systems. Fans are typed according to the mechanism used to propel the air, and further subdivided according to the type of blade. The basic types are axial and centrifugal.

Axial fans are the simplest to understand; they have a fixed-pitch multiple-bladed rotor. Propeller fans are unhoused, whereas vane axial and tube axial fans include a shroud or housing around the impeller.

Vane axial fans have fixed stator blades to straighten the flow after it passes through the rotor blades; tube axial fans do not.

Centrifugal fans consist of a series of blades, arranged at even intervals around a circle like a water wheel, that throw the air from the inside to the outside of the circle as they rotate. Forward curved (FC) fans are commonly used in many air handlers. The blade pass of FC fans is typically less prominent and occurs 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. In backward-curved or backward-inclined blades the air velocity is lower than the tip velocity, so a lower noise level is generated. The forward-curved blades can generate the same air volume at a lower rotational speed, which means that the peak in their spectrum occurs at a lower frequency.

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 exhaust [1].

Fan noise is generated by several mechanisms, including the surge of the air pressure and velocity each time a blade passes, turbulent airflow in the air stream, and physical movement of the fan casing or enclosure. The noise emitted by each fan type follows a series of generalized laws called scaling laws, which have the general form;

$$L_w = K_F + 10\log\frac{Q_F}{Q_{REF}} + 10\log\frac{P_F}{P_{REF}} + C_{EFF} + C_{BFI} \quad (3.1)$$

Table 3-1 Level Correction K_F for Total Sound Power of Fans, dB [11]

Fan Type	**Octave Band Center Frequency (Hz)**							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Centrifugal								
Airfoil, backward curved backward inclined								
< 900 mm	45	45	43	39	34	28	24	19
> 900 mm	40	40	39	34	30	23	19	17
Forward Curved								
All	53	53	43	36	36	31	26	21
Radial Total Pressure (kPa)								
low 1 to 2.5	56	47	43	39	37	32	29	26
meddium 1.5 to 3.7	58	54	45	42	38	33	29	26
high 3.7 to 15	61	58	53	48	46	44	41	38
Vaneaxial Hub Ratio								
0.3 to 0.4	49	43	43	48	47	45	38	34
0.4 to 0.6	49	43	46	43	41	36	30	28
0.6 to 0.8	53	52	51	51	49	47	43	40
Tubeaxial								
Wheel Diameter								
< 1000 mm	48	47	49	53	52	51	43	40
> 1000 mm	51	46	47	49	47	46	39	37
Propeller General Ventilation								
All	48	51	58	56	55	52	46	42

Blade frequency increment correction $C_{BFI,}$ shown in Table 3-2, is added to sound power level in the octave band frequency where blade passing frequency f_{bp} resides in. Blade passage frequency in Hertz is represented by the number of times per second a fan's impeller passes a stationary item and can be calculated as below;

$$f_{bp} = \frac{fan\ rpm\ x\ number\ of\ blades}{60} \tag{3.2}$$

At blade passage frequency and its multiples fans generate a tone. Whether this tone is objectionable or barely noticeable depends on the fan type and point of fan operation.

Table 3-2 Blade Frequency Increment Correction, C_{BFI} [11]

Fan Type	**Blade Passing Octave, f_{bp}**	**C_{BFI}**
Centrifugal Airfoil, backward curved, backward inclined	250 Hz	3
Forward curved	500 Hz	2
Radial blade pressure blower	125 Hz	8
Vaneaxial	125 Hz	6
Tubeaxial	63 Hz	7
Propeller Cooling Tower	63 Hz	5

Point of fan operation has a great influence in fan acoustic performance. Commonly operating point of fan is selected near the maximum efficiency to minimize power consumption and noise generation. As the operating point shifts to right side of maximum efficiency (higher airflow and lower static pressure) noise generated by the fan increases. Low frequency noise can increase substantially at operating points to the left of maximum efficiency (lower airflow and higher static pressure) [1]. Hence, correction factor for operating point C_{EFF} is applied to account this increase.

Table 3-3 Efficiency Correction C_{EFF} [11]

Static Efficiency % 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
below 50	16

The sound is radiated from both intake and discharge of the fan. The formula assumes ideal inlet and outlet flow conditions and operation of the fan at a given efficiency. Fans can also radiate noise through their enclosures and into the surrounding space. This is referred to as casing radiation and may be calculated by subtracting a factor for the insertion loss of the casing. Insertion losses are very dependent on the gauge (thickness) and construction of the fan housing.

The sound power generated by a fan performing at a given duty is best obtained from a manufacturer's test data taken under approved test conditions (AMCA Standard 300 or ASHRAE Standard 68/AMCA Standard 330). Applications of air-handling products range from stand-alone fans to systems with various modules and appurtenances. These appurtenances and modules can significantly affect the air handler's sound power levels. In addition, different types of fans that provide similar aerodynamic performance can have significant acoustical differences. Fan sound levels, even when determined by test, may be quite different once the fan is installed in an air handler, which in effect creates a new acoustic environment. Proper testing to determine the resulting sound power levels once a fan is installed in an air handler is essential. Fan manufacturers are in the best position to supply information on their products, and should be consulted for data when evaluating the acoustic performance of fans for an air handler application. Similarly, air handler manufacturers are in the best position to supply acoustic information on air handlers. For all of these reasons, information obtained directly from manufacturers should be preferred over generic fan-only sound power levels.

3.1.2 Variable-Air-Volume System Noise

In recent years due to emphasis on energy conservation, variable air volume systems have become commonly used HVAC system. VAV 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 take care of ductwork design 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 compared to normal operating point. 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 [1].

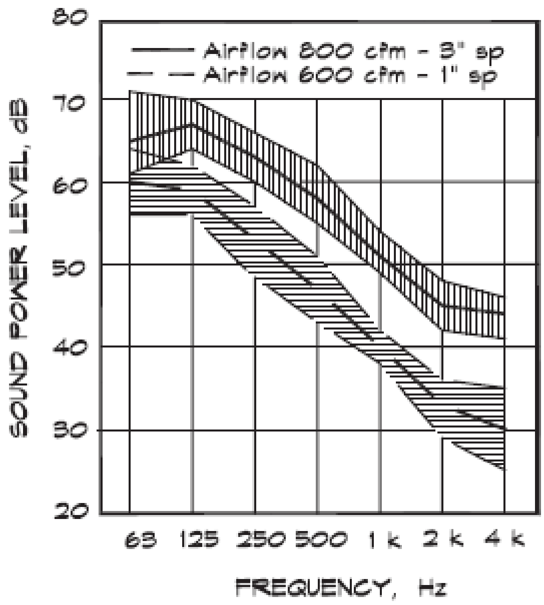


Figure 3-1 Range of discharge VAV Noise Levels at Two operating Points [32]

Blazier [32] has published discharge sound power levels generated by VAV units for two rates of flow (Figure 3-1). In VAV the noise is generated by disturbed flow around the dampers. Many manufacturers publish data on both discharge and casing radiated sound from VAV units. The most useful data are given in terms of sound power levels; however, some manufacturers list data in terms of NC levels, which are obtained by assuming a certain power-to-pressure conversion in the receiving room (usually -10 dB) and sometimes an additional noise reduction (also -10 dB) due to the ceiling tile.

3.1.3 Aerodynamically Generated Sound in Duct

Aerodynamic sound is generated when airflow turbulence occurs at duct elements such as duct fittings (junctions, elbows), dampers, air modulation units, sound attenuators, and room air devices.

Although fans are a major source of sound in HVAC systems, aerodynamically generated sound can often exceed fan sound because of close proximity to the outlet and receiver. When making octave band HVAC ductwork sound calculations using a source-path-receiver analysis, aerodynamically generated sound must be added in the path sound calculations at the location of the element.

3.1.3.1 Straight Ducts

Air flow in ducts can generate noise by creating pressure fluctuations through turbulence, vortex shedding, mixing, and other mechanisms. Steady flow in a straight duct does not generate considerable noise, when compared with other sources in ductwork such as takeoffs, dampers and elbows. In Figure 3-2 shows sound power level data on noise generated in straight duct for various air velocities. Levels generally follow an 18 dB per doubling of velocity scaling law.

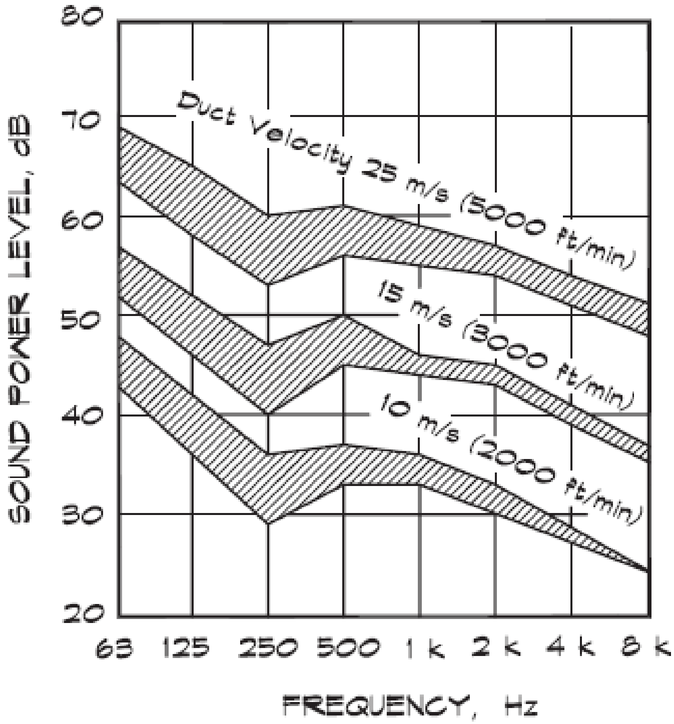


Figure 3-2 Sound Power Spectra of 600x600 mm Straight Steel Duct for Various Air Velocity [33]

3.1.3.2 Air Generated Noise in Junction and Turns

Noise generated in transition elements such as turns, elbows, junctions, and takeoffs can run 10 to 20 dB higher than the sound power levels generated in straight duct runs. Ducts having radiused bends, with an aspect ratio of 1:3 generated no more noise than a straight duct. Elbows having a 90^0 bend are about 10 dB noisier than straight duct. One or more turning vanes can reduce the noise 8 to 10 dB at high and low frequencies while increasing it 3 to 4 dB in the mid frequencies.

Sound power levels generated at elbows without turning vanes, and at junctions can be predicted by following generalized equation [11];

$$L_w\,(f_0) = K_j + 10\log\frac{f_0}{63} + 50\log U_B + 10\log S_B + 10\,log\,D_B + C_B + \Delta r + \Delta T \tag{3.3}$$

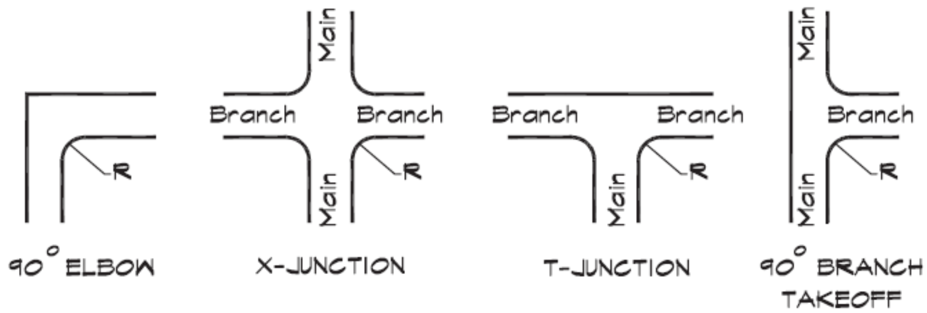


Figure 3-3 Elbows, Junction, and Branch Takeoffs

The flow velocity in the branch line can be calculated as;

$$U_B = \frac{Q_B}{(60\,S_B)} \tag{3.4}$$

The term Δr is a correction for the roundness of bend or elbow associated with the turn and junction [34].

$$\Delta r = \left(1 - \frac{R_D}{0.15}\right)(6.793 - 1.86\,logS_t) \tag{3.5}$$

Where R_D is the rounding parameter for the radius of the inside edge of the bend;

$$R_D = \frac{R}{12\,D_B} \tag{3.6}$$

$$S_t = f_0 \frac{D_B}{U_B} \tag{3.7}$$

The term ΔT is a correction for upstream turbulence, which is applied when there are dampers, elbows, or branch takeoffs upstream. This term is applied when the upstream turbulence is within five main duct diameters distance from turn or junction under consideration.

$$\Delta T = -1.667 + 1.8\, m - 0.133\, m^2 \tag{3.8}$$

$$m = \frac{U_M}{U_B} \tag{3.9}$$

K_j the characteristic spectrum in Equation (3.3) can be calculated using;

$$\begin{aligned} K_j = -21.61 + 12.388\, m^{0.673} - 16.482\, m^{-0.303} \log S_t \\ - 5.047\, m^{-0.254} \log {S_t}^2 \end{aligned} \tag{3.10}$$

And the correction tern C_B depends on the junction type. For X-junction;

$$C_B = 20\ \log \frac{D_M}{D_B} + 3 \tag{3.11}$$

And for T junction;

$$C_B = 3 \tag{3.12}$$

For 90^0 elbows without turning vane;

$$C_B = 0 \tag{3.13}$$

For a 90^0 branch takeoff;

$$C_B = 20\ \log \frac{D_M}{D_B} \tag{3.14}$$

3.1.3.3 Air Generated Noise in Dampers

Damper noise can be estimated with same general formula given in Equation (3.3), although the terms in the equation defined differently:

$$L_w\,(f_0) = K_D + 10 \log \frac{f_0}{63} + 50 \log U_C + 10 \log S + 10\ log\ D_H \ + \ C \tag{3.15}$$

Preliminary calculations have to be undertaken before solving Equation (3.15). The characteristic spectrum is determined from Strouhal number S_t, the blockage factor B and the pressure loss coefficient C. Strouhal number is a dimensionless number used for describing the oscillation of air flow. The pressure loss coefficient is expressed as;

$$C = 15.9\ 10^6\ \frac{\Delta P}{(\frac{Q}{S})^2} \tag{3.16}$$

The blockage factor for the multiblade damper is expressed as;

$$B = \frac{\sqrt{C} - 1}{C - 1} \quad for\ C \neq 1 \tag{3.17}$$

or,

$$B = 0.5 \quad for\ C = 1 \tag{3.18}$$

For single blade damper, blockage factor is,

$$B = \frac{\sqrt{C} - 1}{C - 1} \quad for\ C < 4 \tag{3.19}$$

or,

$$B = 0.68\ C^{-0.15} - 0.22 \quad for\ C > 4 \tag{3.20}$$

The constricted velocity is calculated using,

$$U_c = \frac{Q}{60\ S\ B} \tag{3.21}$$

which gives the Strouhal number,

$$S_t = f_0 \frac{D}{U_C} \tag{3.22}$$

After making the preliminary calculation, the characteristic spectrum for dampers can be calculated as;

$$\begin{aligned} K_D &= -36.6 - 10.7 \log S_t \quad for\ S_t \leq 25 \\ K_D &= -1.1 - 35.9 \log S_t \quad for\ S_t > 25 \end{aligned} \tag{3.23}$$

3.1.3.4 Air Generated Noise by Elbows with Turning Vanes

For elbows with turning vanes Equation (3.3) is used with different definition of terms,

$$\begin{aligned} L_w\,(f_0) = \ & K_T + 10 \log \frac{f_0}{63} + 50 \log U_C + 10 \log S \\ & + 10\ log\ D_C \ + 10 \log n \end{aligned} \tag{3.24}$$

The characteristic spectrum K_T [Reynolds 1990];

$$K_T = -47.5 - 7.69 \log {S_t}^{2.5} \tag{3.25}$$

Where the Strouhal number can be calculated using Equation (3.22) and pressure loss coefficient can be calculated using Equation (3.16). Constricted velocity can be obtained from Equation (3.21). The blockage factor is expressed as;

$$B = \frac{\sqrt{C} - 1}{C - 1} \tag{3.26}$$

3.1.4 Air Terminal Noise

Air terminal units (grilles, registers, diffusers) generated noise is of paramount importance in HVAC noise control. Since it is open into the room and directly affects the receiver, it cannot be attenuated by the addition of downstream devices. The only way to reduce diffuser noise is to reduce velocity by means of additional diffusers or increasing the size of the existing diffuser since diffuser noise is dependent mainly to the air velocity through the device. Often diffuser noise is influenced by the upstream flow conditions that can be modified. Pressure equalizing grilles at the entry to the diffuser can help reduce the contribution due to turbulence.

Sound data on diffuser noise is published by manufacturers in terms of NC levels which usually include a receiver room effect sound correction of 10 dB. Whether using NC levels or sound power levels, the designer should also correct manufacturer’s data for actual room effect, location of air devices, and number of air devices used in a specific design. However, these NC levels are only valid for ideal flow conditions. To achieve ideal conditions, flexible ducts must be straight for at least one duct diameter before the connection to the diffuser and must not be pinched or constricted. Ideal condition is often not met in practice when a duct turn, sharp transition, or a balancing damper immediately precedes the entrance to the diffuser. In these cases, airflow is turbulent and noise generated by the device can be substantially higher than the manufacturer's published data (by as much as 12 dB).

3.2 Primary Sound Attenuators

Sound attenuators which are explained in details in this section are classified basically as duct element sound attenuation, plenum and sound attenuators. All these headings cover sub topics related to more specific HVAC elements which have attenuation features.

3.2.1 Duct Element Sound Attenuators

The duct elements covered under this section include acoustically lined and unlined rectangular ducts, lined and unlined round ducts, elbows, duct split loss, duct and reflection loss. Procedures and formulations for obtaining the sound attenuation associated with these elements are presented.

3.2.1.1 Sound Attenuation in Straight Ducts

Sound wave energy decreases through induced motion of duct walls as a sound wave propagates down an unlined duct. The surface impedance is due principally to the wall mass, and the duct loss calculation goes much like the derivation of the transmission loss. Circular sheet metal ducts are much stiffer than rectangular ducts at low frequencies, in their first mode of vibration. Therefore it is difficult to excite circular ducts in low frequency. As a consequence, sound is attenuated in unlined rectangular ducts to a much greater degree than in circular ducts.

Straight unlined rectangular sheet metal ducts contribute significant amount of low-frequency sound attenuation, however circular sheet metal ducts provide more attenuation at high frequency. The attenuation for rectangular and circular sheet metal ducts without internal insulation is given in Table 3-4 and Table 3-5 accordingly [1]. The attenuation values in Table 3-4 and Table 3-5 is applicable only for ducts with the lightest gages allowed according to Sheet Metal and Air

Conditioning Contractors' National Association, Inc. (SMACNA) HVAC duct construction standards.

Table 3-4 Natural Sound Attenuation in Unlined Rectangular Sheet Metal Ducts [1]

P/S 1/mm	Attenuation, dB/m			
	Octave Band Center Frequency, Hz			
	63	125	250	>250
0.026	0.98	0.66	0.33	0.33
0.013	1.15	0.66	0.33	0.20
0.01	1.31	0.66	0.33	0.16
0.007	0.82	0.66	0.33	0.10
0.003	0.49	0.33	0.23	0.07
0.002	0.33	0.33	0..16	0.07

Table 3-5 Natural Sound Attenuation in Unlined Straight Round Ducts [1]

Diameter mm	Approximate Attenuation, dB/m						
	Octave Band Center Frequency, Hz						
	63	125	250	500	1000	2000	4000
D≤180	0.10	0.10	0.16	0.16	0.33	0.33	0.33
180<D≤380	0.10	0.10	0.10	0.16	0.23	0.23	0.23
380<D≤760	0.07	0.07	0.07	0.10	0.16	0.16	0.16
760<D≤1520	0.03	0.03	0.03	0.07	0.07	0.07	0.07

When the duct is externally insulated, the surface mass is increased. And so the low-frequency attenuation is increased, the losses given in Table 3-4 and Table 3-5 have to multiplied by a factor of two.

When a duct is lined with an acoustical absorbent, sound is attenuated through its interaction with the material as discussed earlier. 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.

A regression equation for the insertion loss of rectangular ducts is expressed as;

$$\Delta L_{duct} = B \left(\frac{P}{S}\right)^{C} t^{D}\, l \tag{3.27}$$

Reynolds' equation was based on data obtained using 25mm to 51mm thick liner having a density of 24 to 48 kg/m^3 and constants are given in Table 3-6 [21]. The Equation (3.27) is valid for perimeter per area (P/S) ratio of 1.1667 to 6 in units of feet.

Table 3-6 Constants Used in Equation (3.27) [21]

	Octave Band Center Frequency (Hz)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
B	0.0133	0.0574	0.2710	1.0147	1.7700	1.3920	1.5180	1.5810
C	1.959	1.410	0.824	0.500	0.695	0.802	0.451	0.219
D	0.917	0.941	1.079	1.087	0.000	0.000	0.000	0.000

For the attenuation in the circular lined ducts a third-order polynomial regression formula was developed as follows [24];

$$\Delta L_{duct} = [A + Bt + Ct^2 + Dd + Ed^2 + Fd^3]l \qquad (3.28)$$

The constants A, B, C, D, E and F used in Equation (3.28) are given in Table 3-7.

Table 3-7 Constants Used in Equation (3.28) [21]

Frequency [H z]	A	B	C	D	E	F
63	0.2825	0.3447	−5.251E-2	−3.837E-2	9.13E-04	−8.294E-6
125	0.5237	0.2234	−4.936E-3	−2.724E-2	3.38E-04	−2.490E-6
250	0.3652	0.79	−0.1157	−1.834E-2	−1.211E-4	2.68E-06
500	0.1333	1.845	−0.3735	−1.293E-2	8.62E-05	−4.986E-6
1000	1.933	0	0	6.14E-02	−3.891E-3	3.93E-05
2000	2.73	0	0	−7.341E-2	4.43E-04	1.01E-06
4000	2.8	0	0	−0.1467	3.40E-03	−2.851E-5
8000	1.545	0	0	−5.452E-2	1.29E-03	−1.318E-5

3.2.1.2 Sheet Metal Duct Elbow

Significant high-frequency attenuation can be obtained by elbow, especially if it is lined. Bend turning angle has to be greater than 60^0 so that it is taken up as elbow.

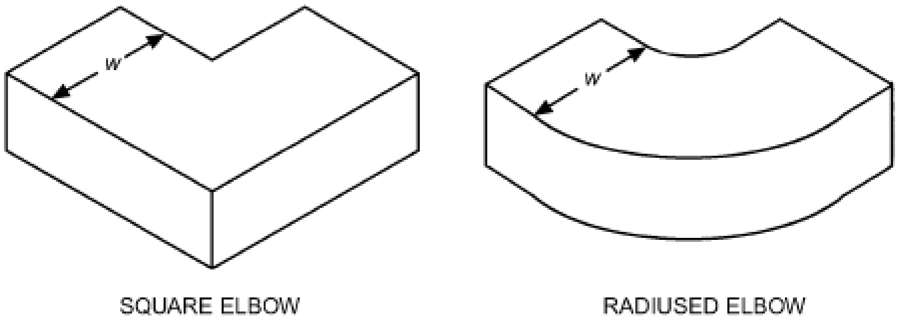


Figure 3-4 Rectangular Duct Elbows [1]

Table 3-8 gives the insertion loss values associated with radiused elbows. Table 3-9 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 3-9 applies only where the duct is lined before and after the elbow. Insertion loss values for unlined and lined square elbows with turning vanes are given in Table 3-10. All the insertion loss values given in Table 3-9 through Table 3-10 is based on the octave band center frequency times the width of the elbow; fw.

Table 3-8 Insertion Loss of Round Elbows [25]

	Insertion loss, dB
fw < 48	0
48≤fw<96	1
96≤fw<190	2
190<fw	3

Table 3-9 Insertion Loss of Unlined and Lined Square Elbows without Turning Vanes [25]

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

Table 3-10 Insertion Loss of Unlined and Lined Square Elbows with Turning Vanes [25]

	Insertion Loss, dB	
	Unlined Elbows	**Lined Elbows**
fw < 48	0	0
48≤fw<96	1	1
96≤fw<190	4	4
190≤fw<380	6	7
380<fw	4	7

3.2.1.3 Duct Split Losses

When sound travelling in a duct encounters a junction, the sound power in the main feeder duct is distributed between the branches accompanying with the junction. The split loss is derived as;

$$\Delta L_{split} = 10\log\left[1-\left(\frac{\sum S_i - S_m}{\sum S_i + S_m}\right)^2\right] + 10\log\left[\left(\frac{S_i}{\sum S_i}\right)\right] \tag{3.29}$$

The corresponding attenuation of sound power that is transmitted down each branch of the junction consists 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 $\sum S_i$ differs from the cross sectional area S_M of the main feeder duct and the frequency is below cutoff. The second component is associated with the energy division among individual branches based on the ratio of the cross sectional area $\sum S_i$ of an individual branch divided by the sum of the cross sectional areas of the individual branches $\sum S_i$. In the expression the second component is dominant for the attenuation of sound power. Values for the attenuation of sound power $\sum S_i$ at a junction that are related to the sound power transmitted down an individual branch of the junction are given in Table 3-11 [1].

Table 3-11 Duct Branch Sound Power Division in dB [1]

$S_i / \sum S_{Bi}$	ΔL_{Bi}	$S_i / \sum S_{Bi}$	ΔL_{Bi}
1	0	0.1	10
0.8	1	0.08	11
0.63	2	0.063	12
0.5	3	0.5	13
0.4	4	0.04	14
0.32	5	0.032	15
0.25	6	0.025	16
0.2	7	0.02	17
0.16	8	0.016	18
0.12	9	0.012	19

3.2.1.4 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. This is to say; 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.

For calculating the end effect attenuation Reynolds [14] published an empirical formula. Its magnitude depends on the size of the duct, measured in wavelengths. The attenuation associated with a duct terminated in free space is;

$$\Delta L_{end} = 10 \log \left[1 + \left(\frac{c_0}{\pi f d} \right)^{1.88} \right] \quad (3.30)$$

And duct terminated flush with wall is;

$$\Delta L_{end} = 10 \log\left[1 + \left(\frac{0.8\, c_0}{\pi f d}\right)^{1.88}\right] \tag{3.31}$$

Where the d is the effective diameter of the duct;

$$d = \sqrt{\frac{4S}{\pi}} \tag{3.32}$$

When the duct ended with a diffuser, end effect attenuation does not occur. Since diffusers smooth the impedance transition between the duct and the room.

3.2.2 Plenums

A plenum is an enclosed space that has a well-defined entrance and exit. Plenum is a part of the air path, and contains an increase and then a decrease in cross-sectional area. A schematic geometry of plenum is shown in Figure 3-5.

Rooms that form part of the air passageway can be modeled as plenums. For instance a mechanical equipment room can be a plenum when the return air circulates through it. In this case the plenum inlet is the intake air opening on the fan. Plenums placed in near fan or in mechanical rooms are generally lined with acoustically absorbent material to reduce fan or other mechanical noise. Plenums are also laid between fan outlet and main duct line to smooth the turbulent airflow.

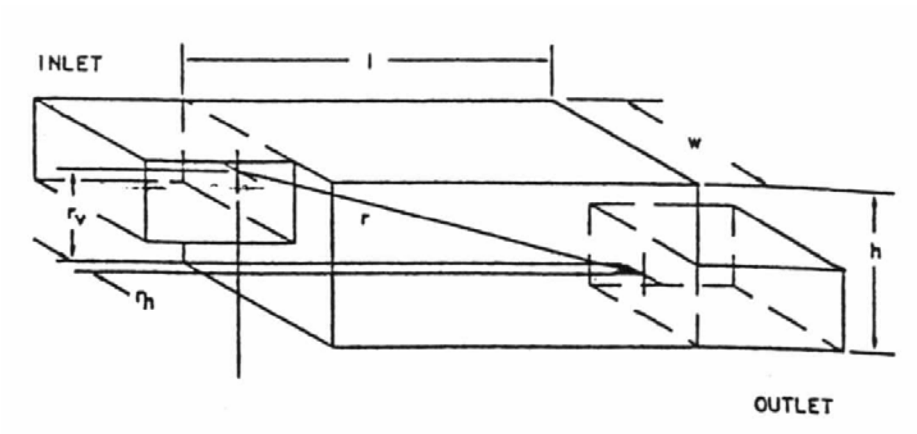


Figure 3-5 Schematic of Plenum Chamber [1]

The cutoff frequency f_{co} is the frequency above which plane waves no longer propagate in a duct. The cutoff frequency for rectangular duct is defined as;

$$f_{co} = \frac{c_0}{2a} \tag{3.33}$$

For circular duct;

$$f_{co} = 0.586\frac{c_0}{d} \tag{3.34}$$

Above the cutoff frequency, as defined by the plenum's inlet duct dimensions, the wavelength of sound is small compared to the characteristic dimensions of the plenum. Plane wave propagation in a duct exists at frequencies below the cutoff. Plenum attenuation is considered in two frequency ranges which are low frequency and high frequency range.

3.2.2.1 Plenum Attenuation-Low Frequency Case

Plenum attenuation depends on the relationship between the size of the cavity and the wavelength of the sound passing through it. Below the cutoff frequency, when the wavelength is large compared with the cross-sectional dimension, a plenum is modeled using plane wave analysis. The plenum attenuation is then given by [35];

$$\Delta L_p = 10 \log \begin{bmatrix} (\cosh(\alpha l) + \frac{1}{2}\left[m + \frac{1}{m}\right] \sinh(\alpha l))^2 \\ x\ cos^2(\frac{2\pi f l}{c_0}) \\ +(\sinh(\alpha l) + \frac{1}{2}\left[m + \frac{1}{m}\right] \cosh(\alpha l))^2 \\ x\ \ sin^2(\frac{2\pi f l}{c_0}) \end{bmatrix} \tag{3.35}$$

Where average absorption coefficient α of the plenum surface is;

$$\alpha = \frac{\sum S_i \alpha_i}{S} \tag{3.36}$$

The transmissivity term m can be written as the area ratio;

$$m = \frac{S_2}{S_1} \tag{3.37}$$

3.2.2.2 Plenum Attenuation-High Frequency Case

When the wavelength is not large compared with the dimensions of the central cross section, the plenum acts like a room and the plane wave model is no longer appropriate. Using the methodology for sound in room, plenum overall transmission loss is established as;

$$\Delta L_p = 10\log\left\{\frac{Q_i S_o cos\theta}{4\pi\left[r+\sqrt{\frac{S_i Q_i}{4\pi}}\right]^2}+\frac{S_o}{R}\right\} \tag{3.38}$$

Where room constant of the plenum is calculated by using average absorption coefficient α given in Equation (3.36),

$$R = \frac{S_p \propto}{(1-\alpha)} \tag{3.39}$$

The value cosθ is obtained from;

$$cos\theta = \frac{l}{r} = \frac{l}{\sqrt{l^2 + {r_v}^2 + {r_h}^2}} \tag{3.40}$$

3.2.3 Duct Silencer

Silencers, sometimes called sound attenuators, sound traps, or mufflers, are designed to reduce the noise transmitted from a source to a receiver. For HVAC applications, the most common silencers are duct silencers, installed on the intake and/or discharge side of a fan or air handler. Also, they may be used on the receiver side of other noise generators such as terminal boxes, valves, dampers, etc. Duct silencers are available in varying shapes and sizes to fit ductwork. Common shapes and configurations of duct silencers include rectangular, round, elbow, tee, and transitional are shown in Figure 3-6.

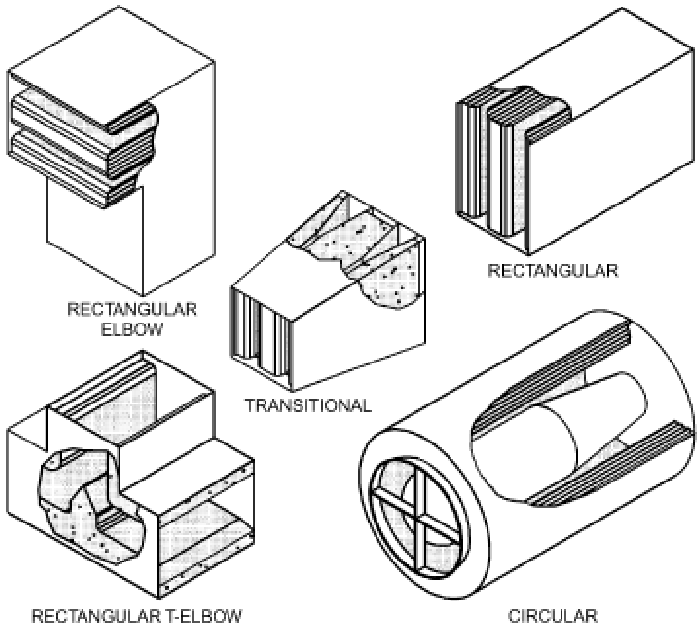


Figure 3-6 Duct Silencer Configuration [1]

Silencers in duct can be rated for the listed four properties below;

- Insertion loss
- Dynamic insertion loss
- Pressure drop
- Generated noise

Insertion loss is the reduction in the sound power level after the silencer in ductwork. Insertion loss is measured as a function of frequency and commonly published in full octave bands ranging from 63 to 8000 Hz.

Dynamic insertion loss is insertion loss with given airflow direction and velocity. A silencer’s insertion loss varies depending on whether sound is traveling in the same or opposite direction as airflow. Silencer performance changes with absolute duct velocity.

Pressure drop is measured across the silencer at a given velocity. Good flow conditions are required at both the inlet and discharge of the silencer. The measuring points are usually 2.5 to 5 duct diameters upstream and downstream of the silencer to avoid turbulent flow areas near the silencer and to allow for any static pressure regain. Airflow-generated noise is the sound power generated by the silencer when quiet air flows through it. This represents the noise base, or the lowest level achievable regardless of high insertion loss values. A silencer's generated noise is a function of frequency and is referenced to specific velocities and airflow direction (forward or reverse). The airflow-generated sound power of the silencer is inversely logarithmically proportional to silencer cross-sectional area.

Silencers create some additional back pressure or flow resistance due to the constriction they present. Generally there is a tradeoff between back pressure and low-frequency attenuation. Sometimes it is necessary to expand the duct to increase the silencer face area and reduce the pressure loss. It is desirable to minimize the silencer back pressure, usually limiting it to less than 10% of the total rated fan pressure. The position of the silencer in the duct also affects the back pressure.

There are three types of HVAC duct silencers; dissipative, reactive and active.

Dissipative silencers use sound absorptive media which is covered by perforated sheet metal. Fiberglass and mineral wool is used as a media for attenuating sound. For dissipative silencers, the attenuation is highest in the mid-frequency range; it is limited in the low and high frequency range. The performance these silencers are a function of silencer length; airflow constriction; number, thickness of splitter; and type of absorptive media.

Reactive silencers are constructed only using metals, both solid and perforated. They use tuned resonators, which are chambers of specially designed shapes and size behind the perforated metal, to reduce sound power at certain frequencies. 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.

Active silencers produce inverse sound wave with opposite phase that cancels noise. They mostly work at low frequency and called noise canceling systems. In these silencers noise in the duct is measured by microphone and converted to electrical signals. Opposite, mirror image sound wave of equal amplitude are generated according to the measured signals. A secondary noise source, generated by silencers, interferes with the primary noise and cancels a significant portion of undesired sound.

CHAPTER 4

IMPLEMENTATION OF SOFTWARE

4.1 Introduction of the Software

This software is developed to predict the sound level passing through the HVAC duct system and reaching the outlet of system. The name of the program is chosen as "HVAC Duct Noise Prediction (HDNP)" so that it reflects the purpose of the program. HDNP projects equipment sound power data through the surrounding (e.g., ductwork) to estimate the sound level that will be the output of the HVAC system. The formulae and data given in Chapter 3 are the basis for this estimate. The programming language is Visual C#.

In calculation sound power level of each element it is assumed that HVAC duct components are installed properly according to relevant regulation and standards. There is no leakage between the connections of duct components.

In HDNP, International System of Units is used. All the units are specified next to each input box in the program.

In HDNP, user can model the conditions of an HVAC system by choosing specific equipment and duct components. HDNP analyzes the sound path and calculate the total sound power level at the outlet of the ductwork. User can continuously adjust the elements used in the program and compare the results effortlessly.

HDNP 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., branch divisions, supply and return system), HDNP determines sound level in each path and branch separately, and gives result accordingly.

It defines the descriptions of equipment and building components that generate, attenuate, reduce or regenerate sound. From the user's input, HDNP models the sound level that will be perceived by the building's occupants. The input of the program is the description of sound paths from HVAC equipment and systems through ductwork. The output of the program is the analysis results in terms of sound power level that can be viewed on screen as a series of detailed tables.

4.2 General Software Structure

Visual C#, an object-oriented programming language, is selected to create code of the HDNP. To briefly summarize the structure of the code is based on divisions of responsibilities for an application or system into individual reusable and self-sufficient objects called class. These classes contain the data and behavior relevant to the objects. A class can be visualized as given in Figure 4-1.

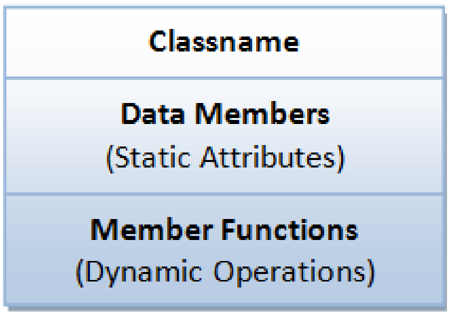


Figure 4-1 Class Properties Block Diagram [36]

An object-oriented design views a system as a series of cooperating objects, instead of a set of routines or procedural instructions. Objects are discrete, independent, and loosely coupled; they communicate through interfaces, by calling methods or accessing properties in other objects, and by sending and receiving messages (Figure 4-2) [37].

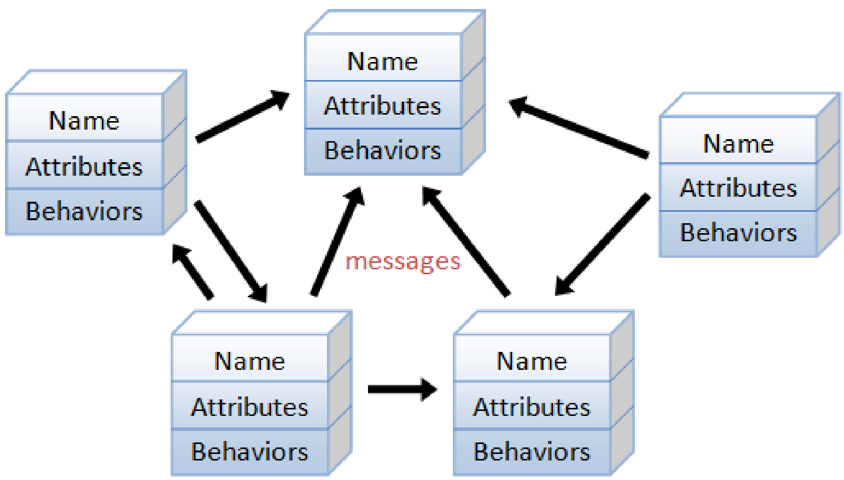


Figure 4-2 An Object-Oriented Program Interaction Schematics

HDNP uses the "Source-Path-Receiver" model to estimate sound power levels in the HVAC system ventilation outlets; it does not cope with the receiver part. In this model, the "source" is the sound-generating device such as fans, and the "receiver" is typically the last element of the ductwork. And the "path" comprises everything that affects the sound as it travels from the source to the receiver in the ductwork. The term elements collectively describe the source, path and receiver components.

HDNP 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, HDNP analyzes each path and outlet individually. Figure 4-3 shows the main general view of the program.

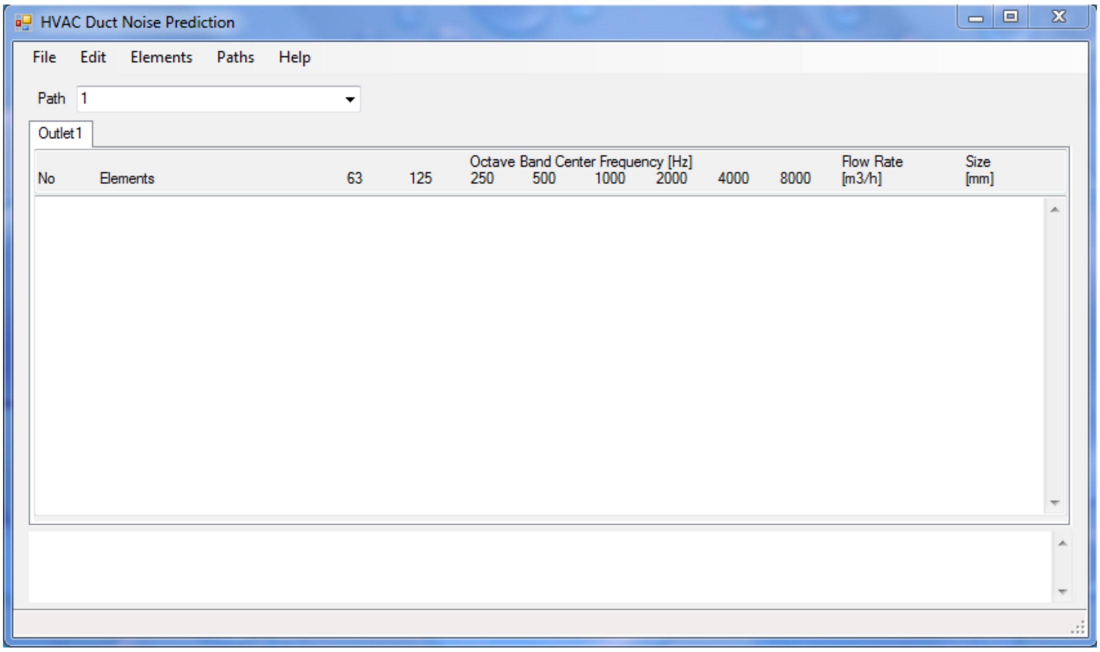


Figure 4-3 Main HDNP Screen View

Main toolbar of the program is composed of five parts and structure of the toolbar is shown in Figure 4-4. Under the 'File' header open, print and exit comments of the program is taking place. 'Edit' toolbar is to edit elements in the path, and is formed by delete and modify comments.

Under elements menu there are three main components such as equipment, duct components and custom. Equipment subgroup is consisting of fan, variable air volume and duct silencer forms. Duct components subgroup is consist of straight duct, elbow, junction, damper, plenum, diffuser and end reflection loss. Under the path toolbar, user can add new path, delete any existing path and also add and remove any branch to the existing path. Under the main toolbar path number and path name screen is placed. And below that the path table is placed and sound power in each octave band is tabulated in.

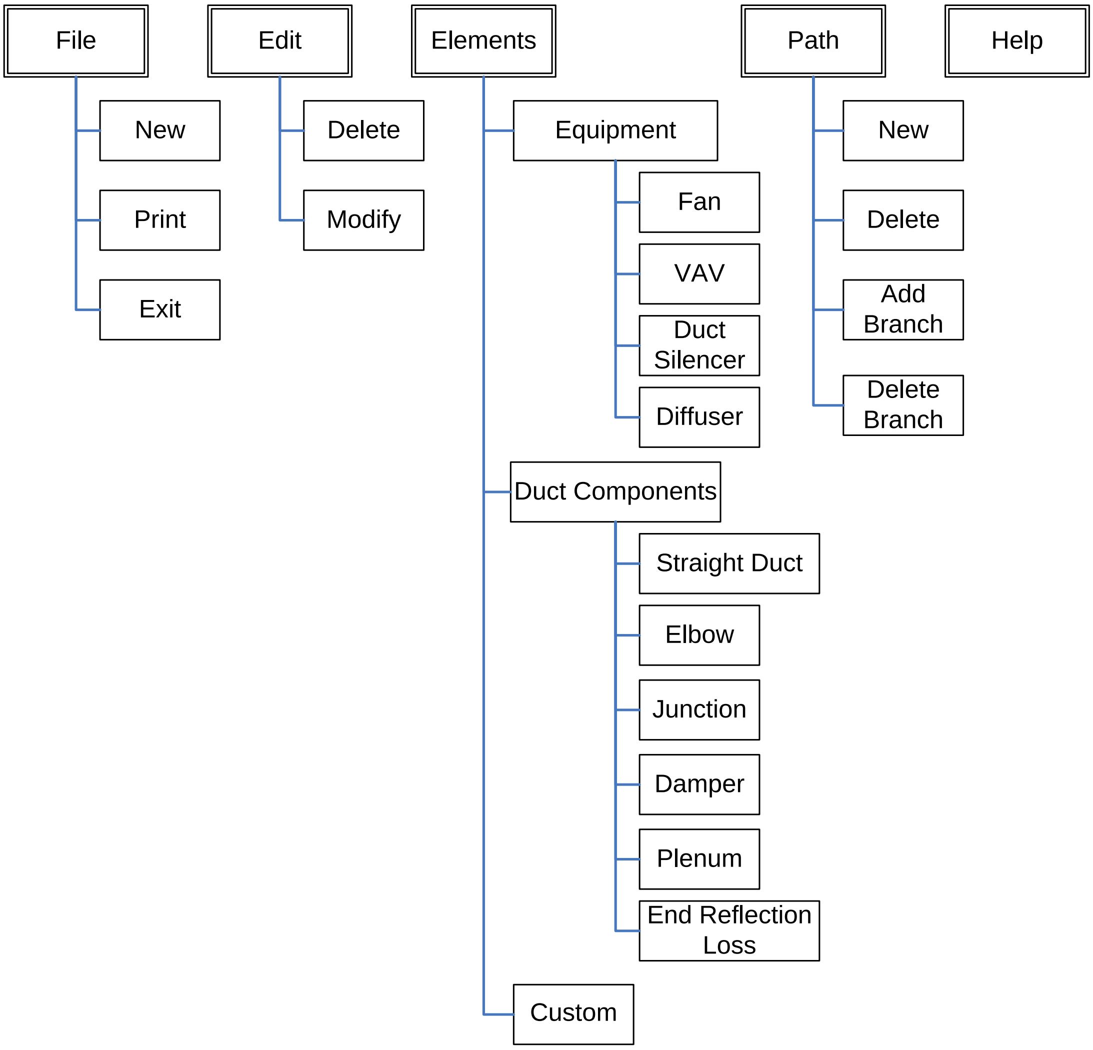


Figure 4-4 HDNP Toolbar Structure

4.3 Implementation of HVAC Duct Elements in the Software

For each element in the path individual form is provided and these forms can be easily activated by clicking the element name in the toolbar.

To introduce the implementation of HVAC duct elements in the software and give an idea about processing of the HDNP, a simple scenario is constructed in this section. In the example sound generator element fan and straight duct with the end reflection loss in the end is introduced as a HVAC sound path.

To model this simple scenario, user has to add new path. Path is workplace in the program that calculations were done. After adding path each elements of the system can be constructed using relevant forms from toolbar menu. As a first element fan toolbar is selected (Figure 4-5) and by filling necessary input for calculation (Figure 4-6) element can be added to the path. After closing the fan form dialog box, HDNP automatically adds fan header and related data in tabular form.

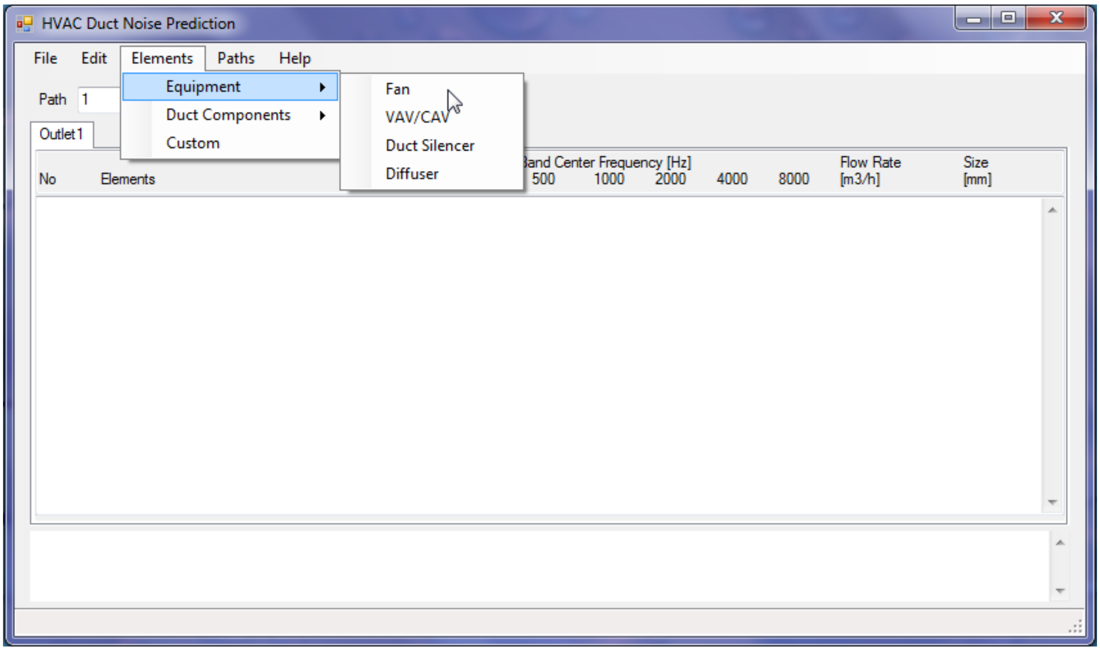


Figure 4-5 Snap Shot from HDNP while Adding Fan Element to a Path

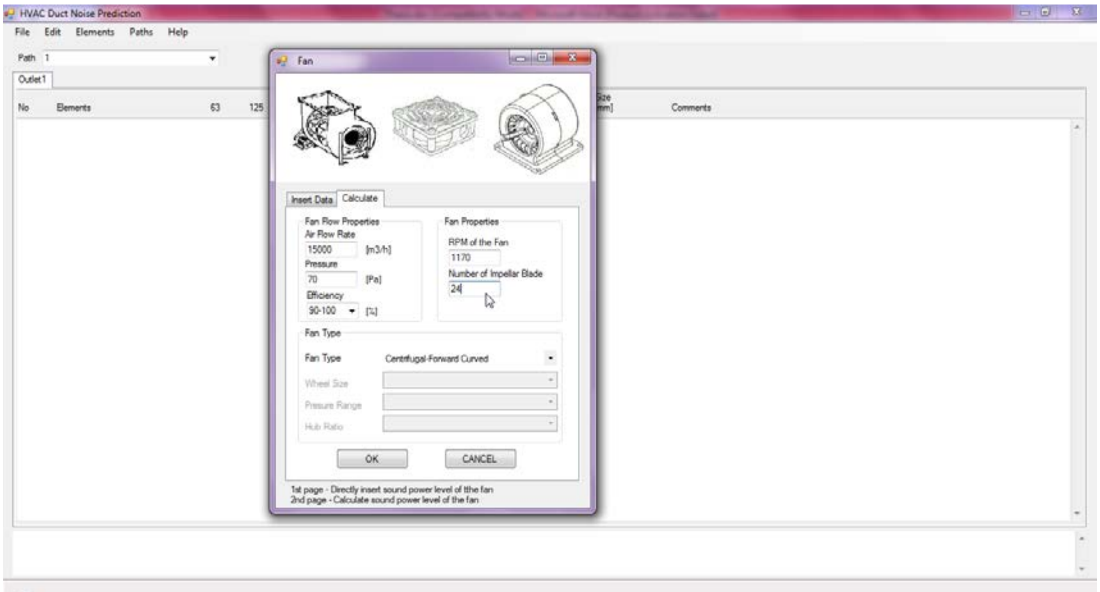


Figure 4-6 Snap Shot from HDNP with Fan Form Pop-up Screen

For the next elements related toolbar menus are selected and inputs are inserted in the forms. When the dialog box of a straight duct (Figure 4-7) and end reflection loss closes (Figure 4-8), HDNP automatically adds elements name and related sound generation and attenuation values respectively in the path table (Figure 4-9).

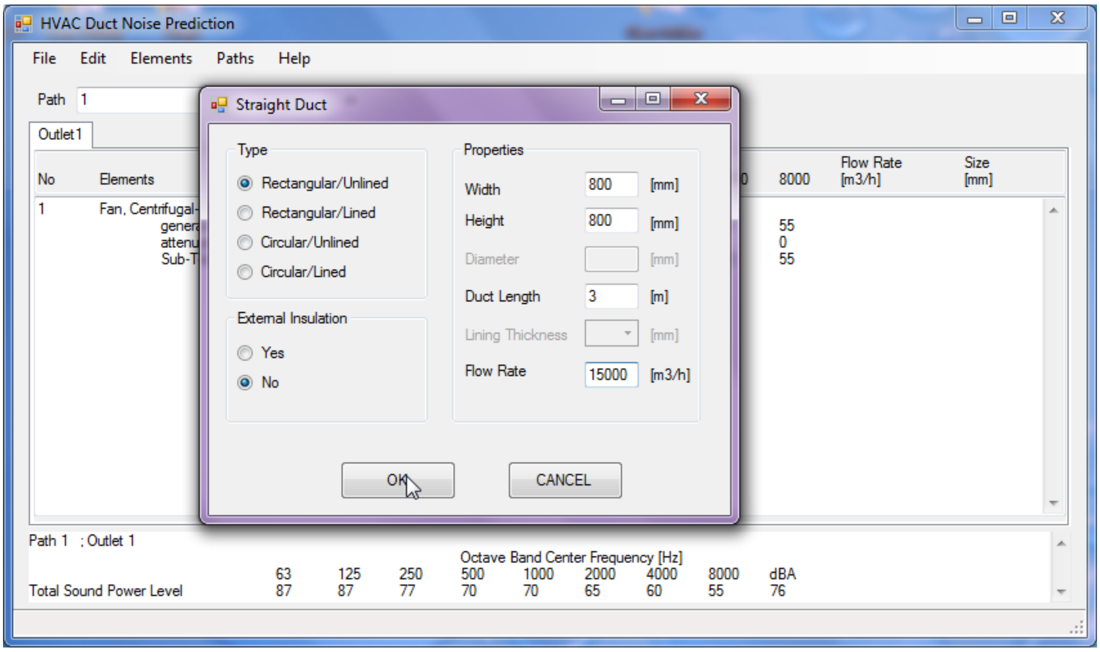


Figure 4-7 Snap Shot from HDNP with Straight Duct Form Pop-up Screen

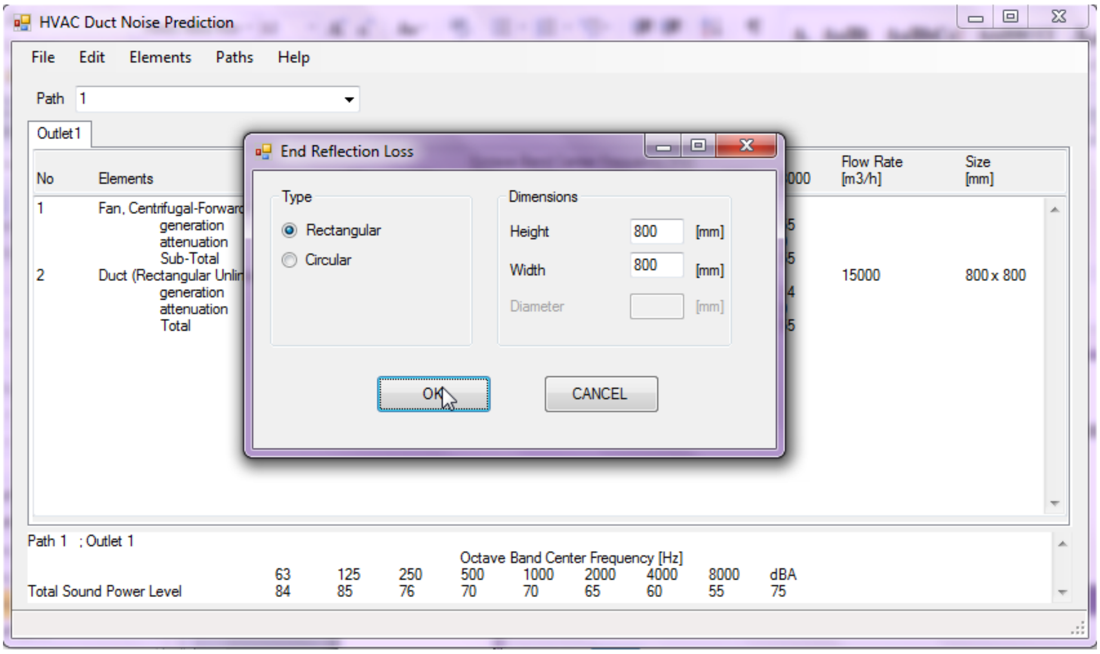


Figure 4-8 Snap Shot from HDNP with End Reflection Loss Form Pop-up Screen

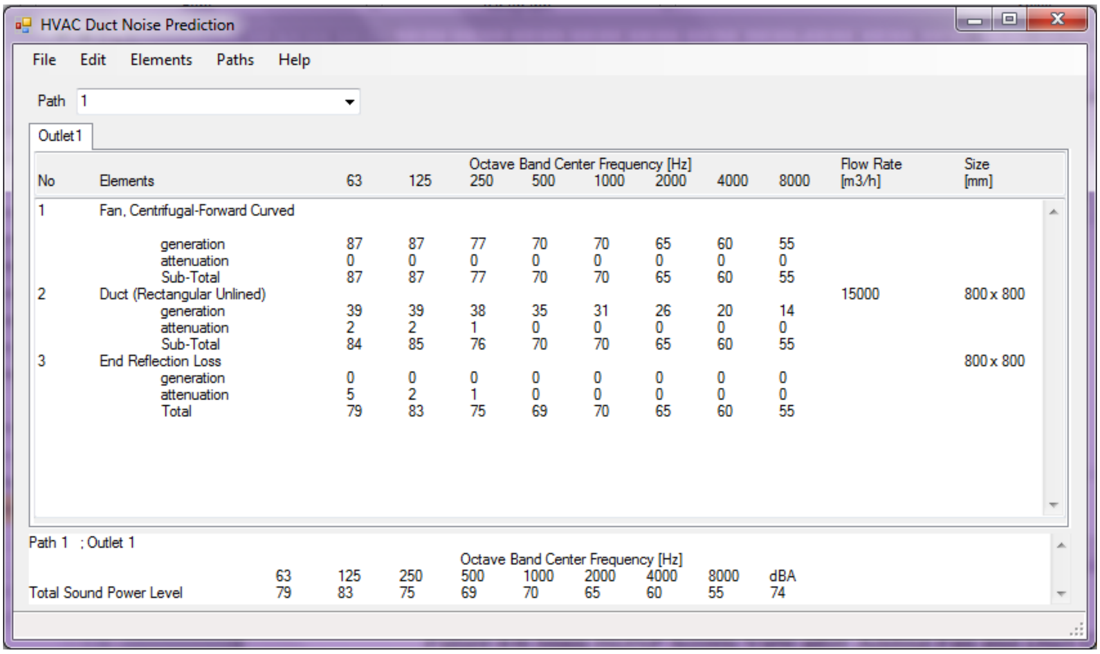


Figure 4-9 A Snap Shot from HDNP with a Complete Path

As can be seen in Figure 4-9 HDNP gives both sound attenuation and generation values for the specific element at the same time. Total calculated sound power level is presented in a different textbox with path name and outlet number.

As components are added, moved or deleted, the program dynamically recalculates the resulting sound power levels. It gives the sub-total sound power level after each element in the path. With line by line display of acoustic calculation of components, contribution of each element to the noise prediction in the ventilation system can be observed.

CHAPTER 5

CASE STUDY

In addition to the knowledge about HDNP software given in Chapter 4, a case study is elaborated to demonstrate the capability and structure of the program.

5.1 Case Study-1

In the case study 1 air is supplied to the HVAC system by the rooftop unit as shown in Figure 5-1. The supply side of the rooftop unit is ducted to a VAV terminal control unit serving the room in question. Path includes fan airborne supply air sound that enters the room from the supply air system through ductwork and the ceiling diffuser.

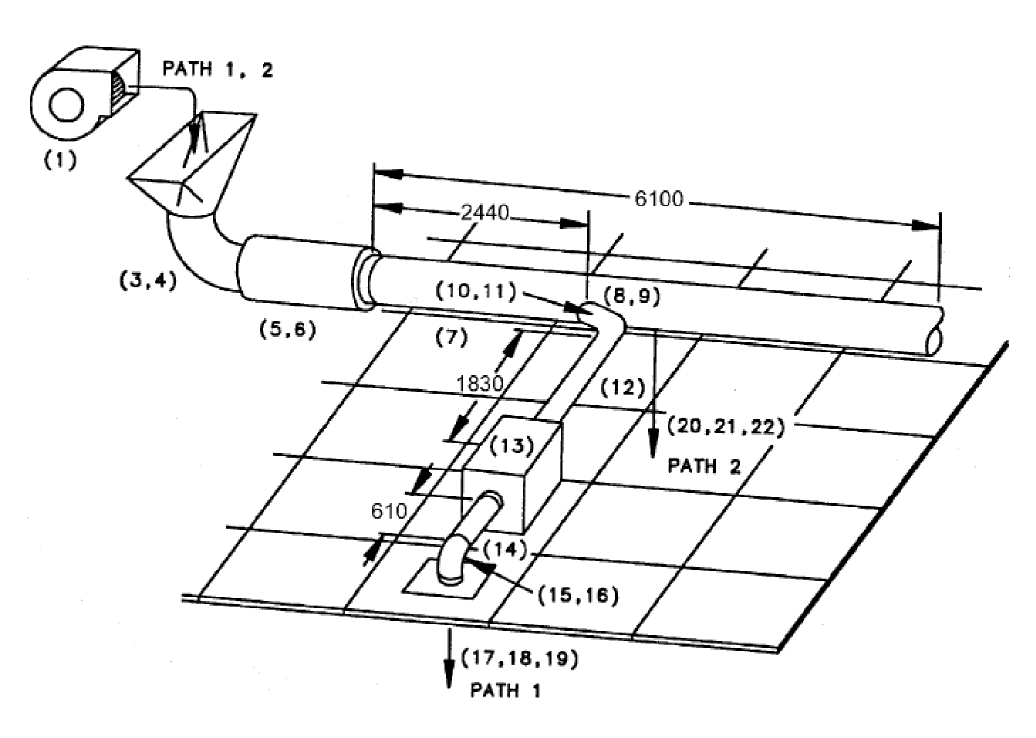


Figure 5-1 Supply Air Layout for Case Study 1 [1]

The main duct of the system is a 560 mm diameter, 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 a 250 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. All the components in the path are tabulated in Table 5-1.

Table 5-1 Case Study 1 Path Components List [1]

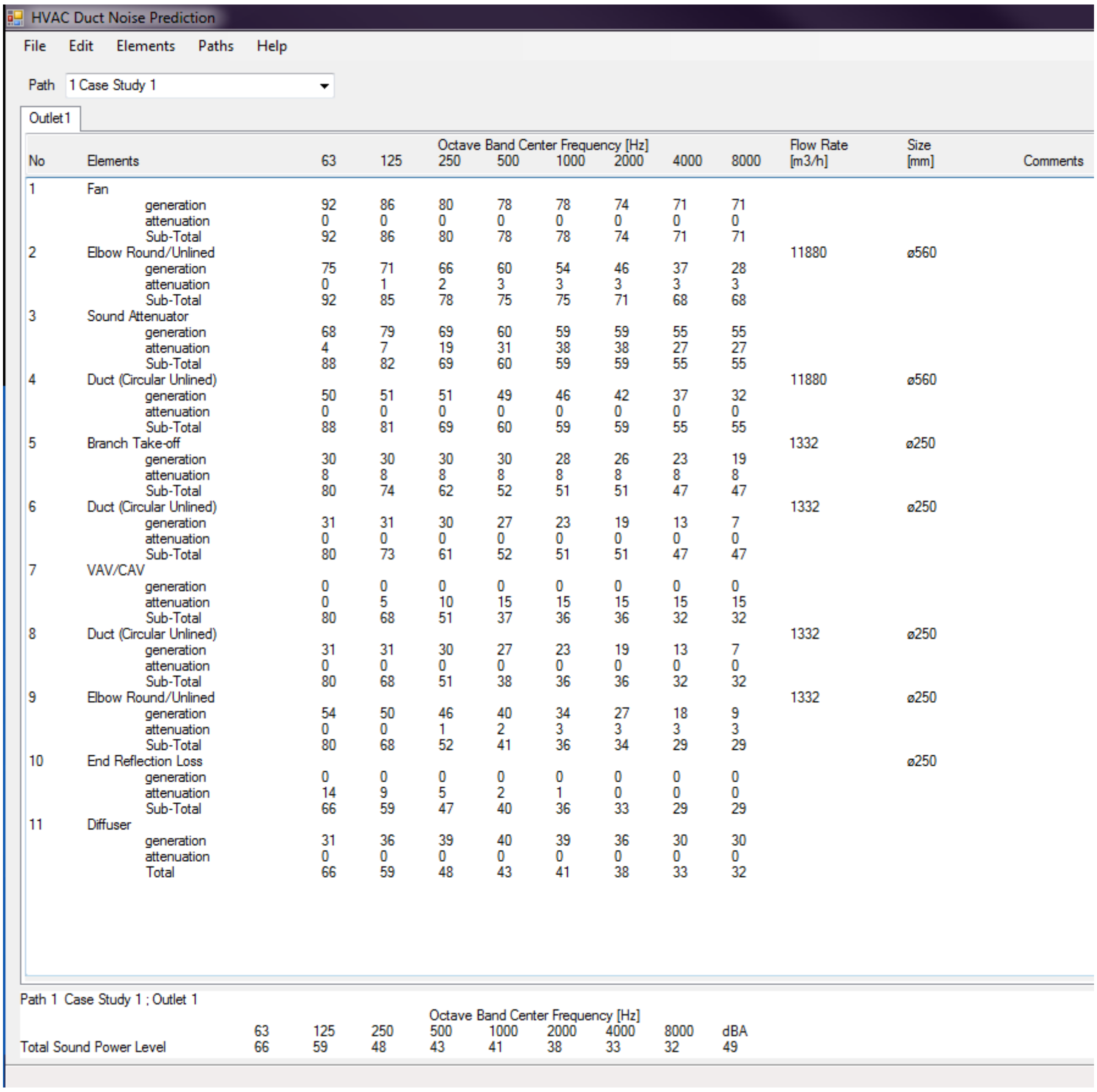
Elements ID	Description	Flowrate [m^3/h]	Size [mm]	Length [m]	Properties
01	Fan	11880	-	-	Supply Air Fan, 11880 m3/h, 620 Pa
02	Elbow	11880	Ø560	-	Unlined radius, 90^0 bend without turning vanes, 320 mm radius
03	Sound Attenuator	11880	-	-	High pressure silencer
04	Duct	11880	Ø560	2.44	Unlined circular
05	Junction	11880 main 1332 branch	Ø560 main Ø250 branch	-	Branch take-off
06	Duct	1332	Ø250	1.83	Unlined circular
07	VAV	1332		-	Terminal volume register unit
08	Duct	1332	Ø250	0.6	Unlined circular
09	Elbow	1332	Ø250	-	Unlined radius, 900 bend without turning vanes, 320 mm radius
10	End Reflection Loss	1332	Ø250	-	Diffuser end reflection loss
11	Diffuser	1332	380x380	-	Rectangular diffuser

The sound power levels, associated with the supply air side of the fan in the rooftop unit, sound attenuator, VAV terminal and diffuser, are taken from manufacturers' data as follows:

Table 5-2 Manufacturer's Data Used in Calculation in Case Study 1 [1]

Elements		Sound Power Level 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	regeneration	92	86	80	78	78	74	71	71
Silencer	attenuation	4	7	19	31	38	38	27	27
	regeneration	68	79	69	60	59	59	55	55
VAV Terminal	attenuation	0	5	10	15	15	15	15	15
Diffuser, 380x380 mm rectangular	regeneration	31	36	39	40	39	36	30	30

With the given data and HVAC system components the overall sound power level at the outlet of the system is calculated. For each component individual form is used in the software and sound power level contribution for all components are tabulated in main screen of the software as shown in Figure 5-2.



HVAC Duct Noise Prediction

File Edit Elements Paths Help

Path 1 Case Study 1

Outlet1

No	Elements		63	125	250	500	1000	2000	4000	8000	Flow Rate [m3/h]	Size [mm]	Comments
			Octave Band Center Frequency [Hz]										
1	Fan												
		generation	92	86	80	78	78	74	71	71			
		attenuation	0	0	0	0	0	0	0	0			
		Sub-Total	92	86	80	78	78	74	71	71			
2	Elbow Round/Unlined										11880	ø560	
		generation	75	71	66	60	54	46	37	28			
		attenuation	0	1	2	3	3	3	3	3			
		Sub-Total	92	85	78	75	75	71	68	68			
3	Sound Attenuator												
		generation	68	79	69	60	59	59	55	55			
		attenuation	4	7	19	31	38	38	27	27			
		Sub-Total	88	82	69	60	59	59	55	55			
4	Duct (Circular Unlined)										11880	ø560	
		generation	50	51	51	49	46	42	37	32			
		attenuation	0	0	0	0	0	0	0	0			
		Sub-Total	88	81	69	60	59	59	55	55			
5	Branch Take-off										1332	ø250	
		generation	30	30	30	30	28	26	23	19			
		attenuation	8	8	8	8	8	8	8	8			
		Sub-Total	80	74	62	52	51	51	47	47			
6	Duct (Circular Unlined)										1332	ø250	
		generation	31	31	30	27	23	19	13	7			
		attenuation	0	0	0	0	0	0	0	0			
		Sub-Total	80	73	61	52	51	51	47	47			
7	VAV/CAV												
		generation	0	0	0	0	0	0	0	0			
		attenuation	0	5	10	15	15	15	15	15			
		Sub-Total	80	68	51	37	36	36	32	32			
8	Duct (Circular Unlined)										1332	ø250	
		generation	31	31	30	27	23	19	13	7			
		attenuation	0	0	0	0	0	0	0	0			
		Sub-Total	80	68	51	38	36	36	32	32			
9	Elbow Round/Unlined										1332	ø250	
		generation	54	50	46	40	34	27	18	9			
		attenuation	0	0	1	2	3	3	3	3			
		Sub-Total	80	68	52	41	36	34	29	29			
10	End Reflection Loss											ø250	
		generation	0	0	0	0	0	0	0	0			
		attenuation	14	9	5	2	1	0	0	0			
		Sub-Total	66	59	47	40	36	33	29	29			
11	Diffuser												
		generation	31	36	39	40	39	36	30	30			
		attenuation	0	0	0	0	0	0	0	0			
		Total	66	59	48	43	41	38	33	32			

Path 1 Case Study 1 ; Outlet 1

	63	125	250	500	1000	2000	4000	8000	dBA
	Octave Band Center Frequency [Hz]								
Total Sound Power Level	66	59	48	43	41	38	33	32	49

Figure 5-2 A Snap Shot from Full Path Analysis Result

When the results are compared with the ASHRAE Handbook [1] example supply air path calculation, a close result can be observed. The predicted sound power level with HDNP at 250, 500 and 1000 Hz is 1-2 dB greater than the ASHRAE calculation. Main reason for that is in the ASHRAE calculation the noise generation from duct elements such as elbows and junction are not taken into account. When the generation calculation for elbow and junction in HDNP closed same result can be obtained as ASHRAE prediction.

Table 5-3 Calculated Sound Power Level Comparison of ASHRAE [1] and HDNP

	Sound Power Level in dB							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
ASHRAE prediction [1]	66	60	46	41	40	38	33	-
HDNP prediction	66	59	48	43	41	38	33	32
HDNP prediction*	66	59	46	41	41	38	33	32

* prediction result with omitted elbow and junction sound generation

5.2 Case Study-2

For case study 2 a real life ventilation system project for meeting room is chosen. Air is circulated by the air handling unit and both supply and return air system is ducted as shown in Figure 5-3. In this case study calculations are made only for the first outlet of the supply air system and compared with Trane Acoustic Program.

Path includes fan airborne supply air sound that enters the room from the supply air system through ductwork and the ceiling slot diffuser. All the components in the path are tabulated in Table 5-4.

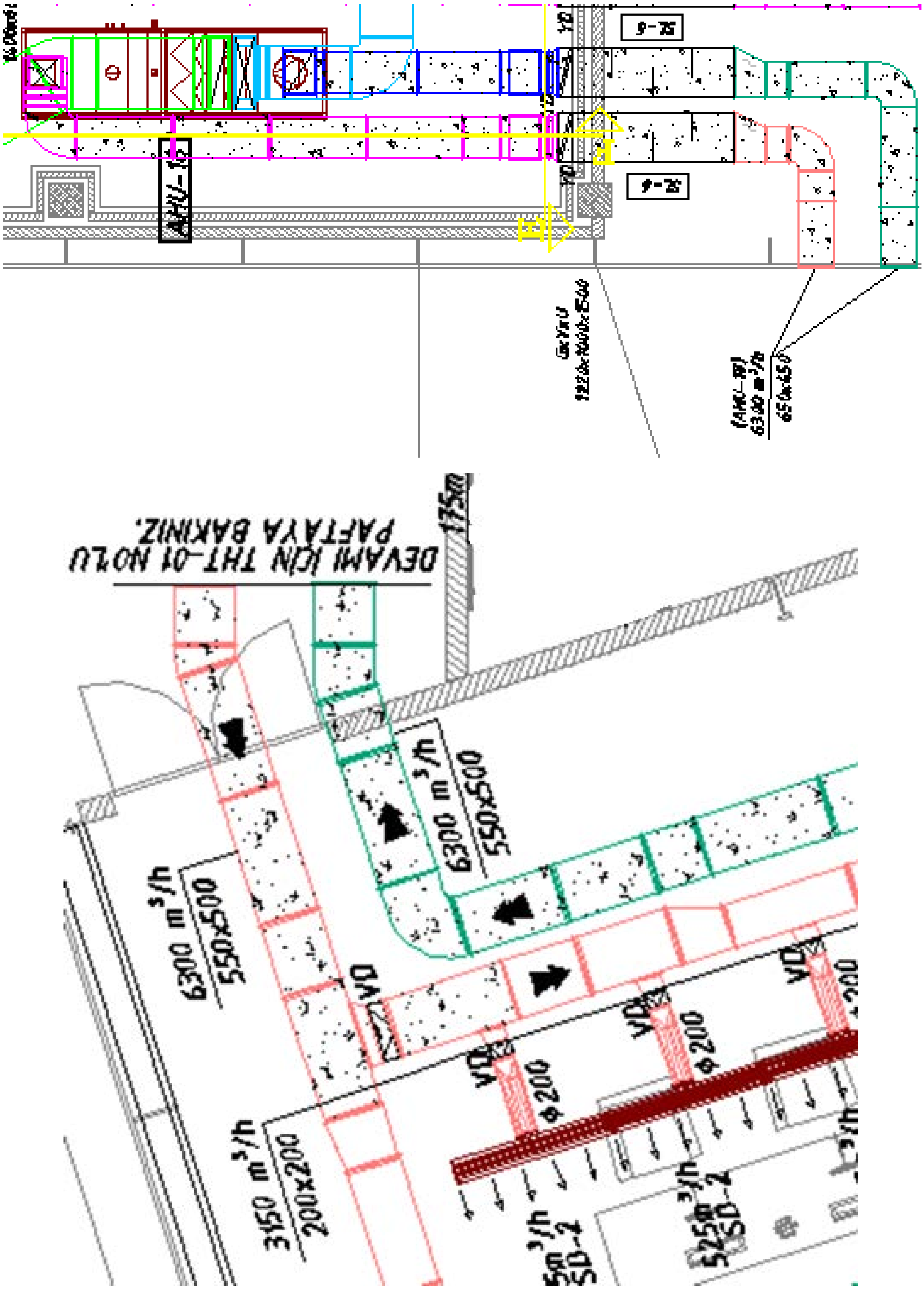


Figure 5-3 Ventilation System Layout for Case Study 2 [1]

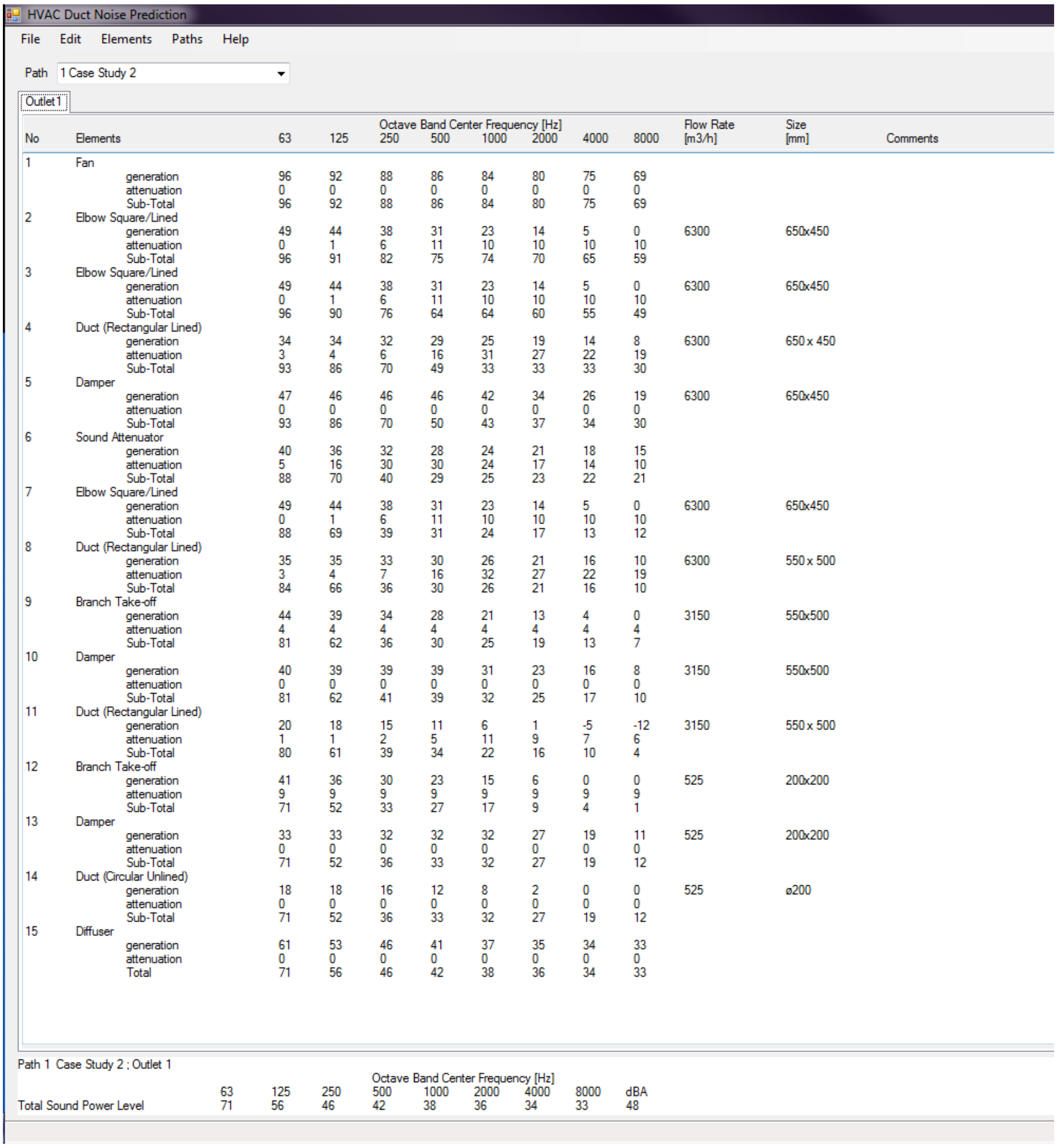
Table 5-4 Case Study 2 Path Components List

Elements ID	Description	Flowrate [m^3/h]	Size [mm]	Lenght [m]	Properties
1	AHU 18	6300	-	-	Supply Air Fan, 174 Pa
2	Elbow	6300	650 x 450	-	Lined radius, 90^0 bend without turning vanes
3	Elbow	6300	650x450	-	Lined radius, 90^0 bend without turning vanes
4	Straight Duct	6300	650x450	7.2	Rectangular lined
5	Damper	6300	650x450	-	Fire Damper, Multiblade
6	Sound Attenuator	6300		-	Splitter attenuator type
7	Elbow	6300	650x450	-	Lined radius, 90^0 bend without turning vanes
8	Straight Duct	6300	550x500 mm	3.4	Rectangular lined
9	Junction	3150 - main continuation,	550x500 - main continuation,	-	90^0 take-off
10	Damper	3150	550x500	-	Multiblade
11	Straight Duct	3150	550x500	1	Rectangular lined
12	Junction	2615 - main continuation,	550x500 - main continuation,	-	90^0 take-off
13	Damper	525	200x200	-	Multiblade
14	Straight Duct	525	Ø200	0.5	Circular unlined
15	Diffuser	525	-	1.5	Slot Diffuser

The sound power levels, associated with the supply air side of the fan in the air handling unit, sound attenuator and slot diffuser, are taken from manufacturers' data as follows:

Table 5-5 Manufacturer's Data Used in Calculation in Case Study 2

Elements		**Sound Power Level in dB**							
		Octave band center frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
AHU 18 Supply Air = 6300 m^3/h at 174 Pa	regeneration	96	92	88	86	84	80	75	69
Sound Attenuator	attenuation	5	16	30	30	24	17	14	10
	regeneration	40	36	32	28	24	21	18	15
Slot Diffuser = 525 m^3/h 1.5m in length, NC36	regeneration	61	53	46	41	37	35	34	33



HVAC Duct Noise Prediction

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Path 1 Case Study 2

Outlet1

No	Elements	63	125	250	500	1000	2000	4000	8000	Flow Rate [m3/h]	Size [mm]	Comments
				Octave Band Center Frequency [Hz]								
1	Fan											
	generation	96	92	88	86	84	80	75	69			
	attenuation	0	0	0	0	0	0	0	0			
	Sub-Total	96	92	88	86	84	80	75	69			
2	Elbow Square/Lined											
	generation	49	44	38	31	23	14	5	0	6300	650x450	
	attenuation	0	1	6	11	10	10	10	10			
	Sub-Total	96	91	82	75	74	70	65	59			
3	Elbow Square/Lined											
	generation	49	44	38	31	23	14	5	0	6300	650x450	
	attenuation	0	1	6	11	10	10	10	10			
	Sub-Total	96	90	76	64	64	60	55	49			
4	Duct (Rectangular Lined)											
	generation	34	34	32	29	25	19	14	8	6300	650 x 450	
	attenuation	3	4	6	16	31	27	22	19			
	Sub-Total	93	86	70	49	33	33	33	30			
5	Damper											
	generation	47	46	46	46	42	34	26	19	6300	650x450	
	attenuation	0	0	0	0	0	0	0	0			
	Sub-Total	93	86	70	50	43	37	34	30			
6	Sound Attenuator											
	generation	40	36	32	28	24	21	18	15			
	attenuation	5	16	30	30	24	17	14	10			
	Sub-Total	88	70	40	29	25	23	22	21			
7	Elbow Square/Lined											
	generation	49	44	38	31	23	14	5	0	6300	650x450	
	attenuation	0	1	6	11	10	10	10	10			
	Sub-Total	88	69	39	31	24	17	13	12			
8	Duct (Rectangular Lined)											
	generation	35	35	33	30	26	21	16	10	6300	550 x 500	
	attenuation	3	4	7	16	32	27	22	19			
	Sub-Total	84	66	36	30	26	21	16	10			
9	Branch Take-off											
	generation	44	39	34	28	21	13	4	0	3150	550x500	
	attenuation	4	4	4	4	4	4	4	4			
	Sub-Total	81	62	36	30	25	19	13	7			
10	Damper											
	generation	40	39	39	39	31	23	16	8	3150	550x500	
	attenuation	0	0	0	0	0	0	0	0			
	Sub-Total	81	62	41	39	32	25	17	10			
11	Duct (Rectangular Lined)											
	generation	20	18	15	11	6	1	-5	-12	3150	550 x 500	
	attenuation	1	1	2	5	11	9	7	6			
	Sub-Total	80	61	39	34	22	16	10	4			
12	Branch Take-off											
	generation	41	36	30	23	15	6	0	0	525	200x200	
	attenuation	9	9	9	9	9	9	9	9			
	Sub-Total	71	52	33	27	17	9	4	1			
13	Damper											
	generation	33	33	32	32	32	27	19	11	525	200x200	
	attenuation	0	0	0	0	0	0	0	0			
	Sub-Total	71	52	36	33	32	27	19	12			
14	Duct (Circular Unlined)											
	generation	18	18	16	12	8	2	0	0	525	ø200	
	attenuation	0	0	0	0	0	0	0	0			
	Sub-Total	71	52	36	33	32	27	19	12			
15	Diffuser											
	generation	61	53	46	41	37	35	34	33			
	attenuation	0	0	0	0	0	0	0	0			
	Total	71	56	46	42	38	36	34	33			

Path 1 Case Study 2 ; Outlet 1

	63	125	250	500	1000	2000	4000	8000	dBA
				Octave Band Center Frequency [Hz]					
Total Sound Power Level	71	56	46	42	38	36	34	33	48

Figure 5-4 A Snap Shot from Full Path Analysis Result for Case Study 2 by HDNP

When analysis results are examined, it can be seen that the critical element in the path is diffuser. Up to the last element the sound generation from the fan is absorbed fairly except in low frequency. Total sound power level obtained for the first outlet of the ventilation system is contributed mostly from diffuser sound generation. Moreover, there is no sound attenuation element after diffuser.

Trane Acoustic Program results for this case study are also given in Figure 5-5 and compared with HDNP in Table 5-6. It can be said that a good consistence can be observed with HDNP and TAP. There exists 1 dB difference in some frequency, mainly occurs from roundup errors in each program.

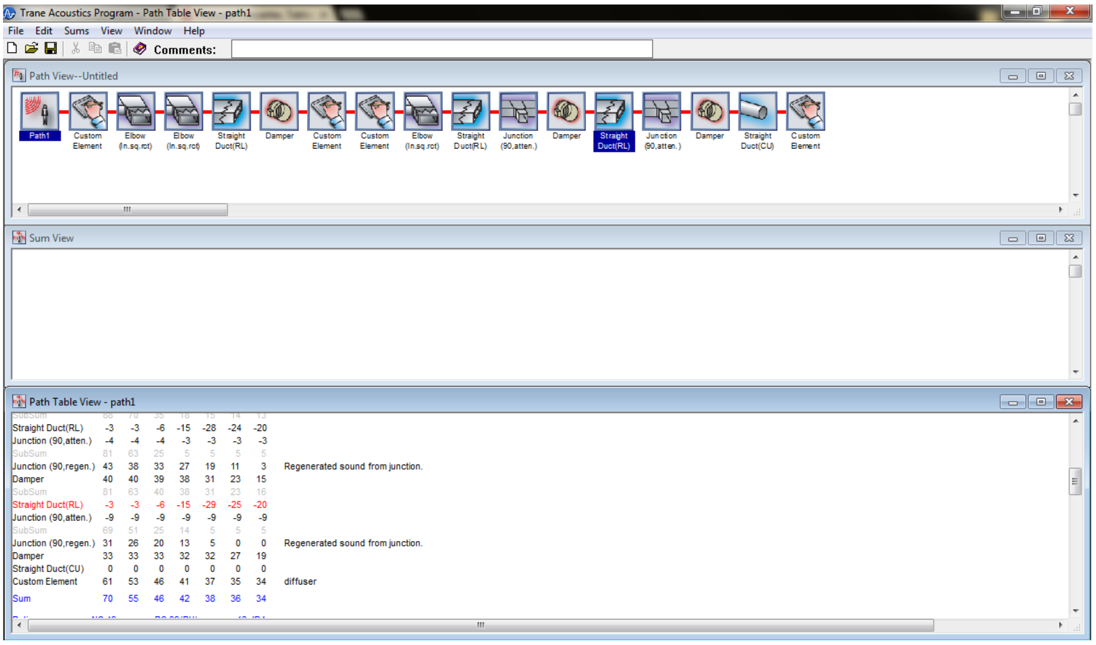

Figure 5-5 A Snap Shot from Full Path Analysis Result for Case Study 2 by Trane Acoustic Program

Table 5-6 Calculated Sound Power Level Comparison of Trane Acoustic Program [27] and HDNP

	Sound Power Level in dB							
	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
TAP prediction [27]	70	55	46	42	38	36	34	-
HDNP prediction	71	56	46	42	38	36	34	33

CHAPTER 6

CONCLUSION

6.1 Summary and Conclusions

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. The acquired formulation and data in question obtained from this survey was exhibited in Chapter 3 and implemented in the algorithm of the prediction software. In addition to defined equipment and components a “custom” entry dialog box was included in the program to enable the user to define custom sound generators or attenuators for the elements.

It is important for such a program to provide flexibility to the user to quickly introduce a path, edit elements in the path when the user needs to change one or more parameters and to try other possible solutions for system design purposes.

A developed program is developed using Visual C#, and it is Windows based application. With utilizing simple dialog box for all elements and easy to use interfaces, HDNP provides user-friendly usage.

HDNP gives the sub-total sound power level after each element in the path and the sound power level at the ventilation system outlet. With line-by-line display of noise calculation of components in the prediction procedure, contribution of each element to the noise prediction in the ventilation system can be observed.

When the simulation outcomes of the software are examined, it works properly and gives a quick response to the user in calculating noise in ducted HVAC system. It also helps HVAC designer to evaluate the ventilation system ductwork in terms of sound power level and gives valuable input for acoustical design of the premises.

6.2 Recommendation for Future Work

Future work is recommended about the improvement of the software to predict the pressure loss in ventilating system. Since pressure loss in the component and operating point of fan directly influence the acoustic performance of the system, doing both pressure loss and noise calculation in software will reduce design and diagnostic time consumed in design phase of the HVAC system.

An option for changing language and unit system in the program can be implemented to improve ease of use.

As another future enhancement of the software HDNP, optimization feature can be added to enable the program for making suggestions to improve sound level of the path. In another words, to achieve specific sound power level in the ventilation system outlet, optimization tool can be adapted to give alternative, such as adaptation of silencer, lining the ductwork.

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