Duct-borne Noise & Vibration onboard Maritime Vessels for Underwater Radiated Noise Management

An Internship Report submitted in partial fulfilment

of requirements for the degree of

Bachelor of Technology

By

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Under the guidance of

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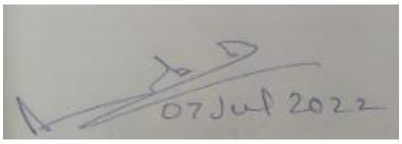
CERTIFICATE

Maritime Research Center, Pune

This is to certify that **Mr. Atharva Vikrant Nagarkar** has successfully completed the project titled **" Duct-borne Noise & Vibration onboard Maritime Vessels for Underwater Radiated Noise Management"** under my supervision for his semester-long project internship at **MRC, Pune**.

Location: Pune



Dr. (Cdr.) Arnab Das
Founder & Director
Maritime Research Center, Pune

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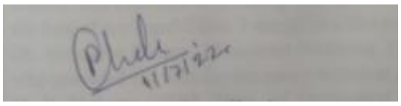

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~ Atharva Vikrant Nagarkar

ABSTRACT

Noise & Vibration studies are one of the most important aspects of designing any dynamic system. Any machinery which has components having high inertia, rotating components, friction, losses and other random forces create excitations at various locations for a system. Ducts are an excellent carrier of various types of vibrations which are created by these excitation forces and carry them to various locations nearby. Thus, if excessive noise levels are generated and transmitted to different ship locations, there are possibilities of detrimental effects such as hearing disabilities for crew and passengers onboard, failure of equipments due to resonance, noise emission to surroundings, violation of stealth requirements etc.

This project focuses on primarily understanding the sources of noise & vibrations onboard marine vessels, transmission of these vibrations to various locations on ships via ducts and attenuation of the generated noise. Using the guidelines established by various classification societies such as American Bureau of Shipping, Bureau Veritas, Lloyd’s Register etc., a software package written in Python is developed to calculate the theoretical attenuation across the duct system for a case under consideration. A harmonic acoustic model is created in ANSYS software to replicate this noise & vibration scenario onboard ships for various duct subsystems.

After the simulation is carried out, a comparison of the output of the Python package and ANSYS model is carried for attenuation of noise levels. This provides a basis for establishing the accuracy of the simulated model. It is found that the error is minimal and the 2 output results are in sync. Thus, this enables us to further improve these models for various ships and ensure the noise levels are restricted within the required limits.

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1. INTRODUCTION

Energy is found in nature in various forms such as mechanical, thermal, electrical etc. Most of these forms of energy are used as sources to do work or as intermediaries to convert energy from one form to another for certain applications. Sound is a medium through which energy propagates by means of oscillation of particles. Vibrations are mechanical oscillations or the intermittent motion of a particle or body, resulting when it is moved from its equilibrium condition. The necessary conditions for oscillatory motion are elasticity and inertia. Elasticity is the ability of a body to return its equilibrium position after it is displaced while inertia is the measure of tendency of a body to resist change in its current state of motion. In order for the particles to fluctuate around their default position, the medium in which sound waves propagate must have both inertia and elasticity. Vibrations can be quantified by measuring displacement, velocity or acceleration in millimetres (mm). Noise is defined as being unwanted sound and is generally a result of vibrations. Noise is measured in frequency and amplitude using Sound Level Meters and its unit is decibels (dB) [1]. By measuring the amplitude of noise, we can identify the precise force or energy of the sound wave and the measure of amplitude indicates the intensity of noise. The pitch or frequency of noise can also be measured in hertz (Hz). Noise and vibration (N&V) analysis is a combination of computational and experimental procedures to measure noise and vibration levels for a system, compare the obtained values with a standard reference and monitor these real-time values over a period of time to detect any possible failure within the system. The marine vessel represents a very complex system of N&V sources determined by the operation of numerous on-board installations and specific activities of crew and passengers. Thus, noise and vibration on-board marine platforms is a critical research area that has multiple applications to it.

1.1 About the Maritime Research Center

The Maritime Research Center (MRC), Pune is a not-for-profit, Section 8 company registered with the Registrar of Companies (RoC), as Foundation for Underwater Domain Awareness. It serves as an inclusive platform for progressing the Underwater Domain Awareness (UDA) Framework and offers a convergence of key stakeholders connected with Maritime Security, Blue Economy, Marine Environment & Disaster Management and Research & Innovation. The comprehensive framework of UDA presented by MRC has been wholeheartedly embraced by key stakeholders for driving discourse and actionable initiatives. MRC is well positioned to fulfill the role as a nodal agency for policy and technology interventions as well as acoustic capacity & capability building requirements for all in the Indo-Pacific Strategic Space and beyond [2].

1.1.1 Think Tank

Any new initiative requires a conducive policy environment to survive and thrive. Policy interventions have to be nuanced and inclusive based on the ground realities. The stakeholders and other associated entities need to synergies their efforts in a seamless manner. The UDA framework proposed by the MRC, encourages pooling of resources and synergizing of efforts across the stakeholders and policy makers [2].

1.1.2 Tech Tank

The tropical littoral waters of the Indo-Pacific Region present very unique challenges and opportunities. The technology intervention has to be a balanced mix of traditional knowledge and the state-of-the-art R&D inputs driven by innovation and Science & Technology tools. The UDA framework driven by the acoustic signal processing, underwater robotics and Artificial Intelligence (AI) based data analytics hardware and software [2].

1.1.3 Skill Tank

The challenges and opportunities in the UDA framework is massive and will require significant human resource to manage the safe, secure, sustainable growth model. Acoustic Capacity and Capability building is a major requirement backed by skill and knowledge-based learning. Multi-disciplinary and multi-functional manpower with adequate and appropriate level of experience and learning will have to be ready to take on this responsibility [2].

1.2 Problem Statement

Develop an acoustic model to illustrate the noise and vibration scenario onboard marine platforms and validate the accuracy of the model by comparing the attenuation of noise levels given by the model with the theories established by the governing authorities (classification societies).

1.3 Methodology

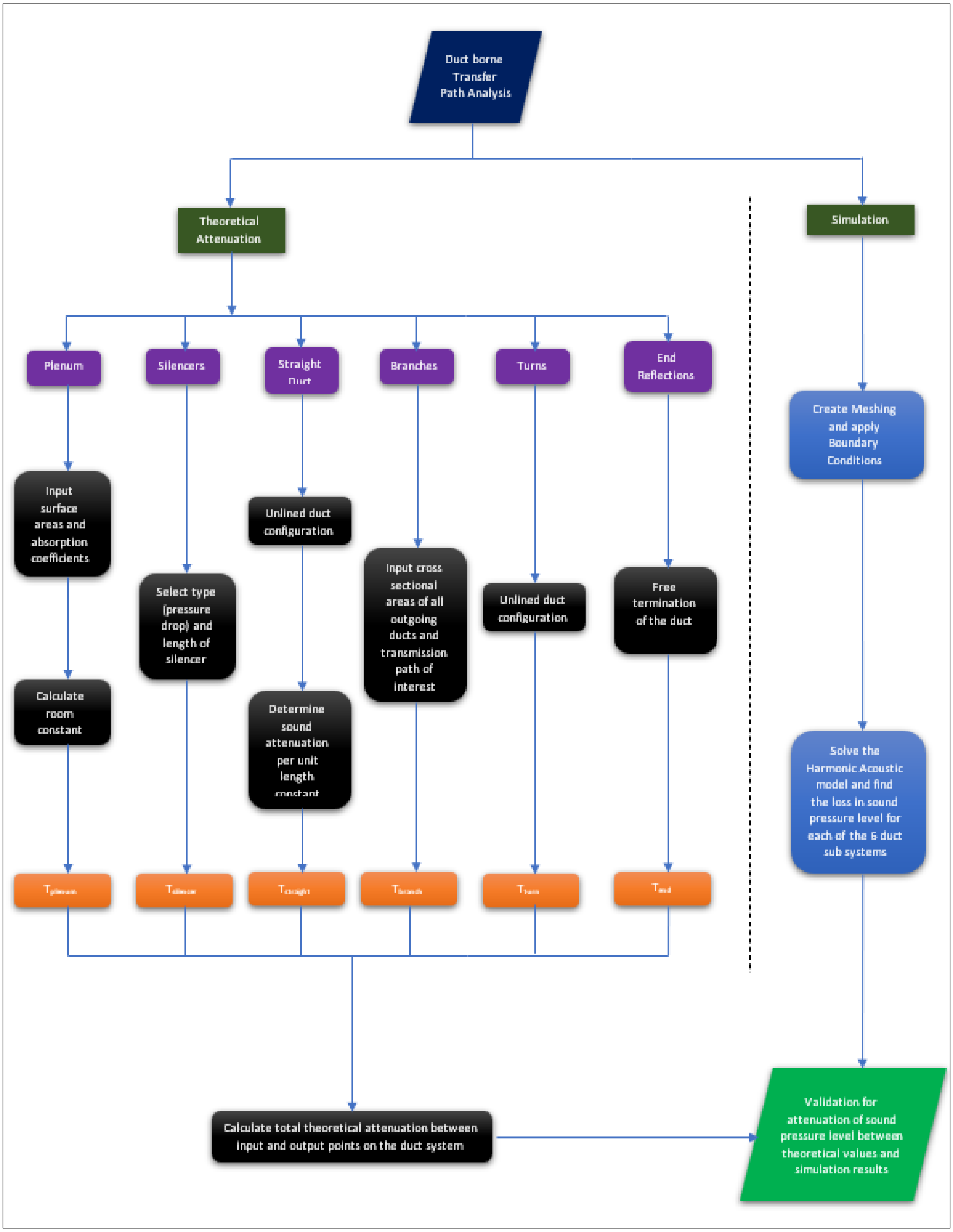

Figure 1: The methodology followed for the project

2. FUNDAMENTALS OF NOISE & VIBRATION ON SHIPS

2.1 Basic Terminologies

2.1.1 Sound Pressure Level

Sound pressure is the pressure measured within the wave relative to the surrounding air pressure. Loud sounds produce sound waves with relatively large sound pressures, while quiet sounds produce sound waves with relatively small sound pressures. Sound pressure level uses a logarithmic scale to represent the sound pressure of a sound relative to a reference pressure. The reference sound pressure is typically the threshold of human hearing [3]. Sound pressure level (SPL) is measured in decibels (dB) using a logarithmic scale as follows:

$$L_p = 20 \log \frac{P}{P_o}$$

Where,

P is the measured root mean square sound pressure level in Pascals (Pa)

P_o is the reference sound pressure level (20 x 10-6 Pa)

2.1.2 Broadband and Narrowband

Vibration amplitudes are often measured using a simple digital or analogue meter giving a single value representing either the peak or root mean square (r.m.s.) amplitude across a range of frequencies (defined by the characteristics of the transducer and meter). This is known as the overall or broadband value. Conversely, a narrowband measurement is one which is limited to a small range of frequencies usually centred on a frequency of interest. The smallest width is determined by the resolution of the analyser [3].

2.1.3 Octave Band

The frequency range of onboard noise analyses for ships and offshore units is usually from 31.5 Hz to 8000 Hz. Considering the large span of the frequency range, it is usually subdivided into frequency bands. The most common one is the octave frequency band. The octave band frequencies cover the normal range of human hearing frequencies. Octave bands are comparatively coarse and use is sometimes made of third-octave bands. The center frequencies of the octave bands for onboard noise analyses are typically: 31.5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1,000 Hz, 2,000 Hz, 4000 Hz and 8,000 Hz [3, 7]. The following equations show the relationship between center frequency and upper/lower bound frequency of an octave band:

$$\frac{f_u}{f_l} = 2 \qquad\qquad f_c = \sqrt{f_u \times f_l}$$

Where,
f_u = upper bound frequency in Hz
f_l = lower bound frequency in Hz
f_c = center frequency in Hz

2.2 Need for N&V Analysis

The N&V management for marine vessel has multiple dimensions, right from the identification of vibration sources, transmission paths through the diverse vessel structure and then the coupling with the water medium as noise transmitted to the environment. Multiple stakeholders have their specific application related requirements and thus it is important to understand these varied requirements to satisfy all parties involved. Following are some of the major reasons why a rigorous N&V study is essential.

2.2.1 Human Resource Management

The human resource on-board marine vessels i.e. the passengers and the crew are significantly affected because of the excessive noise and vibration levels produced by the propulsion and auxiliary machinery. This unwanted noise causes extreme discomfort and may even lead to partial or permanent loss of hearing as well as damage to body organs. On a global scale, 16% of the disabling hearing loss in adults is attributed to occupational noise of which around 5% is due to the shipping industry. According to official statistics, an estimated $242 million is spent annually on workers' compensation for hearing loss disability. Thus, it is crucial to maintain a safe work environment to ensure proper safety for all [4].

2.2.2 Fatigue Failure

Fatigue failure is the formation and propagation of cracks due to a repetitive or cyclic load. Propulsion-induced loads and vibrations are among main causes of fatigue on ships. Higher frequent loads caused by engines and propellers result in forced vibrations with high number of load cycles, are main cause of fatigue. Fatigue failure can be minimized by vibrational analysis of machinery e.g. By a non-intrusive process, we can diagnose motor and bearing problems without the need to interrupt operation of the motor or drive system [4].

2.2.3 Stealth Requirements

In naval applications, excessive noise can cause a ship to be detected by enemies and unnecessary vibrations may pop up on radar detection systems. Acoustic mines are in place at various locations underwater and they may be triggered by unwanted noise levels. The navies throughout the world give very high priority to acoustic stealth and after World War II, massive progress has been made in this field [4].

2.2.4 Underwater Radiated Noise

A detrimental, low-frequency ambient noise radiated by maritime sub-systems because of different machines and experimental activities carried out to the surrounding aquatic environment is called URN. This anthropogenic ocean noise is the cumulative result of the human maritime activities including seismic exploration by the oil and gas industries, military and commercial use of sonar, recreational boating and shipping traffic. This noise harms the peaceful aquatic eco-system for all living beings underwater, reduces the available dissolved oxygen and creates a plethora of impactful problems [4].

2.3 Sources of N&V onboard marine vessels

Vibration in ships not only causes structural fatigue but also ruins the experience of the passengers as well as the crew due to discomfort. Thus, it is extremely important to understand what are the sources of these vibrations. The classification of vibrations is done on the basis of the components which cause these vibrations. They are as follows.

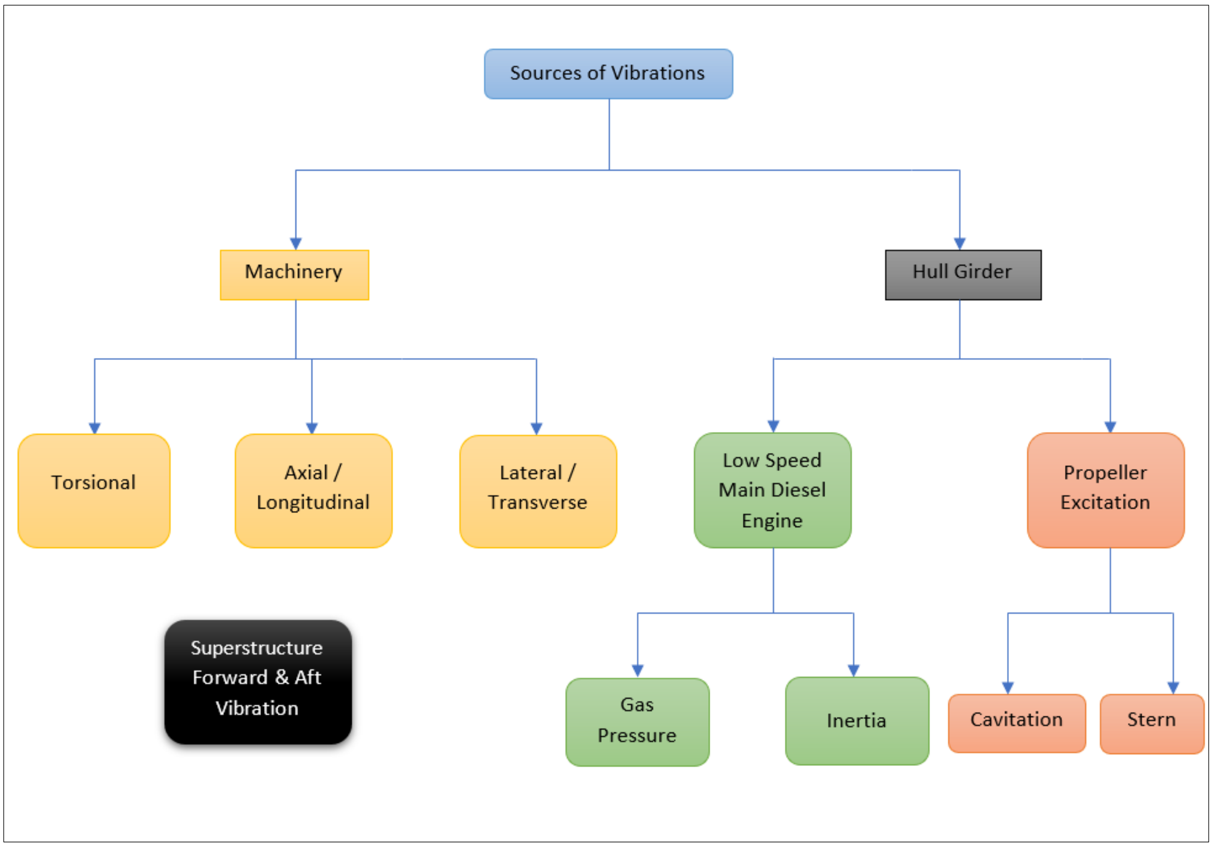

Figure 2: Different sources of Vibrations onboard ships

2.3.1 Machinery Vibrations

Engines, propulsion shafts, gearboxes, propellers, pumps, diesel generators etc. have various moving parts that can induce vibration while operating. These machinery vibrations may be further classified into the following three types.

2.3.1.1 Torsional Vibrations

Torsional vibration can be defined as the angular vibration of an object along its axis of rotation. The main propulsion system of a ship consists of the main engine, which is connected to a propeller by a shaft which is not a single component. Usually, a marine shaft consists of an intermediate shaft and a propeller shaft, which are connected by means of coupling flanges. The presence of connections, like coupling flanges, thrust block, engine connection flange, and the cylinder-piston system in the main diesel engine creates torsion in the rotating shaft system and thus it creates an excitation [3, 4, 5, 6].

2.3.1.2 Axial Vibrations

The axial or longitudinal vibrations of the propulsion system are one of the most interesting cases of machinery vibration and also possibly the likeliest to cause vibration. The velocity of the water incident to the propeller blades determines the thrust generated by the propellers also called as wake. Since the hull has a curvature at the aft, the wake on the propeller is not uniform in nature. The wake on the top propeller disc is different from that on the bottom propeller disc. This change is repeated with every revolution of the propeller. Thus, the thrust generated by the propeller is periodic. It is known as alternative thrust. This alternative thrust acts as the exciting force that results in the axial vibration of the propulsion system [3, 4, 5, 6].

2.3.1.3 Transverse Vibrations

The direction of vibration is perpendicular to the axis of the shaft’s rotation. Because of the curvature of the shafts, the ideal centreline of the shaft and its center of gravity do not coincide with each other. Hence, when the shaft rotates its shifts away from the ideal center line because of the centrifugal force on the center of gravity. This results in a vibratory motion called whirling of shafts [3, 4, 5, 6].

2.3.2 Hull Girder Vibrations

The overall effect of the excitations discussed previously is also propagated to the hull structure. Such vibrations are called Hull Girder Vibrations. The various sources of excitation for hull girdle vibration are as follows:

2.3.2.1 Diesel Engine

For a very long time, this has been known to be the number one cause of vibrations in the hull girder. Three periodic forces and periodic moments acting on the engine govern the excitation of the diesel engine. Two of these three forces remain as one of them, the force along the axis of the shaft is cancelled out by the periodic thrust. The two forces are classified as.

2.3.2.1.1 Gas Pressure Forces

This is the name given to the force exerted at various stages of one revolution of the engine. These forces are caused by the transverse reaction forces resulting from firing orders and are also dependent on the number of cylinders. When the frequency of these forces is in range of the natural frequency of the engine foundation, the latter starts resonating. This causes local vibrations in the bottom structure of the engine room. Hence, lateral stays or top bracing are employed to connect the top portion of the engine to the hull girder in order to prevent the motion. Also, to alter the stiffness of the structure it is recommended that the engine foundation is redesigned at earlier stages of the structural design [3, 4, 5, 6].

2.3.2.1.2 Inertia Forces

In a low-speed main diesel engine, the rotating parts usually have a high mass. Hence, high inertia forces are generated during the acceleration of the reciprocating engine parts. The second aspect of main engine excitation is that of Periodic Moments. In an internal combustion engine, the time is taken by the piston to travel from the Top Dead Centre (TDC) to the Bottom Dead Centre (BDC) differs from the time taken by the piston to travel from BDC to TDC. In a single complete revolution, the time taken in each half is different giving rise to the Second Order Vertical Moments. The RPM of the engine has a frequency half that of these periodic moments. The second order vertical moment in a marine diesel engine with six or more working cylinders plays an essential role in determining whether there is a possibility of hull girder vibration [3, 4, 5, 6].

2.3.2.2 Propeller

Excitation in propeller due to rotation is of two types and are as follows.

2.3.2.2.1 Cavitation

This causes bubble formation which implode on the propeller blade. Usually, the cavitation does not happen on the propeller at every point of a revolution. It occurs only at locations where the total pressure on the blade falls below the vapor pressure of seawater. These imploding bubbles generate periodic excitation force. Hence, the cavitation at various speeds must be factored into the design while making one for the propeller [3, 4, 5, 6].

2.3.2.2.2 Vertical Pressure Forces on Stern

As the propeller rotates, the stern of the ship experiences vertical pressure forces. The frequency of the pressure forces and the excitation frequency of the propeller are similar. Ships that have long overhanging sterns are more prone to this type of excitation, and the vibration is usually felt in the aft section of the vessel [3, 4, 5, 6].

2.3.3 Superstructure Vibrations

With every new model, the length of cargo ships has been increasing at a rapid rate. To accommodate this change, the engine room of most ships is shifted aft from midships in an attempt to reduce the shafting length. Since the length increases, it is necessary to strengthen the longitudinal section. To make this happen discontinuities have to be shifted away from the midship. Hence, it is necessary to shift most of the superstructures towards the aft. In order to get a proper view, the navigation deckhouse has to be at a certain height from the main deck. Usually, the engine room cavity houses the deckhouse structure. This makes it difficult to achieve sufficient stiffness of the structure. Since the propeller and the superstructure are close to one another and the top structures of the deckhouse are quite light, propeller-induced superstructure vibration has become an important aspect of ship vibrations. Superstructure vibrations are related to two types of motion namely rocking and bending. Propeller-induced forces and low-speed diesel engines are the main excitations of superstructure vibrations. During analysis, superstructure vibration and hull girder vibration are taken into account separately. Classification societies have recommended certain procedures associated with each individual classification [3, 4, 5, 6].

2.4 Source-Path-Receiver Model

The modelling of onboard noise analysis for ships and offshore units requires three key elements to be considered: “Source”, “Path”, and “Receiver”, and the procedure is described as Source-Path-Receiver modelling. “Sources” are the equipment which generates airborne noise and structure-borne noise, such as main engines, propellers, compressors, and fans. “Paths” are the air, fluid, or solid structures such as decks and bulkheads through which sound propagates. “Receivers” are the compartments of interest, such as crew cabins, workspaces, and offices. The modelling of the receiver component addresses the location and the acoustic characteristics of these spaces [7].

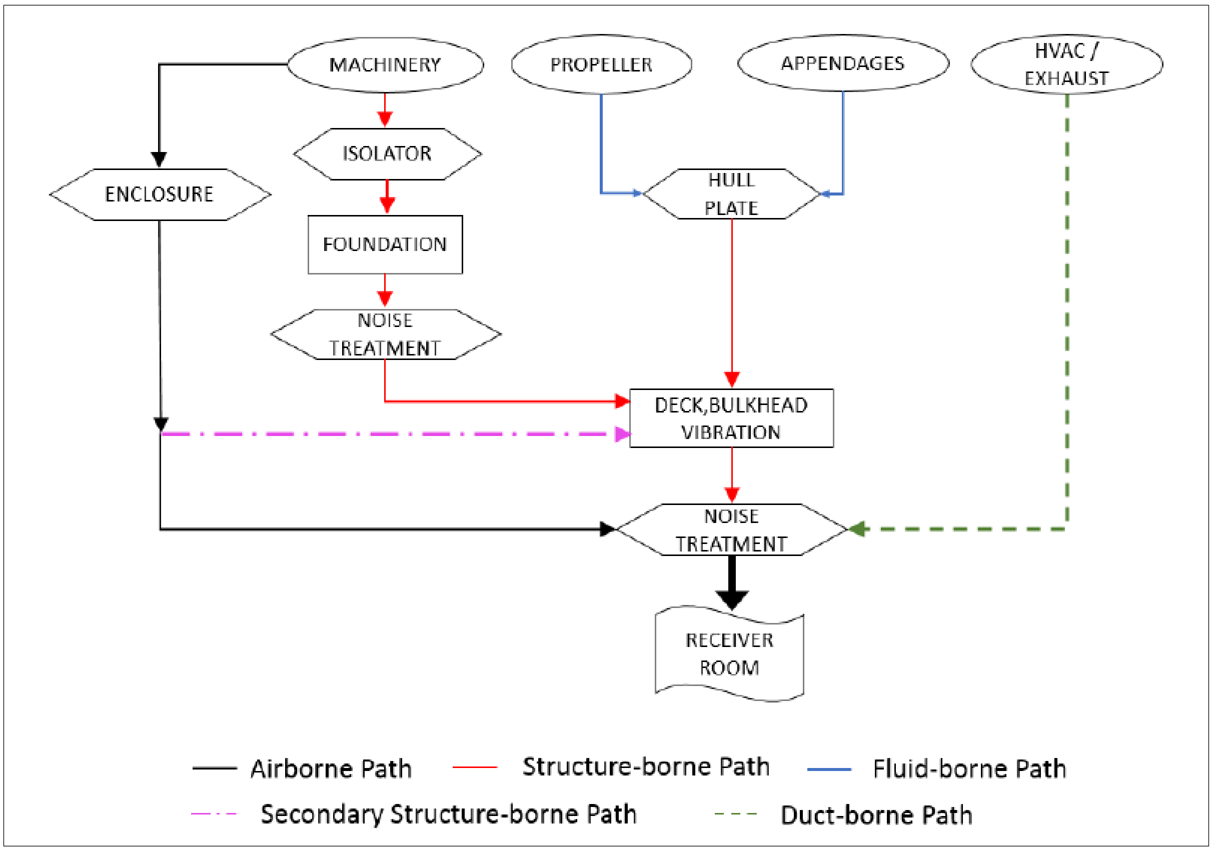

Figure 3: Source-Path-Receiver Model

2.5 Transfer Path Analysis (TPA)

Transfer path analysis (TPA) is a method that has been widely used to estimate the vibration and acoustic sources in the dynamic systems and to identify the vibro-acoustic paths of complex mechanical systems. The vibration energy from sources is transferred to system components and sub-assemblies through paths, such as individual components and physical connections. In conventional TPA, the responses at any locations are expressed as the sum of the path contributions which are obtained with each path and the excitation force. The contribution of each path in the TPA process is mainly defined by the frequency response functions (FRFs) between the excitation forces at input points and the acceleration responses at output points. Typically, it is assumed that the system is linear and time-invariant. The TPA analysis is easily performed with a simple procedure including modal testing for obtaining FRFs and mathematical operations for estimating the vibro-acoustic sources. The most important benefit of the TPA method is the identification of vibration and acoustic sources during the operation of the systems. However, in a very complex dynamic system subject to multiple sources, response signals can be strongly affected by noise, disturbances and additional forces, so it is not easy to accurately identify the pure operational force that we mainly want to know.

Transfer Path Analysis has been a valuable engineering tool for as long as noise and vibrations of products have been of interest. A TPA concerns a product's actively vibrating components (such as engines, gearing systems or turbochargers) and the transmission of these vibrations to the connected passive structures. TPA is particularly useful when the actual vibrating mechanisms are too complex to model or measure directly, as it allows us to represent a source by forces and vibrations displayed at the interfaces with the passive side [6, 10].

2.5.1 Airborne Path

Airborne noise is when sound propagates through air such as machinery casing noise. When airborne sound propagates in a free field, it gradually attenuates over the distance it propagates. However, when it meets a solid object, such as a steel plate, the attenuation increases significantly. Noise sources producing airborne sound could influence open deck areas and locations close to these openings. Due to the high attenuation of the decks and bulkheads in ships and offshore units, the airborne path is usually a critical factor only within a source space itself and the compartments directly adjacent [6, 8, 9,10].

When the airborne sound transmits from one compartment (source room) to its adjacent compartment (receiver room), the sound pressure level of the receiver can be calculated by the formula below:

$$L_{p,R} = L_{p,S} - TL + 10\log\frac{S}{A_E} \;\; dB$$

Where,
$L_{p,R}$ = sound pressure level in the receiver room in dB
$L_{p,S}$ = sound pressure level in the source room in dB
TL = transmission loss in dB
S = area of bulkhead or deck between receiver room and source room in m^2
A_E = sound absorption area of receiver room, in m^2

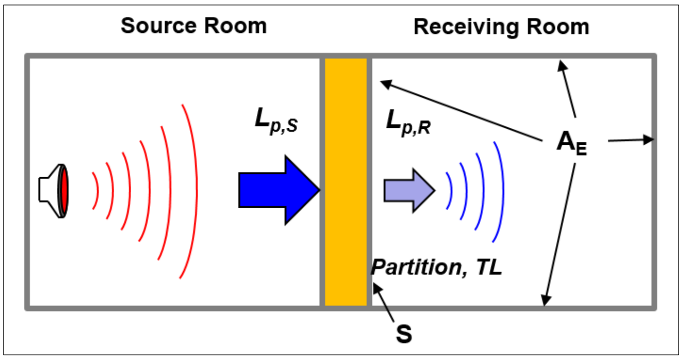

Figure 4: Transmission of sound for airborne path between a source and a receiver

2.5.2 Structure-borne Path

The elements of the structure-borne path in ships and offshore units can be divided into four groups, as listed below. The total structure-borne sound transmission loss from the source to the receiver is the arithmetic sum of all the losses throughout the transmission path [6, 8, 9,10].

i) Structures within the source room: Within the source room, the structure-borne transmission loss from an effective source area to the source room boundary mainly depends on its shape, orientation, and distance to the compartment boundary. “Effective source area” in these Guidance Notes is defined as the region of decks or side shells within which the excitation force almost has no dissipation. For machinery equipment, the effective source area is the deck region immediately below the machinery’s foundation.

ii) Structures beyond the source room: For structures beyond the source room, the structure-borne transmission loss depends on the damping loss factor of the structure, the size of the structure, and the distance from the source to the structure of interest. The larger the damping loss factor and the larger the distance from the source to the structure of interest, the higher the attenuation of the structure-borne sound.

iii) Intersections of structures: For the intersection of structures, such as the junction of two bulkheads, bulkhead and deck, or a deck and the hull, the structure-borne sound transmission loss can be significant. The transmission loss of the intersection depends primarily on three factors: the shape of the intersection, the plate thickness, and the material. Most intersections of structures are right-angles, cross junctions or T junctions. The higher number of intersections between the source and the receiver along the transmission path, the more the structure-borne sound attenuates.

iv) Pillars: Pillars usually act as a rigid coupling between decks at most frequencies of interest. The transmission loss between decks connected by a pillar is almost zero. Therefore, pillars, especially those located in the vicinity of vibration sources, cannot be ignored in the noise analysis as the acoustic energy can transmit through the pillar with almost no dissipation [6, 8, 9,10].

2.5.3 Duct-borne Path

The duct-borne path is where the heating, ventilation, and air-conditioning (HVAC) system transmits sound from air-conditioning equipment such as the air handling unit (AHU) to the duct outlets [6, 8, 9,10]. When the sound transmits via ducts, it attenuates due to the following six factors:

i) Plenum: A plenum is a pressurized housing containing air at positive pressure (pressure higher than surroundings) to equalize pressure for more even distribution. When the plenum is installed with sound absorption linings, it can significantly attenuate the sound energy. Therefore, a plenum can sometimes work as an acoustic silencer device.

ii) Silencers: A silencers is a device commonly used in HVAC systems to attenuate duct-borne noise. It consists of a large amount of sound absorptive materials which dissipate acoustic energy. The sound attenuation effect of the silencer depends primarily on its type and length. There are two types of silencers: high pressure drop silencer and low pressure drop silencer. The sound attenuation effect of the high pressure drop silencer is greater than the low pressure drop silencer. The sound attenuation effect of the silencer also increases with its length.

iii) Straight Duct Attenuation: Sound attenuation will occur even when transmitting in straight ducts. The sound attenuation effect depends on the sound absorptive effect of the inner surface of the duct and the distance it flows by.

iv) Branches where Flow Divides: When the sound transmits through a branch, it will divide into several portions. This will result in sound attenuation for each branch.

v) Turns where Flow Changes Directions by More than 30 Degrees: When sound propagates through a turn, it will attenuate by approximately 1 to over 10 dB depending on the diameter of the cross section of the turn duct and the lining configuration. Both the increase in the cross-sectional area of the duct and the installation of sound absorptive linings will increase the attenuation.

vi) End Reflections at Duct Openings: When sound transmits into a room, part of the sound energy will reflect back rather than transmit into the receiver rooms. This reflection at the duct openings will cause sound attenuation, especially for the frequency range below 250 Hz [6, 8, 9,10].

The total sound attenuation from source to outlet via ducts is the sum of the attenuation caused by the six factors

$$T_{duct} = T_{plenum} + T_{silencer} + T_{straight} + T_{branch} + T_{turn} + T_{end}$$

2.5.4 Fluid-borne Path

The hydro-excitation sources such as propeller, thruster, and wave-slap transmit the hydro-acoustic pressure to the hull by water and result in hull pressure force. The fluid load of ships and offshore units can be divided into two major groups which are, ocean water outside the structure and fluid inside the structure such as ballast water. Normally, the effect of the fluid inside the structure on the on-board noise is slight and can be ignored. Although the water will increase the damping loss factor of the structures by several times, the damping is still too small to affect the structure-borne noise. Ocean water is recommended to be considered. Although its effect on the high frequency range is slight, it may be significant in the low frequency range [6, 8, 9,10].

2.6 Analysis Models for N&V

The shipping industry has relied on empirical models to predict vibration and sound pressure throughout a vessel for many years. Empirical methods have proven useful when the ship to be studied is built of similar material, has similar general arrangement plan and has conventional sources as the numerous ships used to build the empirical models. But these empirical models have their limitations and the credibility of the data on which these models are built is under scrutiny. Thus, In the late 1960s, with the advent of powerful computers, the acoustical finite element method (FEM) became feasible. In this approach the fluid volume is divided into a number of small fluid elements with adaptive mesh sizes, and the equations of motion are solved for the elements based on fundamentals of fluid dynamics. Furthermore, some ship building companies also used FEM to predict first few global modes of the ship and making sure the different sources would not excite the structure with the same frequencies to avoid major resonance problems. Another application of FEM is in the design of the engine foundation. A local FEM model of the engine foundation can be built and the input impedance at the location of the engine and gearbox attachment points can be computed and compared with the impedance of the mounting system. This process ensures a strong impedance mismatch and therefore limiting the amount of vibrational energy getting into the structure. Finally, local FEM models can be used to diagnose local resonance problems by visualizing the mode shapes of certain panels and stiffening or damping as required [11, 12, 13, 14, 15].

The Boundary Element Method (BEM) was developed a little later than the FEM and is a branch of FEM which is mostly combined with FEM and involves the use of fluid properties only on the boundaries of the node elements. For sound propagation problems involving the radiation of sound to infinity, the BEM is more suitable because the radiation condition at infinity can be easily satisfied with the BEM, unlike with the FEM. However, the FEM is better suited than the BEM for the determination of the natural frequencies and mode shapes of cavities [11, 12, 13, 14, 15].

Finally, the latest technique which has grown rapidly over the past few years and is in demand today is the Statistical Energy Analysis (SEA) method. SEA has been established in space, aircraft, automotive and train industry for many years now, and this method is increasingly used in the marine sector. SEA is a method for predicting the transmission of sound and vibration through complex structural acoustic systems. The SEA method calculates the diffusion of acoustic and vibration energy in complex acoustic systems using energy flow relationships. It predicts the average response of the structure, which avoids a large quantity of calculations. The method is particularly well suited for quick system level response predictions at the early design stage of a product and for predicting responses at higher frequencies. In SEA, a system is represented in terms of a number of coupled subsystems and a set of linear equations are derived that describe the input, storage, transmission and dissipation of energy within each subsystem. The parameters in the SEA equations are typically obtained by making certain statistical assumptions about the local dynamic properties of each subsystem (similar to assumptions made in room acoustics and statistical mechanics). These assumptions significantly simplify the analysis and make it possible to analyze the response of systems that are often too complex to analyze using other methods (such as finite element and boundary element methods) [11, 12, 13, 14, 15].

In the SEA method, the entire structure is considered as a system, which can be divided into a number of coupled subsystems, such as plates, beams, and cavities. Each subsystem represents a group of modes with similar characteristics and a storage of energy. The SEA subsystems can be considered to be “control volumes” for vibratory or acoustic energy flow. The successful prediction of the acoustic energy of each subsystem greatly depends on an accurate estimation of three parameters: the internal loss factor, the modal density, and the coupling loss factor.

The internal loss factor is a critical parameter to predict the vibrational response of a structure by SEA and is usually obtained experimentally. It primarily incorporates three different damping losses:

i) Structural damping loss factor which is associated with energy dissipation within the structural element itself
ii) Acoustic radiation damping loss factor
iii) Damping loss factor at the structural boundaries which is associated with energy dissipation at the boundaries of the structural element

Modal density is a parameter describing the energy storage capacity of each subsystem, which depends on the average speed with which waves propagate energy through the subsystem and the overall dimensions of the subsystem. For simple subsystems, such as bars, beams, flat plates, and acoustic volumes, theoretical analysis can be used to calculate this parameter. For complex subsystems, when the subsystems to be modelled are not ideal structural elements, experimental techniques can be used to obtain this parameter.

The coupling loss factor for the link between two coupled subsystems is a measure of the rate of the energy flowing out of a subsystem through a junction to another subsystem. The larger the coupling loss factor, the greater the energy transmission. Theoretical expressions are available for couplings between several simple connections [11, 12, 13, 14, 15].

2.7 URN Management

There have been significant efforts taken so far by the maritime industry to reduce the levels of noise and vibration emissions by various types of ships. The issue was initially addressed so as to prevent structural fatigue damage to the onboard machinery and heavy-duty equipment. For the last three decades, the comfort of passengers as well as the health of the crews has been increasingly considered by all the stakeholders such as ship owners, shipyards and regulatory classification societies. However, with the technological advancements and research, there is no doubt that the increase of underwater noise related to anthropogenic activity at sea induces grave risk on marine life. The adverse effects of the use of powerful sound sources such as low frequency active sonar, air guns which are used by oil industry, pile driving for installation of offshore platform etc. are visible for all to see and have been reported. The hazards created because of underwater noise generated by commercial shipping are presently becoming more acute because of the steady increase of ship traffic and vessel size. Despite the fact that noise levels generated by shipping as compared to other sources such as active sonar, the radiated noise occurs continuously and it has been established that it impacts large maritime areas. The harassment effect on the aquatic life can cause large disturbance on the biologic functions of some marine species, and in the long term, lead to habitat loss and negative consequences on biodiversity. Excessive levels of underwater noise can be especially dangerous for sea creatures like whales and dolphins, hindering their ability to communicate, hunt, migrate and echolocate.

2.7.1 URN and its Significance

The underwater acoustic output generated by commercial ships contributes significantly to ambient noise in the ocean. Underwater noise from commercial ships is generated during normal operation, most notably from propeller cavitation which is known to peak at 50–150 Hz but can extend up to 10,000 Hz. A detrimental, low-frequency ambient noise radiated by maritime sub-systems generated because of the different machinery operating onboard marine vessels which is transmitted to the peaceful aquatic eco-system is called Underwater Radiated

Noise (URN). The major harmful impact for all marine mammals is due to the reduced available dissolved oxygen and thereby creates a plethora of impactful problems.

The Underwater Radiated Noise (URN) Management on-board marine platforms is an interesting research area with varied stakeholder interests. The first is the ship design and manufacturing for efficient operational & maintenance related aspects. The second is the acoustic stealth related naval application for enhanced deployment efficiency to avoid detection by enemy sonars and also acoustic mine avoidance. The third is the growing marine conservation related application pertaining to Acoustic Habitat Degradation. These are multi-dimensional requirements related to safety of the ship, sustainability of the shipping operations and also growth related to the shipping sector [16, 17, 18].

2.7.2 Effects of URN on Marine Eco-system

A sound becomes audible when the receiver is able to perceive it over a background noise. The audible range of hearing for marine fauna spans from as low as 5 Hz up to about 200 kHz. Marine mammals use hearing as their primary sense of perception and are highly dependent upon noise / sound for their navigation and communication. Various other fundamental activities such as finding food, reproduction and hazard detection are also based on sound perception and hence are likely to be sensitive to the increase in environmental noise. Acoustic masking occurs when the presence of one sound (unwanted noise) reduces the ability of an animal to perceive a second sound (of interest). Acoustic masking is considered to be a threat to marine fauna, especially those species that communicate on low frequencies, such as baleen whales. Therefore, an excessive high level of ambient noise in the low frequency range can have a negative impact on their population. The predominant noise levels associated with large vessels are in the frequency range of 5–1000 Hz. Noise levels at higher frequency (above 1000 Hz) will normally decrease with increasing frequency. Therefore, the predominant noise in the low-frequency band will affect the ambient noise over a large ocean area. Moreover, this low-frequency band happens to overlap with the frequency band in the audible range used by some marine mammals. Concerns about the potential impact of ocean noise on marine fauna prompted the International Maritime Organisation (IMO) to release a non-mandatory guideline for the reduction of underwater radiated noise (URN) from commercial shipping in 2014 [19, 20, 22, 23,].

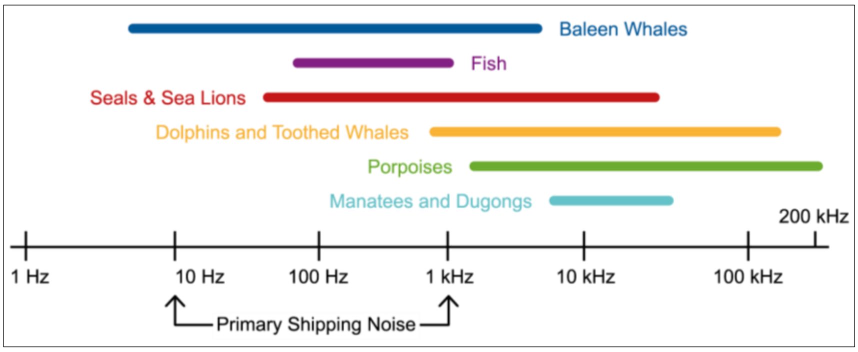

Figure 5: Frequency relationships between marine animal and shipping sounds

2.7.3 Sources of URN

There are two main groups of underwater noise sources. The first is propellers, jets and other underwater propulsion systems. Propellers constitute a major source of underwater noise because of the rotating blades operating in non-uniform flow. The propeller induced URN can occur in two ways. First is the direct radiation of noise from propeller blades due to their vibration. Second is due to the transfer of forces which create imbalanced moments from these blades to hull, which causes the vibration of the hull and ultimately causes radiation of noise.

The second is the machinery vibration caused by propulsion and auxiliary machinery. Machines which have rotating or reciprocating parts generate noise at the fundamental (natural) frequency and their multiples (harmonics). There are numerous principle and auxiliary machineries located at multiple decks inside the ship. The mechanical vibration from these machineries is radiated from the hull through their mounts and the decks in a very complex configuration. The classes of machinery can be divided into two types based on their functions: propulsion machinery and auxiliary machinery. The first contributor is the main propulsion system. Because diesel engine speed varies according to propulsion demand, the noise is generated at frequencies that depend on ship speed. Propulsion turbines, turbine generators, and reduction gears are the dominant sources of propulsion system noise on steam turbine equipped ship. Noise components from rotating auxiliary machinery and other shipboard equipment contribute to the ship overall noise signature, but usually at lower levels than propulsion systems [21, 24].

2.7.4 AQUO Project

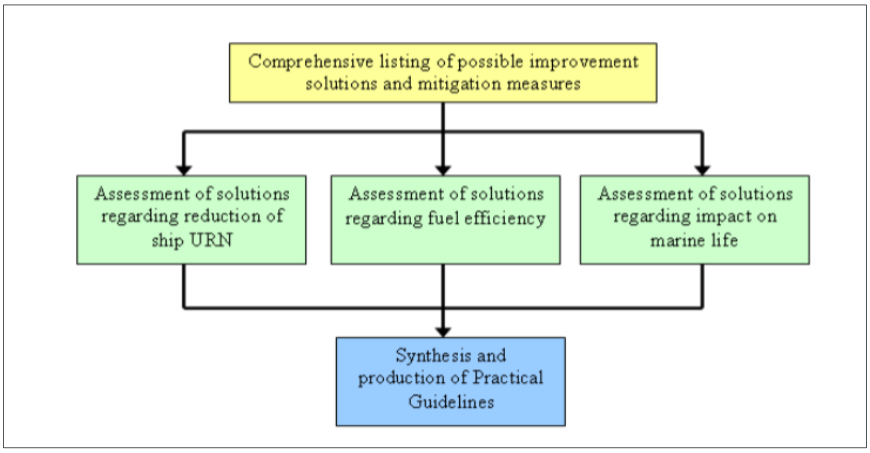

Figure 6: Methodology for noise mitigation measures assessment in the AQUO Project

Considering the impact of URN on marine eco-system, it is imperative that solutions must be found to mitigate these harmful effects of the low-frequency noise. To address this issue, the project AQUO "Achieve Quieter Oceans by shipping noise footprint reduction" (www.aquo.eu) started in October 2012. The AQUO project was built in the scope of FP7 European Research Framework. The final goal of AQUO project is to provide to policy makers practical guidelines, acceptable by shipyards and ship owners. Two types of solutions taking into account bioacoustics criteria are provided: solutions regarding ship design (including propeller and cavitation noise) and solutions related to shipping control and regulation. The overall objective of AQUO is to assess and mitigate noise impacts of the maritime transport on the marine underwater environment, mainly for the protection of marine species, to support the requirements of Directive 2008/56/EC (Marine Strategy Framework Directive MSFD) and related Commission Decision on criteria for Good Environmental Status. The which is followed is shown in the figure below.

2.7.5 URN Reduction

The underwater radiated noise analysis typically requires detailed modelling of propeller behaviour as well as the hull and machinery configurations. In recent years, active research and development have improved the accuracy of analysis tools capable of modelling the broadband noise and tonal noise. To establish a consistent metric for the underwater noise emitted from commercial ships, significant efforts have been made to develop standards for underwater noise measurement, including ISO 17208 and ANSI/ASA S12.64. Due consideration is given to the effect of site selection, environmental conditions, requirements of measurement instrumentation, test procedures, and measurement data analysis and interpretation during sea trials. Additional requirements are provided in classification society rules, and the further development of ISO standards specifically for sea trial measurements in shallow water is currently underway.

There are numerous mitigation measures for different noise sources in practice. The fundamental and widely used method is the Propeller design to reduce propeller cavitation and increase cavitation inception speed. Under the stated working conditions, the propeller creates a pressure field on the blade with a region below the vapor pressure of seawater – causing a seawater phase change. As the water vapor enters a favourable pressure gradient, the vapor pocket collapses back into a fluid creating noise. To lower propeller-radiated noise, cavitation needs to be reduced. This can be achieved by enhancing the propeller design or improving the inflow to the propeller by wake optimization. A well-regulated hull wake can enhance propulsive efficiency and reduce propeller cavitation, and propeller-radiated underwater noise. There are a variety of wake improvement devices, such as Schneekluth duct, Mewis duct, Grothues spoiler, and stern flap. It is important to ensure that the selected device is suitable for the hull shape, propeller design, and operating profile of the vessel. In addition, propeller polishing can remove marine fouling, repair erosions, and reduce surface roughness, which helps to reduce cavitation.

Emerging technologies must be implemented to test the efficiency of the said systems. One such invention is the use of acoustic coatings for noise reduction. There are two types of acoustic coatings that are efficient technological solutions: decoupling coatings and anechoic coatings. Normally, both consist of relatively thick viscoelastic layers with some voids and other inclusions in the matrix. The role of decoupling coatings is to reduce the transmission of hull vibrations to the water, and the role of anechoic coatings is to reduce acoustic reflection from the hull by absorbing incoming sound waves. The figure below is a schematic of a possible solution that includes full hull protection, including an anticorrosive primer system, a noise reduction primer and a foul release, environmentally friendly coating. The solution will potentially provide full environmental protection and superior hull protection for ship owners.

Machinery-induced underwater noise is mainly generated by structure-borne sound. The machinery vibration can first transmit to the foundations and then propagate to the hull structures, resulting in the radiation of underwater noise. Reducing this vibration and isolating the vibration source from the ship's hull are effective ways to mitigate machinery-induced underwater noise. Hence, machinery treatment to lower the machinery vibration source level and reduce the vibration energy transferred to the hull structures is one of the most critical measures to diminish the effect of URN. Some of the processes in this type of treatment include using quieter machinery equipment, installation of resilient mounts to reduce the vibrational energy transferred from the equipment into the ship's structure, using a 2-stage isolation system, acoustic enclosures to absorb engine airborne noise and active vibration cancellation by employing a secondary excitation such as a shaker to cancel the original vibration induced by machinery equipment.

Another one of the famous methods used is hull treatment. Hull treatment solutions can enhance the ship's hydrodynamic performance and therefore improve the wake flow into the propeller and reduce power requirements. Commonly used methods include hull form optimization, installation of hull and propeller appendages such as flow equalizers, and regular cleaning of the hull. Acoustic decoupling coatings and structural damping tiles can also be applied to reduce the radiation efficiency of the hull vibration [24, 25, 26].

3. POLICY FRAMEWORK

United Nation's International Maritime Organization (IMO) plays an important role in handling all international maritime affairs with support from the maritime nations of the world. As a specialized agency of the United Nations, IMO is the global standard-setting authority for the safety, security and environmental performance of international shipping. Its main role is to create a regulatory framework for the shipping industry that is fair and effective, universally adopted and universally implemented. The International Labour Organisation (ILO) is also responsible for the development of labour standards applicable to seafarers worldwide. The IMO's work in relation to noise and vibration management, began with addressing the effects of noise on humans aboard ships in the early 1980s, through the adoption of a code on noise levels on board ships by the Maritime Safety Committee (MSC) which has since been updated at regular intervals.

There are currently over 800 codes, covering all areas of navigation. These include "the Code on noise levels on board ships A.468 (XII)" adopted by the IMO in 1981. The purpose of the Code is to limit noise levels and reduce worker exposure. The Code applies to new ships of gross tonnage (GT) of 1,600 and above. The Code is not intended to be applied to passenger cabins or other passenger spaces, except insofar as such spaces are work areas, in which case they remain within the scope of the Code. MSC after reviewing the previous regulation from 1981 decided to revise it in October 2007. The revision was completed in November 2012 and its amended into effect in July 2014 [27, 28].

All nations require certain standards be met by ships and other marine structures which fly their flag. A classification society, or "Class", is a non-governmental regulatory association which regulates construction of vessels and offshore structures in the maritime industry. The society is responsible for establishing regulations for the construction and classification of ships and offshore structures. Many classification societies are in operation around the world. Of the 10 companies that are members of IACS (International Association of Classification Societies), the most important worldwide are: American Bureau of Shipping (ABS), Bureau Veritas (BV), Det Norske Veritas (DNV), Germanischer Lloyd (GL), Lloyd's Register (LR) and Registro Italiano Navale (RINA) [29]. The following Noise Standards are frequently used by Classification Societies:

- IMO Res. A. 468 (XII) - "Code on noise levels on board ships.
- ISO 2923, "Acoustics - Measurements of noise on board vessels.
- ISO 140, in particular Part 4 (Field measurements of airborne sound insulation between rooms) and Part 7 (Field measurements of impact sound insulation of floors).
- ISO 717, in particular Parts 1 (Airborne sound insulation in buildings and interior elements) and 2 (Impact sound insulation) [30].

Through its "Green Policy", the European Union has imposed increasingly stringent requirements to reduce the environmental impact of all types of transportation. This has been accompanied by the emergence of new Directives for the shipping sector as follows:

- 2001 - EN ISO 2922:2000. This standard specifies the conditions for the measurement of airborne noise emitted by vessels of all types on inland waterways and harbours, except for powered recreational craft, as these are regulated by ISO 14509.

- 2006 - Directive 2006/87/EC. The noise generated by a vessel under way shall not exceed 75 dB(A) at a lateral distance of 25 m from the ship's side and the noise generated by a stationary vessel shall not exceed 65 dB(A) at a lateral distance of 25 m from the ship's side, apart from transhipment operations.

- 2007 - EN ISO 14509-2:2007. This specifies procedures for assessing the maximum noise emitted by powered mono-hull recreational craft of up to 24 metres in length.

- 2009 - EN ISO 14509-1:2009. This standard evaluates emitted noise using calculation and measurement procedures [31].

The absence of environmental requirements regarding ship URN has been widespread in almost all contractual specifications until now, with the exception of the most modern oceanographic and fisheries research ships. The emergence of international, national and regional associations for the protection of marine mammals has led to the drawing up of a series of regulations that address underwater radiated noise and its potentially adverse effect on marine life. Noteworthy in this respect is the International Council for the Exploration of the Sea (ICES), whose Requirement 209 sets a limit to the level of lateral noise radiated underwater by the vessel at 1 m from the ship's side. By means of the ICES methodology, it has been found that the noise radiated by the ship hull at 1 metre from the hull should not exceed 132 dB. Directive 2008/56/EC comprises an international legal instrument which includes human-induced underwater noise in the definition of pollution. The Acoustical Society of America (ASA-URN) has published a Procedure for Measuring underwater radiated noise. In 2010, the DNV Classification Society issued the Silent Class Notation [14], setting different limits for each type of vessel. This notation includes procedures for measuring underwater radiated noise [30, 31].

It is important to recognize the positive impact that suitable habitability criteria and design practices may have on the safety, productivity, morale, and overall well-being of seafarers. The ABS Guide for Crew Habitability on Ships has been developed with the objective of improving the quality of crew member performance and comfort by improving working and living environments in terms of accommodation area design and ambient environmental qualities. It is the widely accepted standard based on which several organizations base their individual crew habitability regulations. These habitability criteria have been chosen to provide a means to help reduce crew fatigue, improve performance and safety, and to assist with crew recruiting and retention. The focus is on 5 habitability aspects of ship design and layout that can be controlled, measured, and assessed. These aspects are broken into 2 categories viz, accommodation areas and the ambient environment. The accommodation area criteria pertain to dimensional and outfitting aspects of spaces and open deck areas where crew members eat, sleep, recreate, and perform routine daily activities. The ambient environmental aspects of habitability pertain to the environment that the crew is exposed to during periods of work, leisure, and rest. Specifically, this Guide provides criteria, limits, and measurement methodologies for the following:

- Whole-body Vibration (separate criteria for accommodation areas and work spaces)

- Noise
- Indoor Climate
- Lighting

The criteria provided in the Guide is for the purpose of improving crew performance and providing a base level of habitability and elements of safety related to habitability [32].

4. LITERATURE REVIEW

Donald Ross published his research as “Mechanics of Underwater Noise” in 1976. This fundamental model was developed based on extensive measurement data from World War II. The basic mechanisms by which acoustic noise is generated onboard ships, transmitted by structures, and radiated into the underwater environment are explained. The classical model proposed by Ross postulates that the source spectrum for an individual ship is proportional to a baseline spectrum with the constant of proportionality determined by a power-law relationship on the ship speed and length. According to this model, propeller cavitation is the primary cause or source of radiated noise. It describes the noise source level as the function of ship parameters.

For the speed of the vessels exceeding cavitation speed, which is the minimum speed at which cavitation begins, source level is given by:

$$S(f) = S_0(f) + S'$$

Here, S_0, the base spectral level, applies to all ships moving at service speeds regardless of their type. The second term S’ is the scaling term that takes into account various ship parameters. The value of the base level can be calculated from the following expression:

$$S_0(f) = 20 - 20log_{10}(f)$$

By analysing again data from WWII vessels, Ross found an average dependency of total source level with speed as follows (note that the following formula is related to the total SL, integrated over frequency, and not to the acoustic pressure PSD):

$$SL = 170 + 53\log\left(\frac{V}{10kt}\right), \text{ in dB ref. μPa}^2 \text{ @ 1 m}$$

This formula doesn’t account for the dependence of SL on ship size. Two alternative expressions are introduced for that purpose, applying to total SL above 100 Hz:

$$SL = 112 + 50\log\left(\frac{V}{10kt}\right) + 15\log T \text{ , in dB ref. μPa}^2 \text{ @ 1 m}$$

$$SL = 134 + 60\log\left(\frac{V}{10kt}\right) + 9\log T \text{ , in dB ref. μPa}^2 \text{ @ 1 m}$$

Regarding cavitating propeller noise, Ross has noted that the dependence on the displacement does not enter the equation for propeller cavitation noise. Therefore, he suggests using an expression for propeller cavitation noise based on the propeller tip speed UT (valid for speeds between 15 and 50m/s) and total number of blades B:

$$SL = 175 + 60\log\left(\frac{U_T}{25m/s}\right) + 10\log\left(\frac{B}{4}\right), \text{ in dB ref. μPa}^2 \text{ @ 1 m}$$

The above formulae don't introduce dependence with frequency. Ross noted that the slope of the PSD is about -5.5 to -6 dB per octave at the higher speeds and as much as -7 to -8 dB per octave at low speeds. Alternatively, a widely used model for merchant ships is expressed in terms of ship speed V and ship length L:

$$SL(f) = 190.5 + 50\log\left(\frac{V}{10kt}\right) + 20\log\left(\frac{L}{150m}\right) - 20\log f \text{, in dB ref. } \mu Pa^2/Hz \text{ @ 1 m}$$ [33]

Urick published his paper in 1983 that indicated the sources of noise on ships, submarines, and torpedoes could be grouped into the three major categories: machinery noise, propeller noise, and hydrodynamic noise. It concentrates on the practical aspects of underwater sound, and offers useful help for sonar problem solving. The research encapsulates the fundamentals and phenomena of underwater sound as applied to the Sonar Equation, the heart of prediction of sonar performance, and the contents lie squarely in the middle between theory at one end and practical technology at the other. Its coverage is broad-ranging from the basic concepts of sound in the sea to making performance predictions in such applications as depth sounding, fish finding, and submarine detection. Urick gives some typical radiated source levels that were measured on various classes of ships current during World War II. It is based on 157 measurements from 77 ships belonging to 11 different categories (mostly freighters, tankers and large warships). The underwater noise for these ships is dominated by propeller cavitation noise. The source levels were summarized in an empirical formula in terms of the propeller tip speed U_T (in feet/s), the displacement T and the frequency f in Hz.

$$SL(f) = 46.5 + 51\log U_T + 15\log T - 20\log f \text{, in dB ref. } \mu Pa^2/Hz \text{ @ 1 m}$$

This formula was found to fit individual measurements to a standard deviation of 5.4 dB. In principle, it is only applicable at frequencies above 1 kHz where propeller cavitation is the principal source of noise. A more convenient formula for the acoustic pressure PSD, for use when the information concerning the propeller tip speed is lacking, was found to be:

$$SL(f) = 95 + 60\log V + 9\log T - 20\log f \text{, in dB ref. } \mu Pa^2/Hz \text{ @ 1 m}$$

In the latter formula, V is ship speed expressed in knots [34].

The RANDI model published in 1994 stands for "Research Ambient Noise Directionality" and can be used to predict ambient acoustical noise levels and directionalities at low to mid-level frequencies for shallow and deep-water environments. This model is a modified form of the Ross Model. Ambient noise due to shipping, wind, flow noise, and system noise are considered. Shipping noise can be calculated for highly variable environments and is done using either a finite element or split-step parabolic equation. Local wind noise is computed based on the range-xindependent theory of Kuperman-Ingenito including both discrete (normal modes) and continuous spectra. Navy-standard and historical databases are used to describe the environment. The noise model particulars, inputs, and outputs are described and illustrated for an example model run. It was developed for use in the design of noise measurement experiments and the analysis of surveillance systems. According to this model the noise level comprises of the following three terms:

$$S_{RA}(f) = S_0(f) + S'(V, L) + S''(f, L)$$

Here S_0 is the RANDI base spectrum, and S' is the scaling based on ship parameters. Note that S_0 and S' in this section are different from the base spectrum and scaling in other sections. The third term, S'', is a relatively small correction term which is only non-zero for low frequencies (approximately below 200 Hz). The base spectrum is given by:

$$S_0(f) = \begin{cases} -10\log_{10}\left(10^{F_1(f)} + 10^{F_2(f)}\right) & \text{for } f < 500\text{Hz} \\ F_3(f) & \text{for } f \geq 500\text{Hz} \end{cases}$$

Here, $F_i(f) = \alpha_i + \beta_i \log_{10} f$, and coefficients are given below:

The scaling function of the spectrum is given by,

$$S'(V, L) = 60\log_{10}\left(\frac{V}{V_{ref}}\right) + 20\log_{10}\left(\frac{L}{L_{ref}}\right) + 3,$$

where the reference speed is, V_{ref} = 12 knots, and the reference ship length is, L_{ref} = 300 feet. Finally, the third component of the RANDI model, the low-frequency correction term, is given by,

$$S''(f, L) = \gamma(f)\frac{L^{1.15}}{3643},$$

where ϒ is a continuous low-frequency weighting function given by:

$$\gamma(f) = \begin{cases} 8.1 & \text{for } f < f_{c1} \\ 22.3 - 9.77\log_{10} f & \text{for } f_{c1} \leq f < f_{c2} \\ 0 & \text{for } f \geq f_{c2} \end{cases}$$

The frequency interval limits are, f_{c1} = 28.4 Hz, and f_{c2} = 191.6 Hz, respectively [35].

Wales and Heitmeyer proposed a model, which instead of taking ship parameters as input uses extensive noise data of ships and seeks to reduce root mean square (rms) error of ships with respect to the mean spectrum proposed in the paper. The model proposed represents the individual ship spectra by a modified rational spectrum where the poles and zeros are restricted to the real axis and the exponents of the frequency terms are not restricted to integer values. An evaluation of this model on the source spectra ensemble indicates that the rms errors are significantly less than those obtained with any model where the frequency dependence is represented by a single baseline spectrum. They proposed a mean spectrum:

$$\overline{S}(f) = 230.0 - 10\log(f^{3.594}) + 10\log\left(\left(1+\left(\frac{f}{340}\right)^2\right)^{0.917}\right);$$

They identified two partitions of 30-1200 Hz frequency band. For frequencies greater than 400Hz, individual spectra showed simple power dependence and for less than 400 Hz, many of the source spectra showed complex frequency dependence. The partition into a low frequency band (30-400Hz) and high frequency band (400-1200Hz) showed a way towards developing separate model parameters. Keeping these points in mind, a rational spectrum model is proposed in the paper. The rational spectrum model provides a ship dependent spectrum that consists of linear combination of approximating functions. This model gives a better estimation of both the individual spectra and the variability than the Ross Model, as claimed by Wales and Heitmeyer in their original paper [36].

Wittekind Model breaks down ship radiated noise into three components namely:
(a) Low frequencies from propeller cavitation.
(b) Medium to high frequencies from propeller cavitation.
(c) Medium frequencies from four-stroke diesel engines.

Hence, the following parameters of the ship are considered in the model:
(a) Displacement.
(b) Speed relative to cavitation inception speed.
(c) Block coefficient as an indicator for wake field variations.
(d) Mass of diesel engine(s).
(e) Diesel engine resiliently mounted yes or no.

Source levels in the Wittekind model are given in terms of 1/3 octave band levels instead of source spectrum levels as follows:

$$SL = 10log_{10}\left(10^{\frac{SL1\ (f_k)}{10}} + 10^{\frac{SL2(f_k)}{10}} + 10^{\frac{SL3(f_k)}{10}}\right)$$

The first contribution $SL_1(f_k)$, represents the low-frequency cavitation noise, $SL_2(f_k)$ represents the high-frequency cavitation noise and $SL_3(f_k)$the diesel engine noise [37].

5. DUCT-BORNE TPA

The noise due to HVAC systems has a significant impact on living and working spaces inside ships. This is a very tangled problem since there is a lack of regulation regarding ships noise or vibration generated by a specific equipment. Indeed, regulations are applied to the overall levels in the accommodation spaces and in some working areas, as well as for air-borne noise radiated from exhaust and ventilation systems. These levels should be measured once the ship construction is almost finished and close to delivery, during sea trials or during the operations of the ship. If from these tests, results indicate that the levels are not satisfactory, then the compliance with the regulations becomes very difficult and expensive to obtain. The need of an effective preliminary design of the HVAC systems that can estimate their noise impact is therefore definitely advisable and recommended. Computer simulations of sound propagation inside air ducts can help the choice of the best acoustical properties of components and their geometrical design, giving great advantages over experiments in terms of reducing costs and obtaining accurate results in a faster way. When sound propagates through a duct system it encounters various elements that provide sound attenuation. These are lumped into general categories, including straight ducts, turns, plenums, branches, silencers, end effects, and so forth. Other elements such as tuned stubs and Helmholtz resonators can also produce losses; however, they rarely are encountered in practice. Each of these elements attenuates sound by a quantifiable amount, through mechanisms that are relatively well understood and lead to a predictable result. This project includes the simulation of such duct components in the ANSYS software to develop a harmonic acoustic model.

5.1 Meshing

ANSYS is a finite element analysis (FEA) is a mathematical representation of a physical system comprising a part/assembly, material properties and boundary conditions. A general technique like FEA is a convenient method to represent complex behaviours by accurately capturing physical phenomena using partial differential equations. Meshing is one of the most important steps in performing an accurate simulation using FEA. A mesh is made up of elements which contain nodes (coordinate locations in space that can vary by element type) that represent the shape of the geometry. An FEA solver cannot easily work with irregular shapes, but it is much happier with common shapes like cubes. Meshing is the process of turning irregular shapes into more recognizable volumes called “elements.” There are two main types of meshing methods. For these purposes, we are referring to 3D models: Tetrahedral element meshing or “tet” and Hexahedral element meshing or “hex”. Hex or “brick” elements generally result in more accurate results at lower element counts than tet elements. If it is a complex geometry, tet elements may be the best choice. These default or automatic meshing methods may be enough to get you where you need to go, however, there are additional methods that can give you more mesh control.

We can use a Multizone method, which is a hybrid of hex and tet elements that allows us to mesh different parts of the geometry with different methods. This enables less geometry preparation and allows more local control meshes. With sweep meshing, the mesh actually “sweeps” the mesh through the volume and faces to help create an efficient mesh with regular sizing. Deciding which mesh method to use usually depends on what type of analysis (explicit or implicit) or physics you are solving for and the level of accuracy you want to achieve. A few other options are cartesian meshing and layered tets that are used for specific analyses like additive manufacturing. Meshing controls enables a more precise mesh. ANSYS enables us to control local meshes, instead of a global mesh that meshes the entire CAD with the same method. Some examples of local meshing controls include local sizing, refinement and sphere of influence defeaturing of the geometry. Let's take a motorcycle frame for example. We apply a general mesh approach across the entire geometry but a different strategy is preferred where the weld and bolted connections are. Using local meshing controls allows us to create a more refined mesh at these locations and not mesh the entire part with smaller elements, which would take longer to solve.

A good quality mesh equals more precise results. A poor mesh can result in convergence difficulties, which can lead to incorrect results and false conclusions. The quality of your mesh depends on a few scenarios like:

- What type of analysis we are conducting?
- How much time we want to invest in the mesh?
- How much time we want to invest in solving?

In some instances, we look for a quick solution — something that will help clarify a design decision. In this instance, we do not want to spend a lot of time setting up the mesh. Other times, we want a very precise solution or result, which requires some time and effort into setting up the mesh with different methods and controls. A good mesh has a quality criterion that fits the needs (analysis type, level of accuracy, time) such as element quality and aspect ratio. Ultimately, it is recommended that understanding the geometry and using controls to get the best mesh possible will lead to a better product design and analysis [38].

In the simulation carried out for this project, a mix of tet and hex meshing along with auto configurations has been used for a rectangular duct channel having a cross section of 50mm × 50mm. Different factors such as geometry, shape and sizing, frequency and wavelength along with limitation on using the student version of ANSYS which has certain product limitations have resulted in using such kind of meshing. It can be improved with certain upgrades, however, under the given conditions the effort to achieve the best possible use of available meshing techniques is attempted.

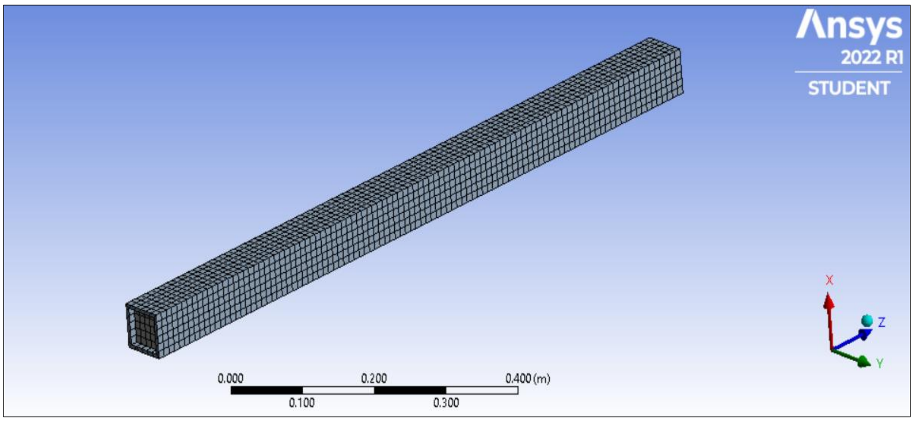

Figure 7: Meshing for straight duct

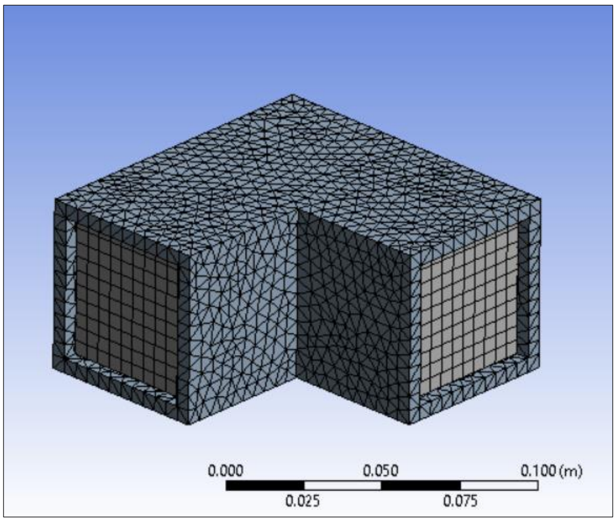

Figure 8: Meshing for turn of a duct

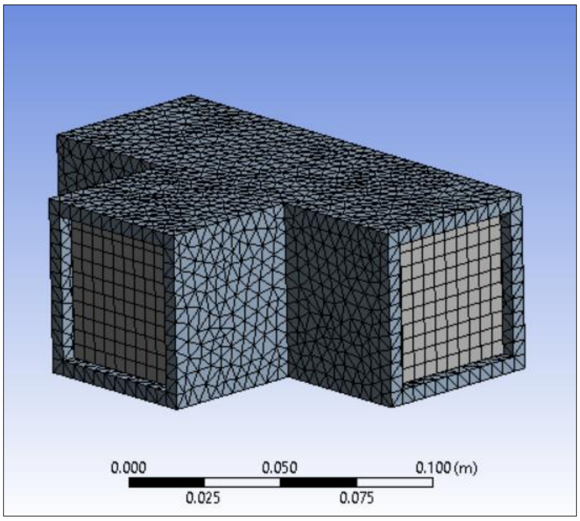

Figure 9: Meshing for branch of a duct

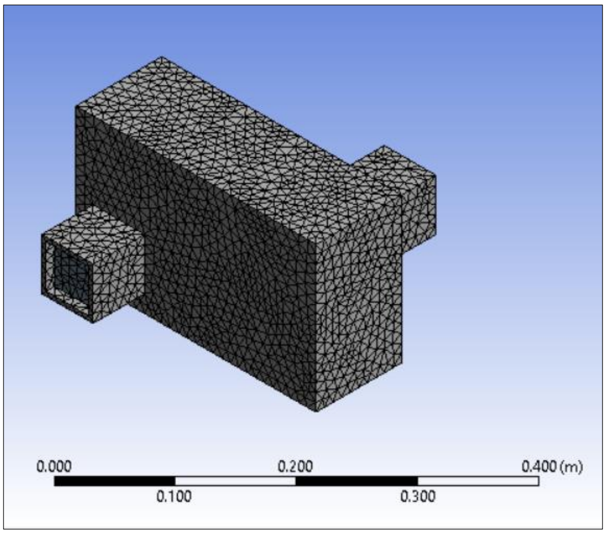

Figure 10: Meshing for duct plenum

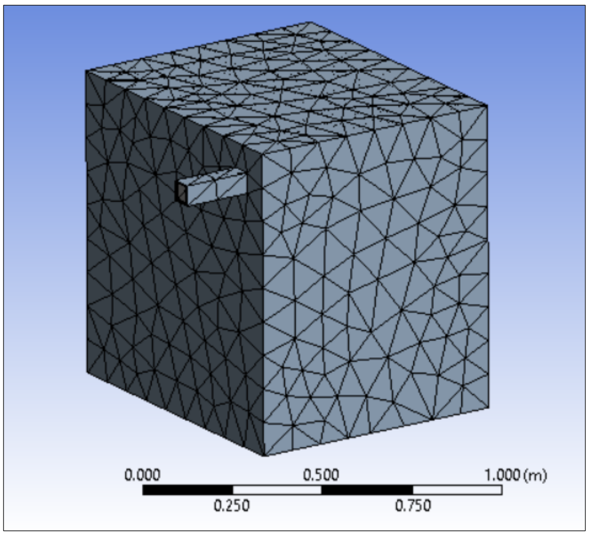

Figure 11: Meshing for opening of a duct into a room for end reflection loss

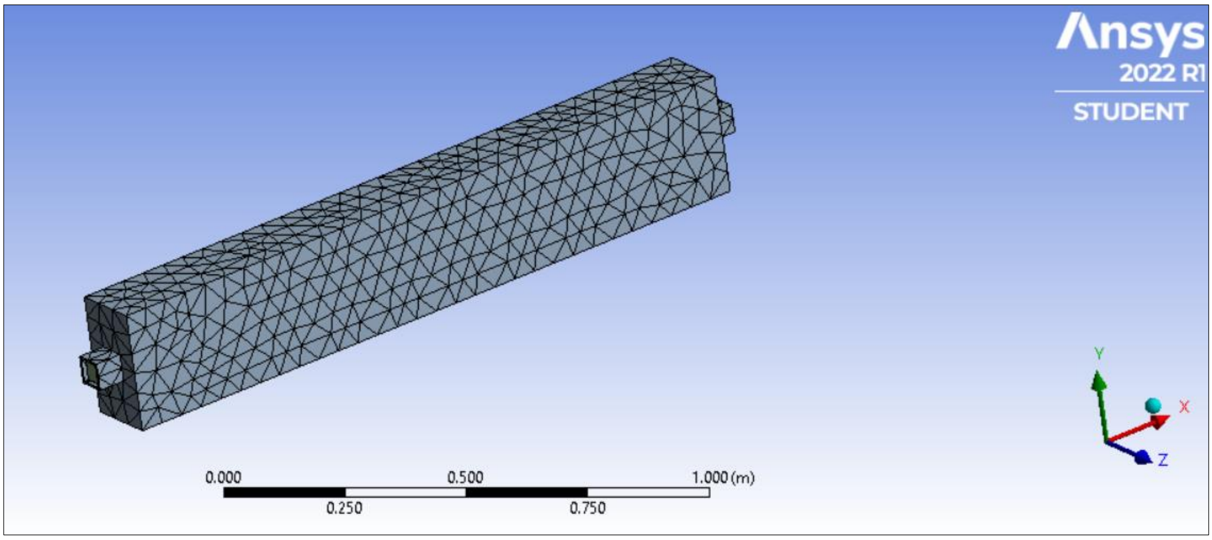

Figure 12: Meshing of a silencer

5.2 Boundary Conditions

The way that a FEA model is constrained can significantly affect the results and requires special consideration. Over or under constrained models can give an output that is so inaccurate that it is worthless to the engineer. In an ideal world we could have massive assemblies of components all connected to each other with contact elements but this is beyond the budget and resource of most people. We can however, use the computing hardware we have available to its full potential and this means understanding how to apply realistic boundary conditions.

Boundary conditions define the inputs of the simulation model. Some conditions, like velocity and volumetric flow rate, define how a fluid enters or leaves the model. Other conditions, like film coefficient and heat flux, define the interchange of energy between the model and its surroundings. Boundary conditions connect the simulation model with its surroundings. Thus, these boundary conditions define the applicability of numerical methods and the resultant quality of computations can critically be decided on how those are numerically treated. Without them, the simulation is not defined, and in most cases cannot proceed. Most boundary conditions can be defined as either steady-state or transient. Steady-state boundary conditions persist throughout the simulation. Transient boundary conditions vary with time, and are often used to simulate an event or a cyclical phenomenon. Initial conditions are a different type of condition that are active only at the beginning of the simulation [39, 40]. The following boundary conditions are used in the simulation:

i) **Analysis Settings:** A harmonic acoustic simulation is based on obtaining a required parameter at a certain frequency or a set of frequencies. It may be broadband or narrowband type of measurement. In this type of boundary condition, this set of frequency is specified. Throughout this project, 1/3rd Octave band frequencies are used and thus applied in the simulation.

ii) **Acoustics Region:** The acoustic region is the one where all the different acoustic parameters such as mass sources, impedances, surfaces and other specific conditions can be applied. In this simulation, components such as air channel inside the duct, absorptive surfaces, surface velocities and perforated plates constitute the acoustics region.

iii) **Physics Region:** The physics region on the other hand defines the structural components of the simulation. The duct outer channels which are made of steel or similar materials come under this type of boundary condition. Acoustic conditions cannot be applied to faces / bodies under the physics region.

iv) **Mass Source:** A Mass Source excitation is used to create a sound wave. This is the most basic type of creating a random wave emitted equally in all directions. Specifying the magnitude based on previously used research models allows us to replicate the vibrations on ships at the input of the duct.

v) **Radiation Boundary:** To avoid waves to be reflected at the inlet and the outlet, an acoustic radiation boundary is defined at the ends of the duct channel imitating the air flow region. This condition is called the Robin condition.

vi) **Absorption Surface:** Sound is absorbed on the inner walls of the duct and there is a reduction in sound level as the wave travels inside the duct. Absorption surface is used to simulate this attenuation inside the ducts by specifying the absorption coefficient of the material which absorbs the sound. This condition is applied to specific duct components only as it is not the cause of attenuation for all the duct components.

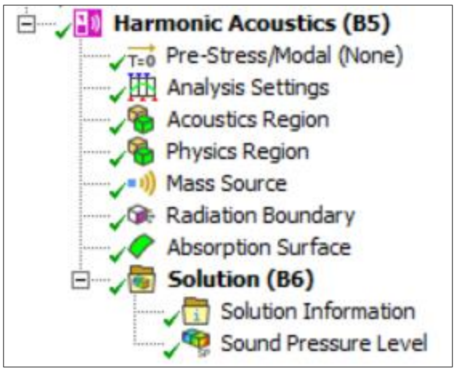

Figure 13: Boundary Conditions applied for the harmonic acoustic model

5.3 Straight Duct

As a sound wave propagates down an unlined duct, its energy is reduced through induced motion of the duct walls. The sound attenuation through a rectangular duct depends mainly on dimensions of the cross section of the duct and on the frequency content of the noise source. 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, particularly in their first mode of vibration, called the breathing mode, and therefore are much more difficult to excite. As a consequence, sound is attenuated in unlined rectangular ducts to a much greater degree than in circular ducts. Since a calculation of the attenuation from the impedance of the liner is complicated, measured values or values calculated from simple empirical relationships are used. Empirical equations for the attenuation can be written in terms of a duct perimeter to area (P/A) ratio. A large P/A ratio means that the duct is wide in one dimension and narrow in the other, which implies relatively flexible side walls. Empirical formulations show that the attenuation decreases with frequency until the 250 Hz octave band as shown in the first equation below and then remains constant as represented by the second equation [41, 42].

$$T_{straight} = 75.1 \cdot \left(\frac{P}{A}\right)^{-0.25} \cdot f^{-0.85} \cdot L$$

$$T_{straight} = 0.025 \cdot \left(\frac{P}{A}\right)^{0.8} \cdot L$$

Where,

$T_{straight}$ is the attenuation in dB, P is the perimeter in m, A is the cross section in m^2 and L the length of duct in m.

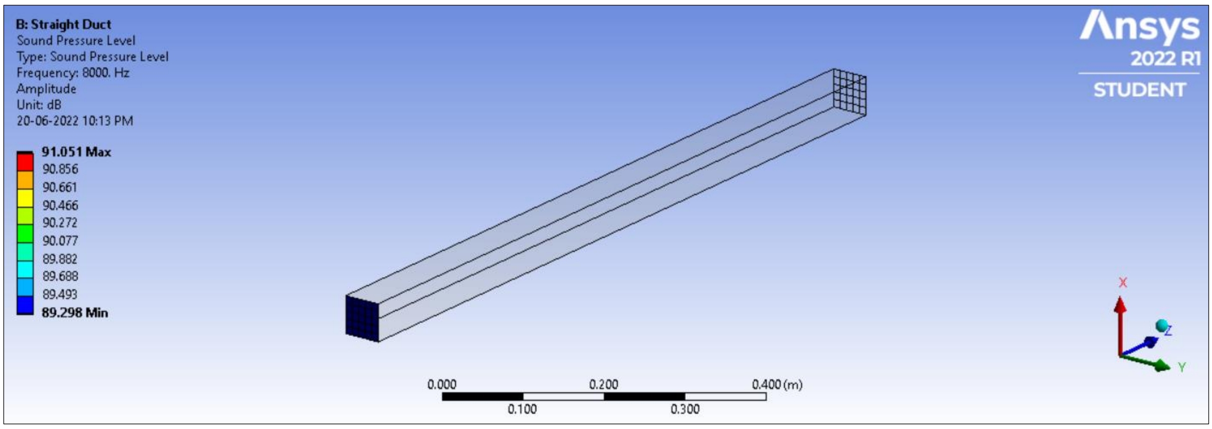

Figure 14: Straight duct simulation

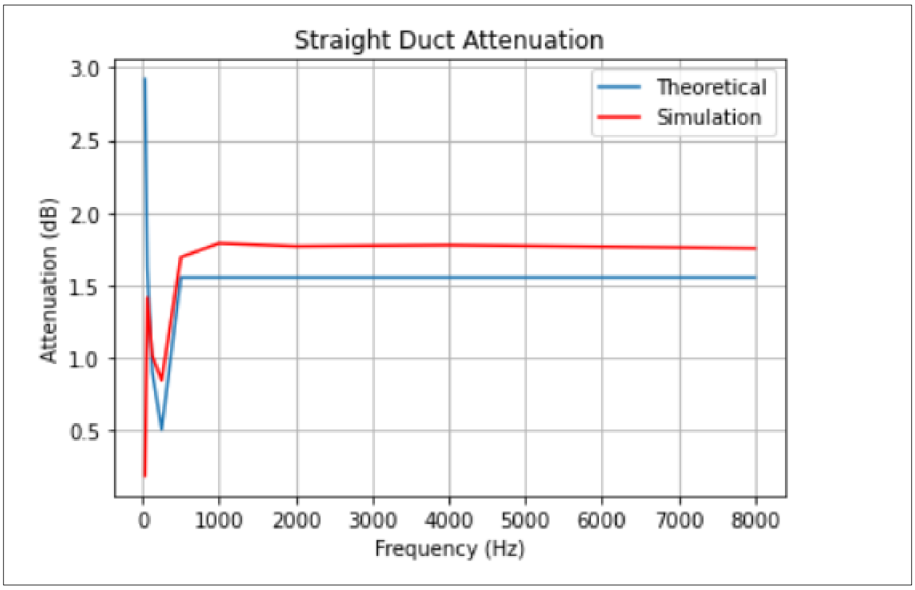

Figure 15: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for a straight duct

We can conclude from the graph above that the outputs in terms of sound pressure level at the inlet and outlet of the duct are in agreement with each other. The noise level varies erratically for frequencies below 500 Hz and afterwards remains constant as indicated by the empirical relations mentioned previously. The only glitch appears to be at the lowest octave band centre frequency of 31.5 Hz. This is because generally, low frequency model for such kind of simulations needs to be dealt specifically. This acoustic model enables us to account for the interaction between an acoustic pressure wave in a viscous fluid and a rigid wall for specific structures according to Low Reduced Frequency (LRF) approximation. But because of software licensing limitations, it is not possible in this case.

5.4 Turns

A sharp bend or elbow can provide significant high-frequency attenuation, particularly if it is lined from the inside which creates damping. When sound propagates through a turn, it will attenuate by approximately 1 to over 10 dB depending on the diameter of the cross section of the turn duct and the lining configuration. Both the increase in the cross-sectional area of the duct and the installation of sound absorptive linings will increase the attenuation [7].

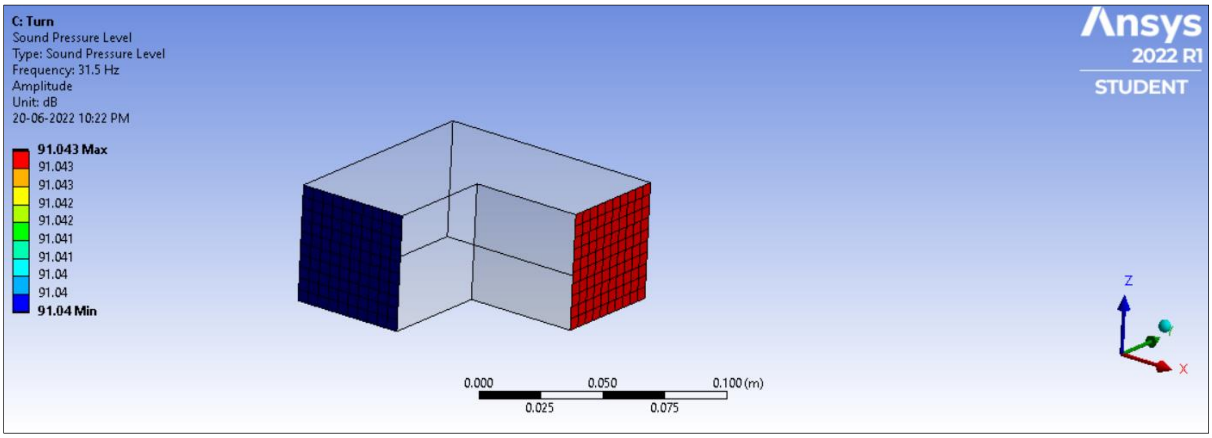

Figure 16: Simulation for turn of a duct

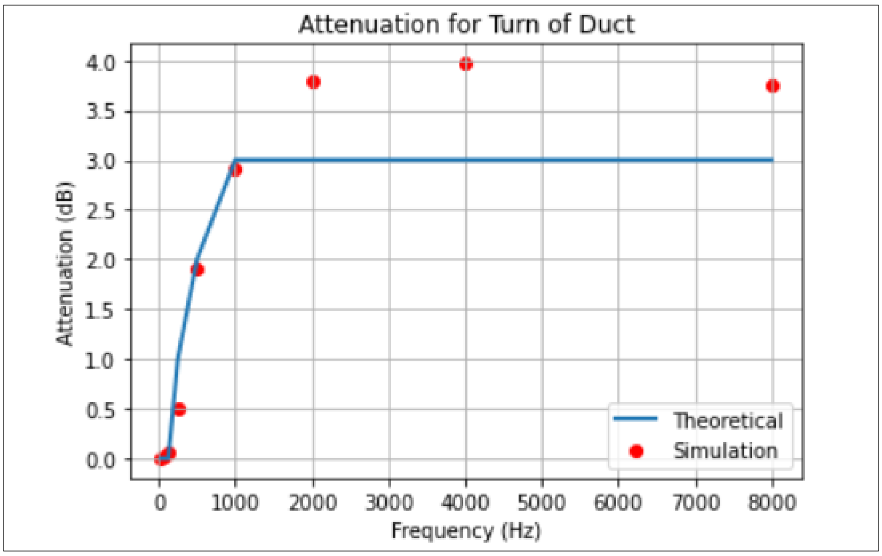

Figure 17: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for a turn of a duct

From the above graph, it is evident that the theoretical and simulation sound attenuation values match each other for frequencies upto 1000 Hz. Above this threshold frequency, we interpret the simulation as a high frequency model. The empirical values which form the basis of the theoretical inputs for the graph are established for a higher duct effective diameter than the one considered for simulation purposes. This approximation generates the relatively small error. This approximation is because the simulation software has restrictions related to meshing. A higher diameter duct has a very coarse meshing and gives erratic results for all frequencies while a smaller diameter duct with a fine meshing gives the result seen above.

5.5 Branches

When there is a division of the duct into several smaller ducts there is a distribution of the sound energy among the various available paths [7, 41]. The sound attenuation caused by the division of the sound energy through a branch (T_{branch}) can be calculated by the formula below.

$$T_{branch} = 10 \log\left(\frac{A_{BT}}{A_p}\right) \ dB$$

Where,

A_{BT} = total cross-sectional area of all ducts leaving the branch point in m^2

A_p = cross-sectional area of the duct associated with the transmission path of interest in m^2

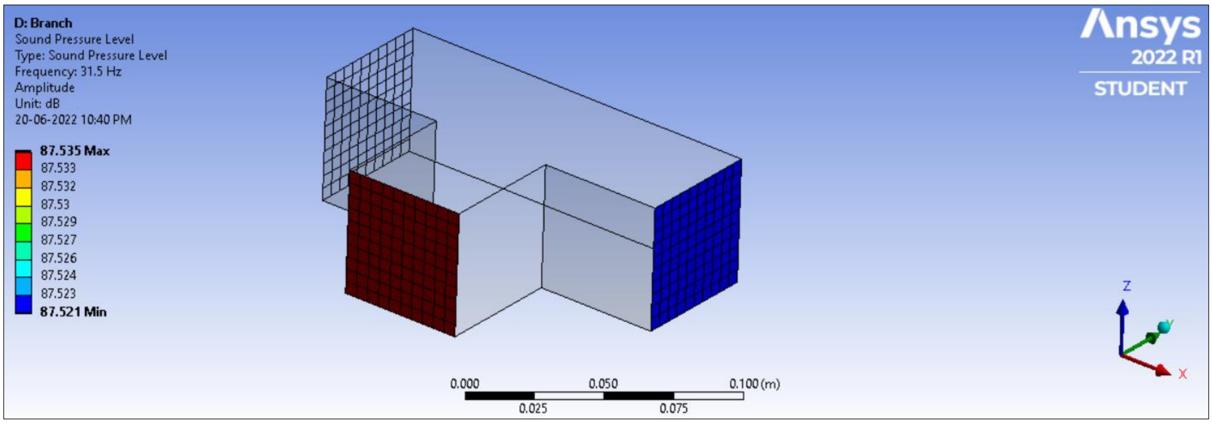

Figure 18: Simulation for branching of duct

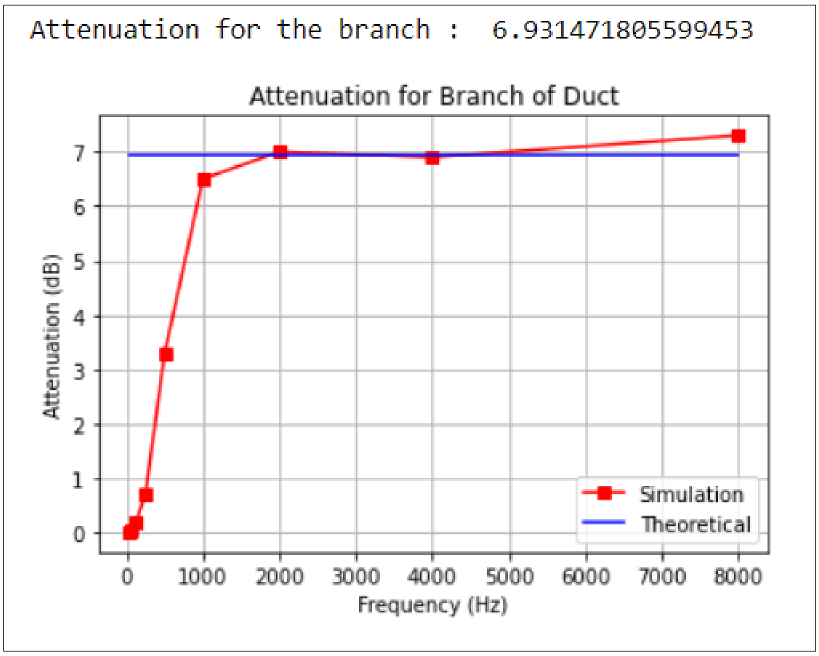

Figure 19: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for branching of duct

The theoretical relation given is defined for a high frequency model which is for frequencies above 1000 Hz. Thus, we can see from the above graph that for these range of frequencies, we get a good agreement in the sound attenuation output of the simulation and the theoretical model. For a low frequency model, the attenuation is negligible and thus these values are generally neglected for simulation purposes.

5.6 Plenum

A plenum is an enclosed space that has a well-defined entrance and exit, which is part of the air path, and that includes an increase and then a decrease in cross-sectional area. A return-air plenum located above a ceiling may or may not be an acoustical plenum. If it is bounded by a drywall or plaster ceiling, it can be modelled as an acoustic plenum; however, if the ceiling is constructed of acoustical tile, it is usually not. Rooms that form part of the air passageway are modelled as plenums. For example, a mechanical equipment room can be a plenum when the return air circulates through it. In this case the intake air opening on the fan is the plenum entrance.

Plenum attenuation depends on the relationship between the size of the cavity and the wavelength of the sound passing through it. When the wavelength is large compared with the cross-sectional dimension—that is, below the duct cut-off frequency—a plenum is modelled as a muffler, using plane wave analysis and simulated as a low frequency model. When the wavelength is not large compared with the dimensions of the central cross section, the plane wave model is no longer appropriate, since the plenum behaves more like a room than a duct. Under these conditions we use the methodology developed for the behaviour of sound in rooms.

To decide between the 2 models, we will calculate the wavelengths (λ) of sounds at the frequency (f) extremes of the octave band: 31.5 and 8,000 Hz at 30°C (303 K) for air.

- The speed of air (v_w) at 30°C for air is calculated as

$$v_w = 331 \times \sqrt{\frac{T}{273}} = 331 \times \sqrt{\frac{303}{273}} = 348.7\ m/s$$

- The maximum wavelength (λ_{max}) is

$$\lambda_{max} = \frac{v_w}{f} = \frac{348.7}{31.5} = 11\ m$$

- The minimum wavelength () is

$$\lambda_{max} = \frac{v_w}{f} = \frac{348.7}{8{,}000} = 43\ mm$$

We can see that the wavelength is comparable to the dimensions of the duct and hence we have to adopt a high frequency model and treat the plenum as a room where the attenuation takes place because of absorption of sound waves by inner surfaces of plenum. Thus, only frequencies above 1000 Hz are considered for simulation model [41, 43].

The empirical relation to calculate sound attenuation of a plenum (T_{plenum}) is,

$$T_{plenum} = 10\ \log R - 10\ \log A_0 - 16 \quad dB$$

Where,

R = room constant for the plenum in m^2
A_0 = cross-section area of the outlet duct from the plenum in m^2

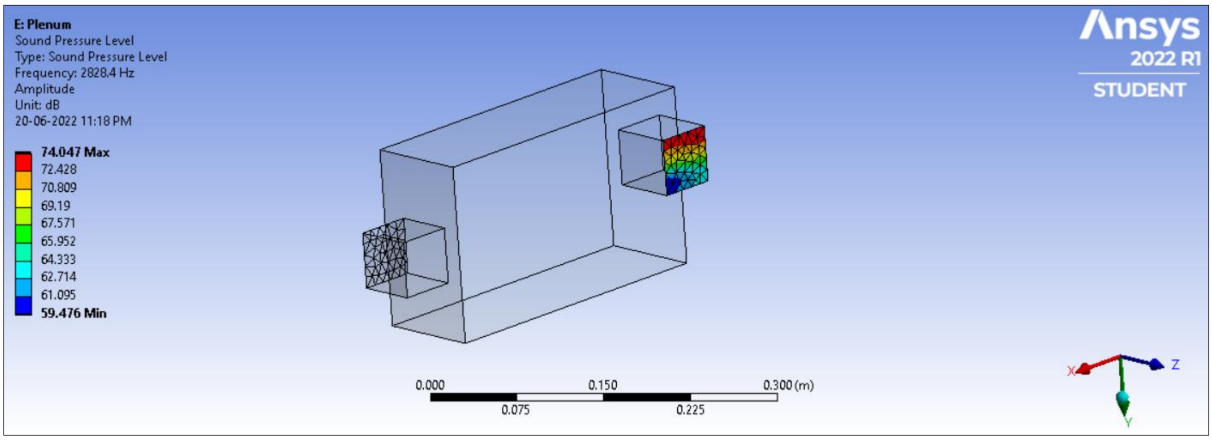

Figure 20: HVAC plenum simulation

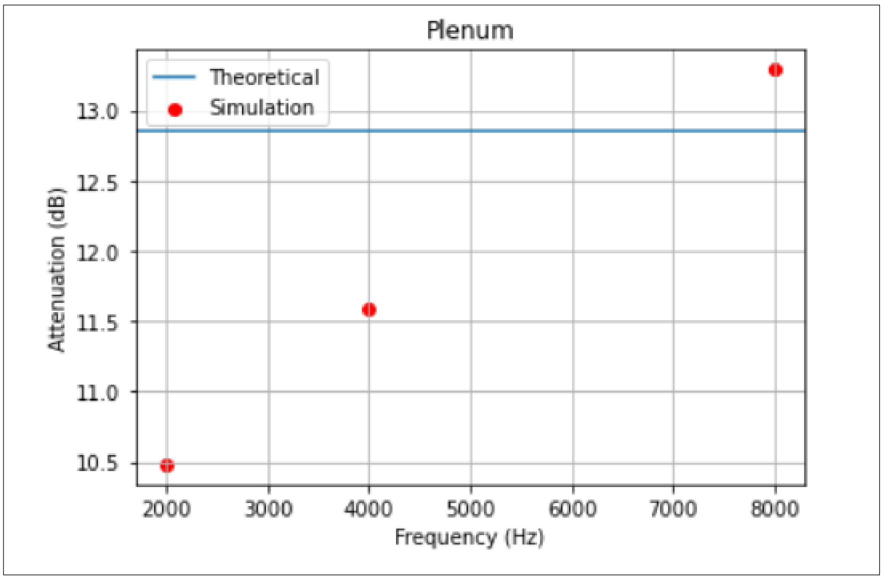

Figure 21: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for a plenum

5.7 End Reflection at Duct Openings

When a sound wave propagates down a duct and encounters a large area expansion, such as that provided by a room, there is a loss due to the sudden and enormous area change known as the end effect. The end effect does not always follow the simple relationship. At low frequencies sound waves expand to the boundaries of the duct. At very high frequencies the sound entering a room from a duct tends to radiate like a piston in a baffle and forms a beam. Therefore, it does not interact with the sides of the duct and is relatively unaffected by the end effect. The phenomena called End Reflection Loss (ERL) may be described as the apparent change in sound pressure observed at a duct termination, as derived by the difference between the incident and reflected sound pressure at the duct termination or boundary. ERL occurs in the plane wave region, defined at frequencies below the cut-off (f_c) for ducts or openings:

$$f_c = \frac{c}{2D_e}$$

Where,

c = speed of sound in air in m/s
D_e = equivalent duct diameter in m

The effective diameter for a rectangular duct is calculated as,

$$D_e = \sqrt{\frac{4A}{\pi}}$$

Where,

A = cross section area of the duct

In this case, the cut-off frequency is calculated to be 425 Hz and hence in the octave band, frequencies only upto 250 Hz are considered.

The empirical formula for calculating end effect depends on the size of the duct, measured in wavelengths [7, 41, 44, 45]. This is expressed in the formula as a frequency-width product. The attenuation associated with a duct terminated in free space is for a frequency (f) is,

$$T_{end} = 10\ \log\left[1 + \left(\frac{c}{\pi f D_e}\right)^{1.88}\right] \quad dB$$

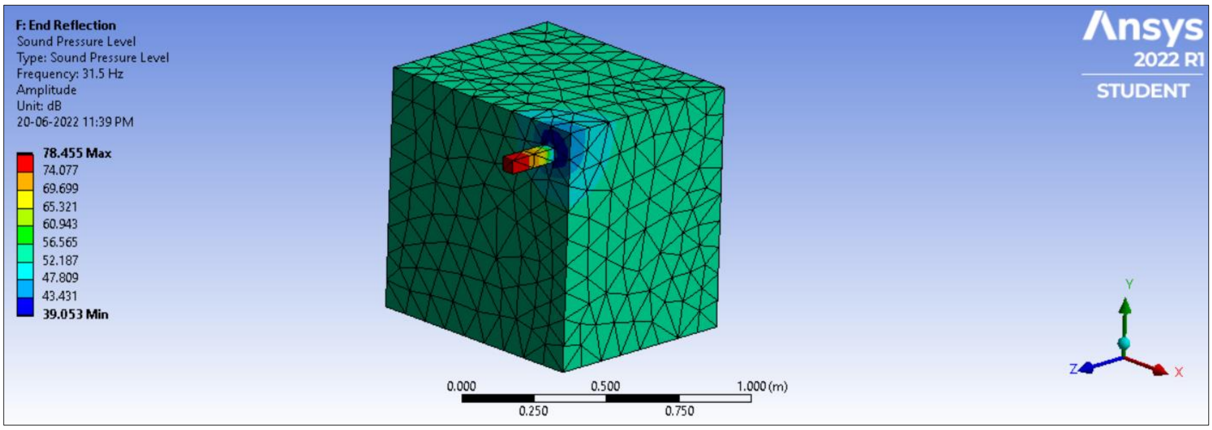

Figure 22: Simulation of duct opening into a room to calculate end reflection loss

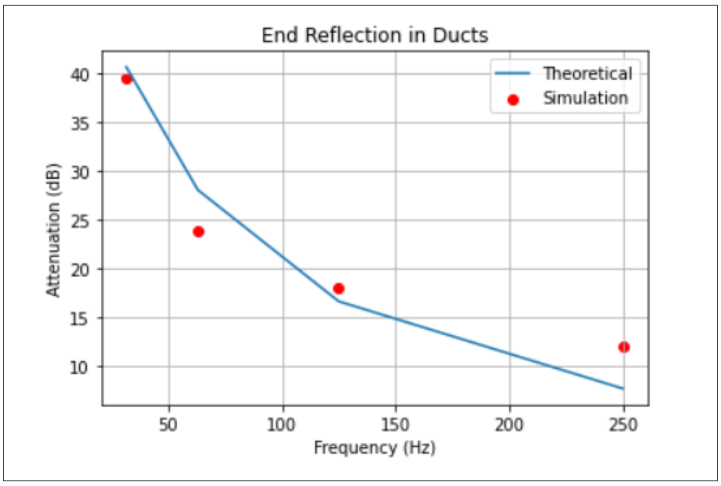

Figure 23: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for duct opening

This graph shows us that for frequencies below the cut-off frequency, the simulation and the theoretical models have similar outputs.

5.8 Silencers

Silencers are commercially available attenuators specifically manufactured to replace a section of duct. They are available in standard lengths in one-foot increments between 3 and 10 feet, and sometimes in an elbow configuration. They consist of baffles of perforated metal filled with acoustic absorptive, which alternate with open-air passage ways. Silencer manufacturers publish dynamic insertion loss (DIL) data on their products. This is the attenuation achieved when a given length of unlined duct is replaced with a silencer. Insertion loss data are measured in both the upstream and downstream directions at various air velocities. Silencer losses in the upstream direction are greater at low frequencies and less at high frequencies. Silencers create some additional back pressure or flow resistance due to the constriction they present. Silencers that minimize this pressure loss are available but there is generally a trade-off 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, relative to other components, also affects the back pressure [7, 41, 46, 47].

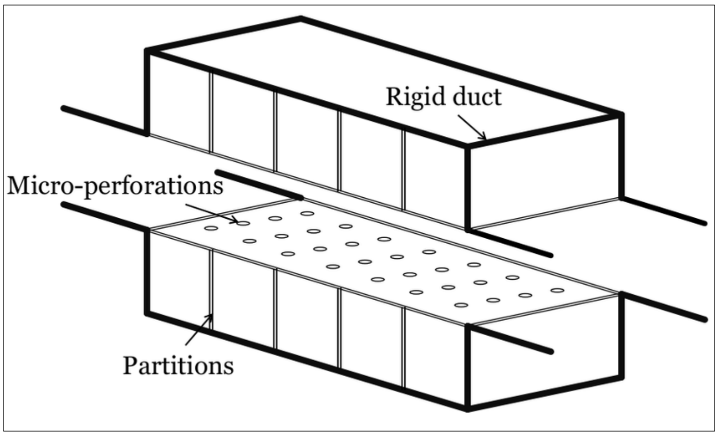

Figure 24: Assembly of Silencer

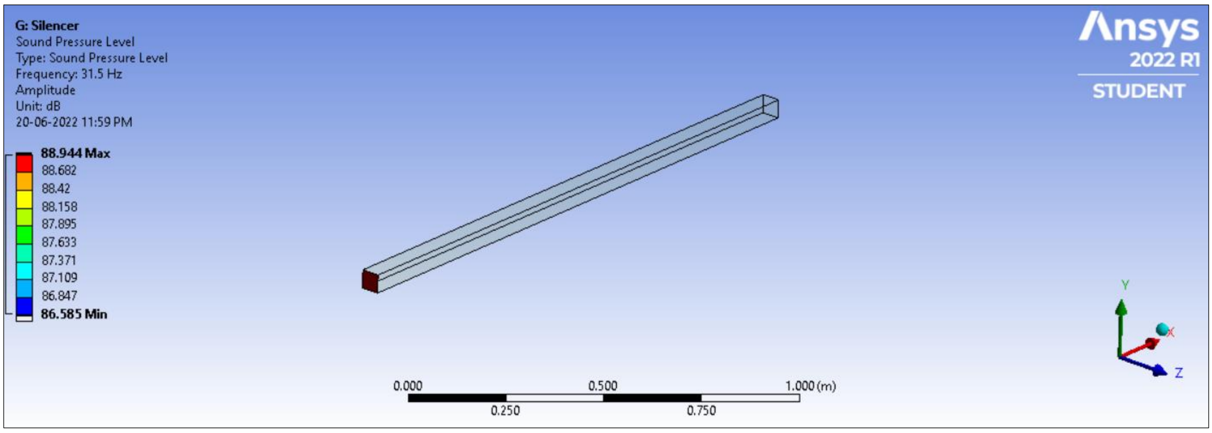

Figure 25: Silencer simulation. The air channel inside the silencer can be seen in the above image

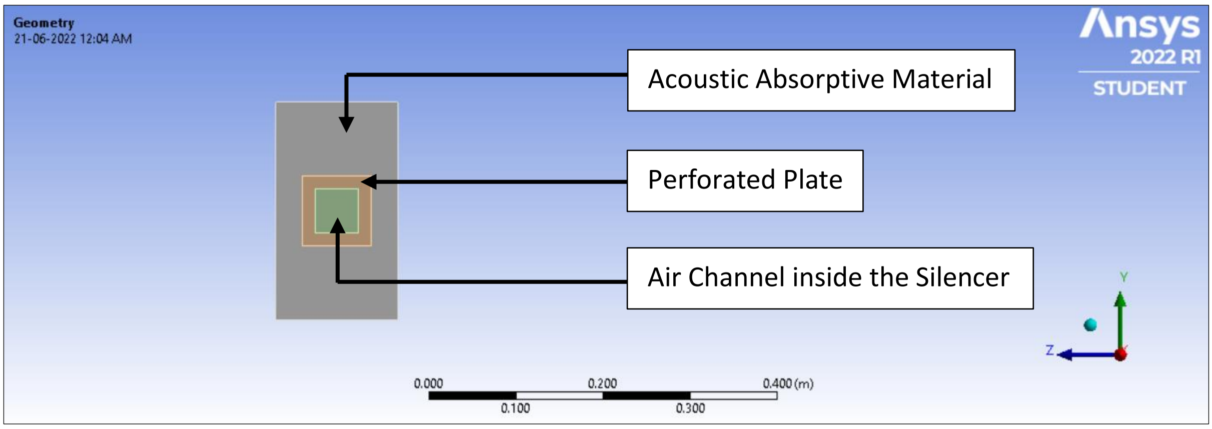

Figure 26: Cross sectional view of a typical silencer showing the material composition

The silencer consists of numerous partitions / baffles which absorb sound. However, due to software limitations, it is possible to only simulate one single continuous baffle. This limitation incorporates some error between the simulation and empirical values as seen in the graph below.

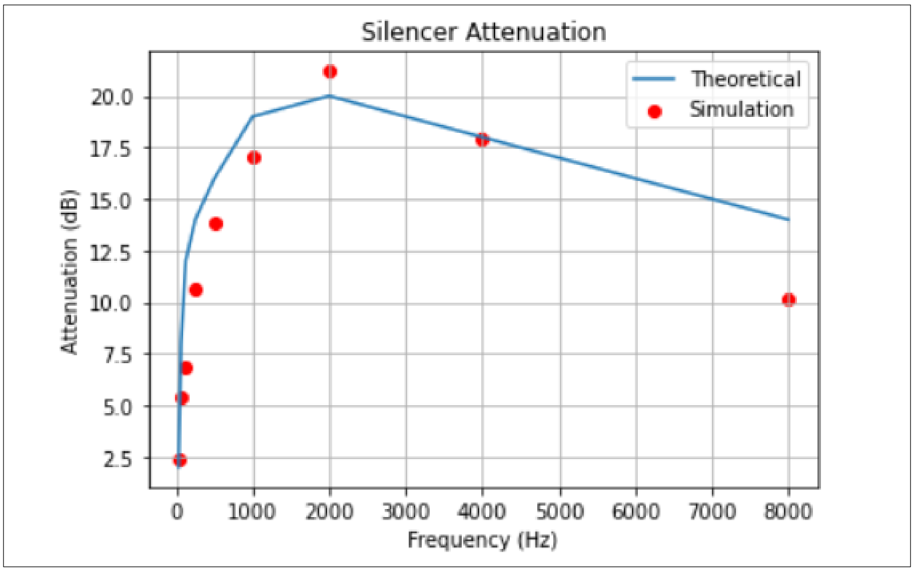

Figure 27: Graph of theoretical and simulation values for attenuation of sound pressure level over the octave band frequencies for a silencer

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