PREDICTION OF NOISE TRANSMISSION IN A SUBMERGED STRUCTURE BY STATISTICAL ENERGY ANALYSIS

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ABSTRACT

PREDICTION OF NOISE TRANSMISSION IN A SUBMERGED STRUCTURE BY STATISTICAL ENERGY ANALYSIS

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The aim of this study is to develop a sound transmission model that can be used to predict the vibration and noise levels of a submerged vessel. The noise transmitted from the mechanical vibrations of the hull of a submarine and the turbulent boundary layer excitation on the submarine are investigated. A simplified physical model of the submarine hull including the effects of bulkheads, end enclosures, ring stiffeners and fluid loading due to the interaction of the surrounding medium is presented in the study. An energy approach, i.e., Statistical Energy Analysis (SEA) is used for the analysis because the characterization of the hull of the structure can be done by a very large number of modes over the frequency range of interest and the deterministic analysis methods such as finite element and boundary element methods are limited to low frequency problems. The application consists of the determination of SEA subsystems and the parameters and the utilization of power balance equations to estimate the energy ratio levels of each subsystem to the directly excited subsystem. Through the implementation of SEA method, the sound pressure levels of the hull of the structure are obtained. In terms of military purposes, the sound levels of the submarine compartments are vital in the aspects of the preserving of submarine stealth.

Keywords: Submerged Structure, Noise Transmission, Statistical Energy Analysis, Structure-Borne Noise.

ÖZ

SU ALTI YAPILARINDA İSTATİSTİKSEL ENERJİ ANALİZ YÖNTEMİ İLE GÜRÜLTÜ TAHMİNİ

Cavcar Yayladere, Bahar
Yüksek Lisans, Makina Mühendisliği Bölümü
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Bu çalışmanın amacı, bir su altı yapısının titreşim ve gürültü seviyelerini elde etmek için kullanılan bir ses iletim modeli geliştirmektir. Bir denizaltının gövdesinden kaynaklı mekanik titreşimler ve denizaltı üzerinde türbülanslı sınır katmanından dolayı oluşan uyarmadan iletilen gürültü incelenmiştir. Su altı yapısının bölmeleri, son muhafazaları, halka güçlendiricileri ve su altı yapısında çevreleyen ortamın etkileşimi nedeniyle oluşan sıvı efektleri dahil olmak üzere denizaltı gövdesinin basitleştirilmiş bir fiziksel modeli çalışmada sunulmuştur. Analiz, enerji yaklaşımlı metot kullanılarak yapılmıştır çünkü ilgili frekans aralığında su altı gövdesinin yapısal modları geniş bir aralıkta karakterize edilebilir ve sonlu elemanlar ve sınır elemanlar yöntemleri gibi deterministik analiz yöntemleri düşük frekanslı problemleri çözmekle sınırlıdır. Bu sebeple, İstatistik Enerji Analizi (İEA) sualtı yapısının gövdesi aracılığıyla yayılan titreşimin iletimini tahmin etmek için uygulanmıştır. Uygulama, İEA alt sistemlerinin ve değişkenlerinin belirlenmesi ve her bir alt sistemin doğrudan uyarılmış alt sisteme enerji oranı seviyelerini tahmin etmek amacıyla güç dengesi denklemlerinin kullanımını içerir. İEA yöntemi uygulayarak, su altı gövde yapısı içindeki ses basıncı düzeyleri elde edilir. Askeri amaçlar düşünüldüğünde, denizaltı bölmelerinin ses seviyeleri denizaltının gizliliğini koruması yönünden önemlidir.

Anahtar kelimeler: Su Altı Gövdesi, Gürültü Tahmini, İstatistiksel Enerji Analizi, Gövde Kaynaklı Gürültü.

To My Dear Family

&

To My Love

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SYMBOLS

A	Area
c_0	Speed of Sound in Air
c_B	Bending Wave Speed
c_f	Speed of Sound in Water
c	Phase Speed
c_g	Energy Group Speed
c_L	Longitudinal Wave Speed
δf	Average Frequency Spacing Between Resonances
$\Delta\omega$	Band Width
d	Distance
D	Diameter
E	Elastic (Young's) Modulus
E_i	Energy of Subsystem *i*
η_i	Damping Loss Factor
η_{ij}	Coupling Loss Factor
f	Center Frequency of the Band in Hz
f_c	Critical Frequency
F	Force
G	Shear Modulus
h	Thickness
J	Torsional Moment of Rigidity
I_p	Polar Area Moment of Inertia
k	Wave Number
k_L	Wave Number of Longitudinal Waves
k_B	Wave Number of Bending Waves
L	Length
L_P	Sound Pressure Level
L_w	Sound Power Level

m	Mass
$M_i(\omega)$	Modal Overlap Factor
μ	Poisson's Ratio
N	Number of Modes
ΔN	Number of Modes in Frequency Band $\Delta\omega$
n	Modal Density
$n(\omega)$	Modal Density (rad/s)
n(f)	Modal Density (Hz)
ω	Center Frequency of the Band in rad/s
P	Power
p	Pressure
P_{in}	Power Input
P_d	Power Dissipated within Subsystem *i*
P_{ij}	Power Flow from Subsystem *i* to Subsystem *j*
$<p^2>$	Mean Square Acoustic Pressure
Q	Torsional Rigidity
ρ	Density
ρ_0	Density of the Fluid
R	Radius
t	Time
RT	Reverberation Time
τ_{ij}	Wave Transmission Coefficient
U	Volume Velocity
$<v^2>$	Mean Square Velocity
X_0	Distance from the leading edge of turbulent boundary layer
z	Height

ABBREVATIONS

ABN	Air-borne Noise
SBN	Structure-borne Noise
SEA	Statistical Energy Analysis
FEA	Finite Element Analysis
FEM	Finite Element Model
CLF	Coupling Loss Factor
TBL	Turbulent Boundary Layer

CHAPTER 1

INTRODUCTION

1.1 General Overview

The study of a submerged structure moving underwater involves the prediction of its structural and acoustical behavior. Since underwater noise limits the military effectiveness of the structure and also makes the underwater structure to be detected, the characteristics of vibration and radiated noise are to be determined. Vibrations induced by driving systems and by the motion of the structure in the medium are transmitted through the hull of the structure and result in adverse effects on the structure. The noise generation and transmission are a major concern of naval researchers and special attention is paid to identification of noise sources and transmission paths on structure in submarine design. In this manner, for such complex vibro-acoustic systems, it is vital to determine the system parameters for the analysis of vibration and noise characteristics.

Noise sources on submarines can be mainly characterized in three categories; namely propeller noise, machinery noise and hydrodynamic noise. Propeller noise originates from different mechanisms. For example, the displacement of the water by the propeller blade generates pressure waves in water. Marine propeller noise can be classified into cavitating and non-cavitating noise. Cavitation of the marine propeller is the most prevalent noise source in sea and the dominant noise source of any single marine vehicle. Notwithstanding, submarines and torpedoes often run deep enough under sea to avoid cavitation [1]. In the absence of cavitation, propeller noise has distinct tones associated with the propeller blade frequencies and a broadband noise at higher frequencies (Figure 1-1). The narrowband tonal characters are caused by interactions of the propeller with unsteady flow fields and the structural vibrations induced by the resulting time-dependent forces. The broad-band noise occurs due to the inflow of turbulence into the propeller and trailing edge noise. Moreover, the fluctuating forces due to the propeller rotation in an unsteady fluid make the shaft and thrust bearings vibrate and thus, generate sound at low frequencies, typically less than 100 Hz. Propellers transmit noise to the structure through the shaft line and through the pressure field generated at the submarine's stern.

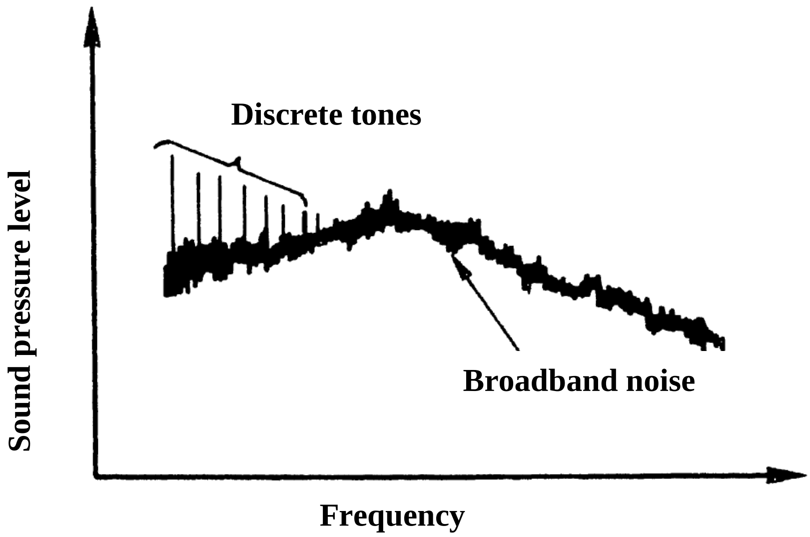

Figure 1-1 Idealized non-cavitating propeller noise spectrum [2]

The main propulsion machinery and associated component such as engines, gearboxes, air conditioning units, pumps etc. are the primary noise sources as well as the propellers. These rotating or reciprocating machines radiate noise to the surrounding air and transmit noise to the structure through its foundations.

Hydrodynamic noise is related with the movement of a fluid medium past a structure. The flow over the structure causes fluctuation and irregularities in the flow and this turbulence in the boundary layer creates a pressure field. Although direct radiation from boundary layers is very weak, a turbulent boundary layer along an elastic solid boundary can develop considerable noise levels. The structural vibrations excited may have distinct resonance peaks in the radiated noise spectrum.

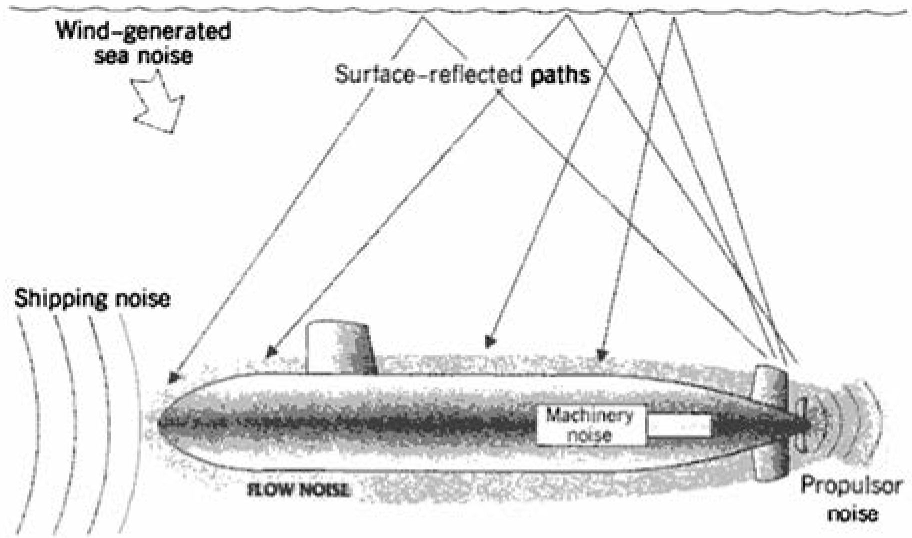

Figure 1-2 Noise sources on a submarine [3]

These noise sources can be treated as the ones that radiate sound directly into the fluid medium (Figure 1-2). However, many noise sources on submarines or underwater structures are not in direct contact with the fluid. Vibrations generated by these sources to radiating surfaces are transmitted to the structure and the surrounding air. The transmission of structural vibrations and radiation of sound produced by the structural vibrations are related with the structure-borne sound. The structure-borne sound is the major cause of noise problems in compartments of submarine and moreover, the sound radiation of submarine is critically important in the aspect of stealth ability of submarine.

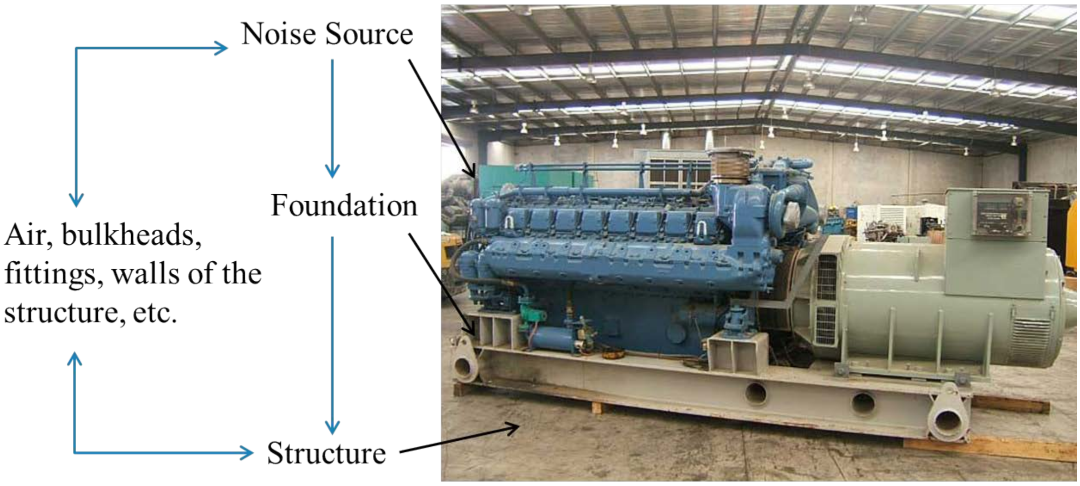

Figure 1-3 Secondary noise transmission in submarine

Acoustic radiation of an underwater structure, which stems from mechanically excitation of ribs and the hull of the structure and the radiation efficiency of the structure, is to be assessed in the manner of characterization of self-noise. From the aspect of structural acoustics of an underwater structure, the estimation of sound pressure level stemming from the machinery, structure's functional systems and from the flow is needed. The methods such as finite element and boundary element are insufficient or impractical when the system is characterized by a very large number of modes over the high frequency range of interest. Moreover, the higher order mode analysis is sensitive to small changes in structure geometry and material property. For such complex problems, Statistical Energy Analysis (SEA) is inevitable methodology that takes into account the irregularities of the system. On the other hand, the moderate detail requirements make SEA less exhausting and expensive than finite element methods. The fundamental of this approach is the study of energy balance with the division of the system into discrete subsystems. The response levels of these subsystems are assumed to be homogeneous. The subsystems are associated through linearly coupled with each other. By solving these linear equations, the response of the subsystems and the average response of the system are obtained in terms of energy input from excitation forces.

The application of SEA method in marine structures is not as popular as the application in aerospace structures. On the other hand, the deterministic models are used to investigate the vibrations of the hull of the underwater structure and the radiation sound. However, it is inadequate in terms of the vibration transmission through the submarine hull because the structure itself has irregularities, discontinuities such as frames, bulkheads. Moreover, deterministic analysis gives less accurate results for high frequency problems.

Due to the fact that structure-borne noise modeling of an underwater structure is an important issue, a careful assessment and analysis should be conducted. In this study, the aim is to present the studies on modeling such complex structures and to provide a SEA model to predict the parameters of the vibrations of the structure and the radiated sound.

1.2 Motivation for the Study

It is crucial that submarines or submerged bodies in sea run as quietly as possible to reduce the probability of detection. An underwater structure can be observed by other structures through its radiated noise. Submarine stealth depends on the level and character of its radiated noise. Significant vibration levels may stem from the submarines' machinery and the flow are transmitted through the hull of the structure, which is unwanted for quiet operation. Many noise sources, namely, the auxiliary machines of the submarines are not in direct contact with the fluid but the vibrations generated by these sources when not absorbed by special mountings or techniques in-situ are transmitted to radiating surfaces. Propellers are located on the longitudinal axis of a body of revolution, aft of the control surfaces. The fluctuating forces created by the propeller can be transmitted through the radiating surfaces and remarkably high radiated sound levels can be observed. In addition, a turbulent boundary layer along an elastic solid boundary can also generate significant noise levels although direct radiation from boundary layer is very weak comparing to other noise sources.

The noise levels both for the near-field and far-field are important at the sides of noise exposure of the submarine crew and survivability of sensitive equipment mounted in submarine and detectability of the structure, successively. Reducing this structure-borne radiated noise is necessary for the structure in order to provide comfortable and quiet working space for the crew and the inside equipment and preserve its stealth in operational conditions against the enemy vessels around the structure.

The power can be transmitted through bending, longitudinal, transverse shear and torsional waves to the structure. The noise radiated from a structure into an adjacent media is mainly caused by the flexural (bending) motion of the structure. Longitudinal vibrations are generally responsible for the generated sound due to cross sectional alterations interacting with them. However, in marine structures the power radiated by longitudinal waves may have some significance. When the surrounding fluid is air, the radiated sound power by longitudinal waves is comparatively much smaller than the sound power radiated by bending waves.

The radiated noise from a submarine is correlated with the vibration levels of the hull of the structure. Dynamic modeling of a submarine is carefully assessed for the prediction of the vibro-acoustic response. The frequency range of interest for submarine applications is wide, from 1 Hz to several kHz. The numerical model should be able to solve the problem in this frequency range for a cylindrical shell. Therefore, structure-borne noise prediction using SEA method is the basic concern of this thesis.

1.3 Scope and Objectives

The aim of this study is to investigate the vibration and noise characteristics of a submarine model by using SEA approach. When implementing the method, the parameters related with the structure-borne noise problem are defined. In order to perform an effective structure-borne analysis, three major concepts have to be considered:

1. The interaction between the noise source and the structure,
2. The transmission through the steel structure,
3. The interaction between the structure and the adjacent media.

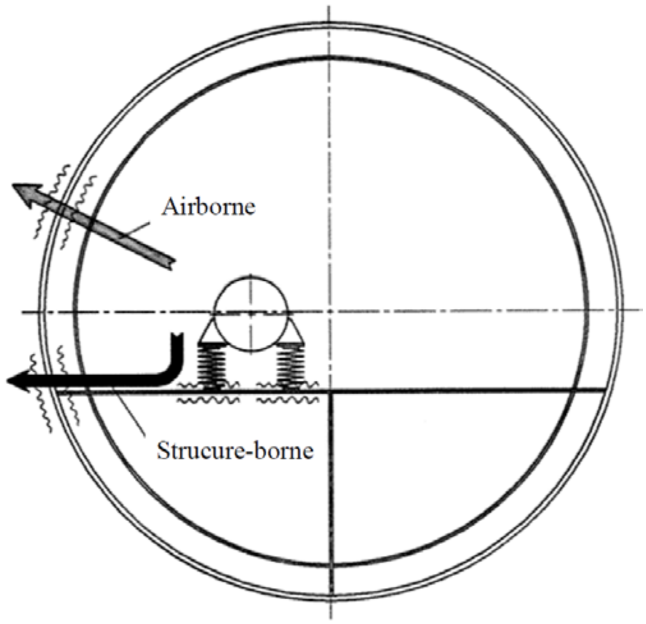

Figure 1-4 Structural acoustics of a submarine

In Figure 1-5, the block diagram showing the study of structural vibrations with the specified concepts.

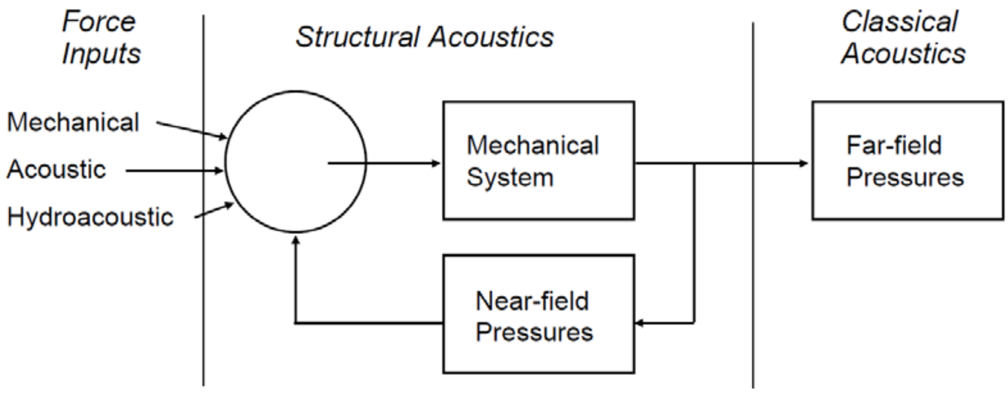

Figure 1-5 Vibro-acoustic mechanical system response [4]

In this study of structural acoustics of submarine, it is important to determine the sound pressure levels generated by the vibrations of submarine hull in near field and far field. The applicability of SEA method in estimation of the noise transmitted to the mounted equipment, especially sonar arrays on submarine is investigated with this thesis. The results can be considered as a first step of more detailed SEA analysis of structure-borne noise from the noise sources to the hull. It is the part of the overall noise analysis of a submarine and the process of creating proper tools for submarine design.

With the specified concepts, it is easier to reach an effective result applying SEA. The studies on structure-borne sound were first conducted in 1940’s, in the field of submarine noise reduction and building acoustics [1]. Thus far, in many engineering fields, especially in architectural, marine and aerospace, structure-borne sound concept has been introduced and many studies have been developed. The developed methods and equations are used in the analysis.

1.4 Outline of the Discussion

The outline of the dissertation presented is given below:

In Chapter 2, a theoretical background of the subject is explained. A detailed survey on sound radiation of underwater structures is presented. Then, SEA studies with the theory and its application to the engineering fields are discussed. Applications of SEA method on the prediction of sound radiation of the underwater structures are explained.

In Chapter 3, the basic theory of SEA method is introduced. The SEA equations and parameters are given. The general procedure of the method is provided.

Chapter 4 gives the necessary information on model development of this study. The submarine model to be analyzed is described in detail. The application methodology of SEA approach to the model used in this study is explained. The techniques for the analysis in the software products are developed in the chapter.

In Chapter 5, the analysis is described and the results are presented in the tabulated forms and figures.

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

CHAPTER 2

LITERATURE SURVEY

2.1 Survey on Sound Radiation of Underwater Structures

The prediction of sound radiated from underwater structures is carried out on the base idea of considering the structure as a cylindrical shell. Early work on vibration transmission of cylinder shells in the absence of fluid loading is firstly presented. Hodges et al. [5] presented a detailed model for vibration transmission which takes account the effects of cylindricality and the internal degrees of freedom and resonances of the ribs.

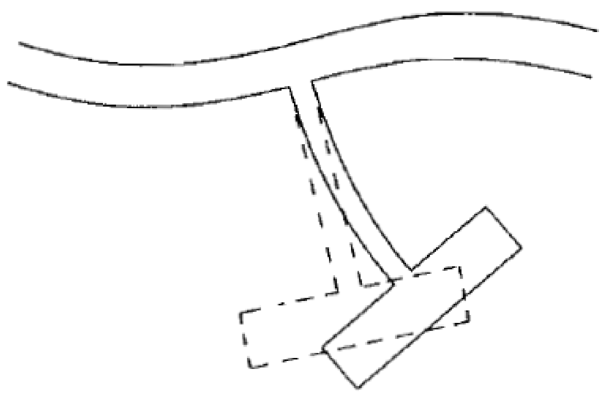

Figure 2-1 Sketch of a portion of the cylinder shell and rib cross section in motion, (greatly exaggerated sketch)

They also compared the theoretical modeling with a range of measurements on ribbed cylinder [6]. Leissa [7] summarized the studies on shells and cylindrical shells and presented the set of equations contributing the thin shell theories. The propagation of waves in shells with periodic stiffening rings was modeled using spectral finite element method with a small number of elements by Solaroli et al [8]. In this study, the stiffening elements were modeled as lumped elements using Euler-Bernoulli theory for curved beams. The discontinuity introduces by the stiffening elements generated a typical stop/pass band pattern of periodic structures. Thus, the prediction of the wave propagation in the structure was carried out and the comparison of the experimental and analytical results was achieved.

Some research showed that fluid loading affects the response of a submerged vessel significantly. The fluid loading coefficients for a finite cylindrical shell were determined by Sandman [9]. Liao et al. [10] remarked an analytical study on vibration and acoustic radiation from a submerged periodic cylindrical shell with axial stiffening.

Liao et al. [11] recently proposed an approximate analytic method to study the sound radiation characteristics of a finite submerged axial periodically stiffened cylindrical shell which is excited by radial forces harmonic forces. The axial stiffeners interacting with the cylindrical shell was described by additional impedance on the model. With the derived expressions, the characteristics of the vibration and sound radiation of the shell were dependent on the mechanical impedance of the shell, the radiation sound impedance and the additional impedance from the axial stiffeners. It was pointed out that the axial stiffener number did not affect the radiated sound power significantly as the structural damping did. Structural damping also affected the radial quadratic velocity on the surface of the shell remarkably. These results could be used for noise reduction problems.

Caresta and Kessissoglou [12] developed a model including complicating effects such the presence of bulkheads, end enclosures, ring stiffeners and fluid loading due to the interaction with the surrounding medium to predict the acoustic signature of a submarine hull at low frequencies. The thin walled cylindrical shell model was analyzed using Flügge equations, which ware more accurate at low frequencies. The frequency response functions of the axial and radial displacements at the drive point location were presented and the sound radiation due to the radial motion of the cylindrical hull was also predicted. The study also pointed that the bulkheads affect structurally and acoustically.

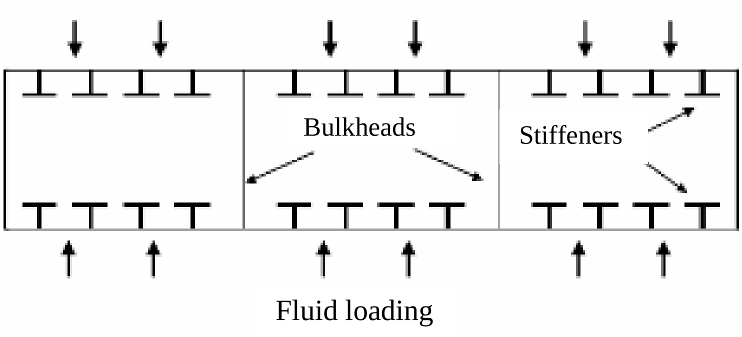

Figure 2-2 Schematic diagram of the cylindrical hull

The detailed description of the dynamic behavior of the submarine structure and structure-borne radiated noise for a cylindrical hull was also presented [13]. The low frequency vibration transmission and acoustic radiation were studied for an underwater structure model including both the cylindrical hull and the conical end caps. The structural and acoustical responses of the system were obtained by using a semi-analytical method. Excitation from the propeller was modeled as a fluctuating radial point force at the ends of the submarine and axisymmetrically distributed load acting on the cylindrical hull. The radial force excitation resulted in greater sound pressure radiated to the far-field when compared with the sound radiation due to axisymmetric loading on the cylindrical hull. The importance of bending modes was pointed out for the evaluation of the sound pressure radiated from a submarine.

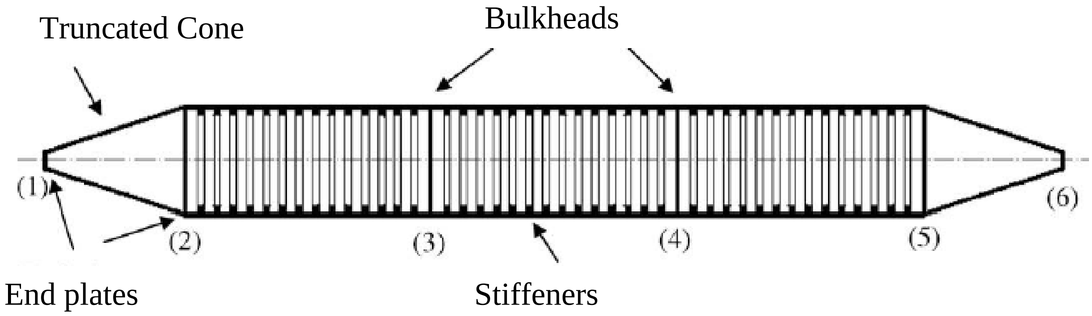

Figure 2-3 Schematic diagram of the model used by Caresta and Kessissoglou [13]

Burroughs and Hallander [14] suggested analytic expressions for the far field acoustic radiation from a fluid-loaded, ribbed cylindrical shell driven by different mechanical drives. The model including the effects of fluid loading, internal damping and interactions between ribs and the shell was used in the approach. The predictions were compared with the measurements and the effects of the shell properties and the type of the drives were examined.

Junger [15] summarized the presented studies on vibrations of shells and plates in an acoustic fluid and on the sound field. The approaches included the Laplace and plane-wave approximation, asymptotic and normal mode solutions, the wave number concept and sound radiation by spatially non-periodic plates and cylindrical shells. The author acknowledged that the focus of the study was on analytical, narrow-band acoustic fluid-structure interactions.

From the aspect of fluid-structure interaction, Ai and Sun [16] studied on the vibration of an underwater cylindrical shell. To evaluate the vibration characteristics of the shell with adding the influence of hydrodynamic and hydrostatic pressures, the analysis on FEM software ANSYS® was employed. In the evaluation of natural frequency and sound radiation of the underwater structure, Ai et al. considered hydrostatic pressure as initial stress stiffness to the system. The analysis remarked that hydrostatic pressure should not be disregarded for acoustic analysis and the characteristics of sound radiation change with water depth.

Khambaswadkar [17] examined the acoustic optimization of an underwater vehicle, lightweight torpedo in his master thesis. The sound radiation from the torpedo was investigated through modeling noise source with the help of experimental data for the gear machinery. He used finite element method both for the fluid and the structure. He revealed that the fluid-structure interaction and the infinite boundary conditions were critical issues on the study of acoustic analysis of underwater structures.

Choi et al. [18] studied on non-axisymmetric vibration and acoustic radiation of a submerged cylindrical shell of finite length with internal elements. They developed a modal-based method in order to analyze the acoustic radiation of axisymmetric submerged finite length shell with bulkheads and subjected to non-axisymmetric time-harmonic loads. The relationship between the surface pressure and the displacement of the shell was determined by Fourier series expansions. Lagrange multipliers were utilized for the connections between the shell and the substructures within the shell. They presented an example of cylindrical shell with hemispherical end caps and bulkheads applying the method and showed that the wave numbers of the dominant flexural waves were the same with the infinite, fluid-loaded cylindrical shell. In addition, it was pointed out that bulkheads could affect the wave numbers of the flexural waves by making them larger. The energy radiations of the shell with bulkheads and without bulkheads were provided with the study.

2.2 Survey on SEA

Statistical Energy Analysis or SEA is a method for analysis of the dynamics between a structure and an acoustical space. The method has been widely used since 1960s. First, Lyon and Smith developed two independent studies on linear resonators. Lyon pointed out the power flow between two lightly coupled, linear resonators which were excited by independent noise sources while Smith focused on the response of a resonator excited by a diffuse, broad band sound field [19]. Lyon concluded that the power flow occurred from the resonator having higher energy to the one having lower energy. Smith resulted that the system reached a limit response when the radiation damping of the resonator exceeded its internal damping. Lyon and Maidanik [20] examined the experimental and theoretical work on the prediction of the response of a single mode system interacting with the multi-modal system.

Several authors have studied evaluation of SEA parameters. Maidanik [21] studied on the dynamic behavior of the sub-systems on SEA model developed by imposing uncoupled boundary conditions, without the effects of coupling impedances. With this study, the multi-mode power transfer coefficient was expressed as an estimate of the average of the individual mode-pair coupling coefficients. This modal approach can be used for the acoustic interaction between the structures, where coupling is light.

Langley and Bercin [22] described an advanced approach for the analysis of high-frequency vibration in the absence of the non-diffuse wave fields caused by the structural joints of the system. Wave intensity analysis consisted of modeling the directional dependence of the vibrational wave intensity in each structural component by using a Fourier series. The conventional SEA was achieved by the addition of non-direct coupling loss factors to the resulting power balance equations. This method could be applicable for bending and in-plane vibrations of plate-structures.

Another study conducted by Manning [23] offered a formulation for SEA parameters, namely power inputs, damping loss factors and coupling loss factors, by employing mobility functions. In the study, the simplifications as averaging the parameters over frequency or over an ensemble of dynamic systems, which were simplifications for the evaluation of SEA parameters, made the method possible for the application of complex structural-acoustic problems. Moreover, Cacciolati and Guyader [24] described a calculation of SEA coupling loss factors by using measured point mobility. The validation of theory was provided with an experiment on plates coupled in three points. The reasonability of the results was obtained through the experiment.

The responses of two multi-modal systems coupled through a point spring were provided by Keane and Price [25]. The spring had a varying coupling loss factor. The modified formulation on proportionality was used in their approach. With their study, the coupling parameters were suggested with a non-conventional approach. In another study, they examined the high frequency noise and vibration prediction on stiffened panels, which are associated with aerospace and marine applications. The stop and pass bands in the periodic structure provided easiness to implement SEA method and the results were especially improved for the intermediate frequency range [26].

Kurtoğlu [27] examined the study of acoustically induced stress analysis for fuselage skin panels of a basic training aircraft. He used SEA method to obtain the stress levels of the panels, modeled including the frames and stringers. The responses for turbulent boundary layer excitation and propeller noise were provided with two different models. The radiation effects from panels to exterior and interior of the sample skin panel as well as the pressurization of the skin panels were investigated separately. The commercial software VA One® was used in this vibro-acoustic analysis. The results were compared with those obtained from AGARD method.

In his study, Kiremitçi [28] investigated the interior and exterior noise characteristics of an aircraft using SEA method. Two different models, one with a ribbed plate model and the other one with hybrid elements, were presented. In modeling SEA, modal densities, coupling loss factors and damping values were determined to predict the interior and exterior sound pressure levels. VA One® commercial software was employed in the analysis. The comparison for two models was provided with the study.

2.3 Applications of SEA on Acoustic Radiation of Underwater Structures

Problems dealing with the dynamics of the interaction of an elastic structure with fluid have been an interest of many engineering applications. The problems can be classified into two categories which are interior and exterior fluid-structure applications. The interior studies are related with the sound propagation within the structure and can be exemplified as piping systems. The exterior studies are associated with the propagated sound that is due to the vibrations of a structure immersed in a fluid medium. In marine structures, exterior fluid-structure applications can be found widely.

There are several studies on the application of SEA method on vibrations of cylindrical shells. The studies about the evaluation of modal densities of cylindrical shells with SEA method were conducted by Heckl [29], Miller et al. [30] and Szechenyi [31] independently. These studies are the development for the SEA method applied to the cylindrical shells.

Liu, Keane and Taylor [32] examined the study on fluid-structure interaction, which is for an interior application, by using statistical energy analysis. The vibration of a coupled fluid filled pipe was carried out with finite element analysis and with the help of the results of the analysis, the model for statistical energy analysis was developed. In the study, two approaches were presented for coupled interior fluid-loaded pipe system. For acoustic and solid, domains direct finite element models were used whereas solids were modeled by modal representation in the coupled system. The dynamic response predictions were obtained for the combined model. The results for the combined model were compared with the ones solved with full finite element method. The study emphasized that the adaptation of modal sub-systems for the combined models was not necessary to get more accurate results than finite element technique. It was also pointed that the frequency averaging method was employable with the addition of experimental data if possible.

Blakemore et al. [33] modeled a fluid-loaded ribbed cylindrical shell with an extended form of SEA method dealing with the periodicity of the structure. The vibration transmission through the pressure hull was studied taking into account the fluid-structure interaction, the pass/stop band characteristics and the effects of bulkheads and the irregularities of the structure. The extension of the method was provided by breakdown of subsystems in a different way and explicit modeling of energy density variation within a subsystem due to damping. They claimed that the fluid around the structure affected not only in the aspect of mass loading of the structure but also adding damping through radiation and fluid avoiding vibration attenuators. The characteristics of transmission modes were determined and through this calculation, the transmission modes on the sections of the ribbed cylinder are grouped into four classes. Three classes were flexural wave motion having different transmission and damping characteristics whereas the other class was related with the surface motion. The transmission and reflection behavior at the junction were considered and the power flow calculations through the elements of the system could be calculated.

Another study of noise transmission on naval platforms done by Tso and Hansen [34] can be found in the literature. They presented a survey of different methods on noise and vibration transmission in naval ships and submarines. This review concluded that SEA could be successful way to analyze the vibration transmission of ship structures formed by mainly plate elements. Moreover, further effort was needed to vibration transmission of ship structures often having complex subsystems such as shells and beam stiffened plates and shells. They also provided their research studies on coupling loss factors (CLF) of cylinder plate coupled structure and a uniform plate with periodic stiffeners. Using wave approach, they developed the SEA modeling for naval ship applications. The experimental and theoretical values of CLFs of the two example structures were compared and they pointed out that a good agreement was achieved in the study. In the analysis of coupled periodic structures, a simplification that only bending waves could be considered in the analysis was offered by the authors. They pointed out that the coupling between the ring stiffened cylindrical shells needed a further investigation and the fluid loading effects had to be taken into account in the analysis of ship platform noise.

CHAPTER 3

THEORY OF SEA

3.1 Introduction

Energy based method, Statistical Energy Analysis, is an approach for the analysis of high frequency dynamic problems. *Statistical* refers to the meaning of drawn of the variables from statistical population. *Energy* is related with the utilization of energy variables whereas *Analysis* means that the process is a method of studying the features and relations of the whole system. SEA is a general approach to identify the interaction of a dynamic system.

The objective of SEA is to predict the response of a dynamic system which is subjected to an assumed distribution of external sources of time-stationary vibrational power input. The SEA method assumes that the frequency band of analysis contains a large number of modes of the considered system and that these frequencies are close to resonance frequencies. The basic concept is that the vibrational energy behaves similarly to heat diffusion from a warm medium to a cold one at a rate proportional to the difference of temperature between the two regions. Using the analogy, the dynamic system is divided into subsystems such that each subsystem has the similar resonant modes. The same type modes such as flexural, torsional, acoustical, etc. can be regarded as subsystems. For instance, a bounded acoustic space can be treated as a subsystem having acoustic modes. The identified subsystems are coupled via junctions and the energy transfer occurs through the junctions. Power balance equations are developed to estimate the averaged vibration level of each subsystem and the vibration level of the coupled subsystems. The concepts of power input, power transmission and power dissipation are defined with system parameters, which provide the determination of energy distribution among the subsystems.

The proportionality of the power flow between the subsystems to the differences of the modal energies of the coupled subsystems bring out that the energy is dissipated within a subsystem related with the loss factor.

The fundamental assumptions that are generally made in the development of SEA models can be stated as [19]:

1. The input power spectrum is broadband and the excitation forces are statistically independent.
2. Energy is not generated in the couplings between the subsystems. The effect of dissipation of energy is added to subsystem damping loss factors in case of dissipation of energy in junctions.
3. The damping loss factor is all the same for equal modes within a subsystem and analysis frequency band.
4. Modes within a subsystem do not interact except to share an equal division of energy.

These parameters are related with the modal approach rather than the concept of travelling waves in the wave approach. Although the same equations and predictions are usually governed by these two approaches, the physical meanings are different from each other.

3.2 Basic SEA Equations

SEA is an analytic method used for the calculation of power flow between connected substructures. The principles of SEA can be illustrated as shown Figure 3-1. The system consists of two connected subsystems.

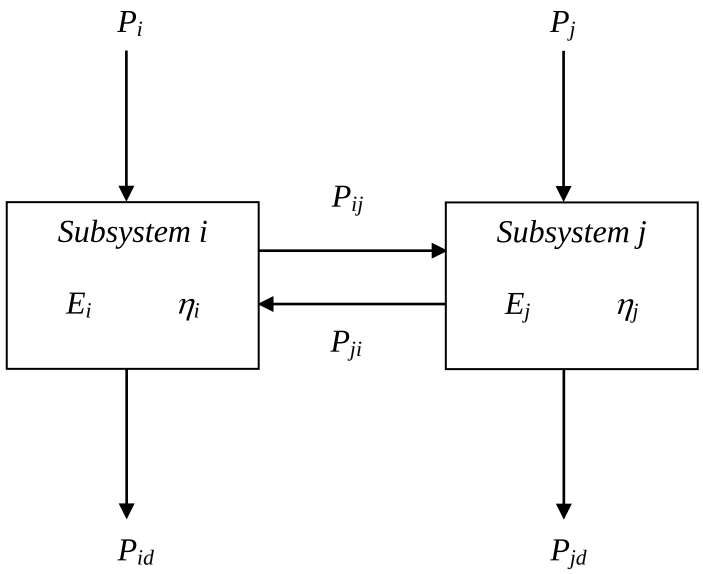

Figure 3-1 Two subsystem SEA model

In the model, energy flow occurs in, out and through the subsystems. The energy flowing to a subsystem consist of external source excitations, P_i and P_j, the energy flowing out of the subsystems is related with the dissipated energy, P_{id} and P_{jd}, and the energy flowing through the subsystems is the energy radiated or transmitted to the other subsystems, P_{ij} and P_{ji}.

The dissipated energy from a substructure can be obtained with the equation;

$$P_{id} = \omega E_i \eta_i \tag{3.1}$$

where η_i is the damping loss factor and E_i is the vibrational energy of the subsystem i in a frequency band with a center frequency of ω. The damping loss factor is a measure of the rate at which energy is exhausted out of a subsystem due to energy dissipation.

The power transmitted from the subsystem i to subsystem j is proportional to the difference of average modal energies of their uncoupled resonant modes, in the frequency range of interest. The power flow between the two subsystems can be written as follows;

$$P_{ij} = \omega(\eta_{ij} E_i - \eta_{ji} E_j) \tag{3.2}$$

In the equation (3.2), ω is the representative frequency, η_{ij} and η_{ji} are the coupling loss factors and E_i and E_j are uncoupled total energies. The coupling loss factors present the characterization of the connection of the two subsystems. It is a measure of the rate of the energy transmitted from one subsystem to another or vice versa.

The "reciprocity relationship" of the coupling loss factors and modal densities can be shown as in equation (3.3). The relationship can be derived from reciprocity equations.

$$n_i \eta_{ij} = n_j \eta_{ji} \tag{3.3}$$

With the substitution of the "reciprocity" or consistency relationship, the equation (3.2) can be arranged as;

$$P_{ij} = \omega n_i \eta_{ij} \left(\frac{E_i}{n_i} - \frac{E_j}{n_j}\right) \tag{3.4}$$

The equilibrium of energy flow for the subsystems is given as in equations (3.5) and (3.6) for the subsystem i and the subsystem j, respectively.

$$P_i = P_{id} + P_{ij} \tag{3.5}$$

$$P_j = P_{jd} + P_{ji} \tag{3.6}$$

Using the definitions for the dissipated and transmitted power, the equation (3.5) can be given as follows;

$$P_i = \omega E_i \eta_i + \omega n_i \eta_{ij} \left(\frac{E_i}{n_i} - \frac{E_j}{n_j}\right) \tag{3.7}$$

Thus the power balance equations can be expressed in the following matrices (3.8). As a consequence, damping loss factors and coupling loss factor data for only one transmission direction are needed to obtain the energy exchange between the two subsystems. Moreover, the substitution of "reciprocity relationship" results in a symmetric loss factor matrix.

$$\begin{bmatrix} P_i \\ P_j \end{bmatrix} = \omega \begin{bmatrix} n_i\eta_i + n_i\eta_{ij} & -n_i\eta_{ij} \\ -n_j\eta_{ji} & n_j\eta_j + n_j\eta_{ji} \end{bmatrix} * \begin{bmatrix} E_i/n_i \\ E_j/n_j \end{bmatrix} \tag{3.8}$$

For a generalized case having *m* number of subsystems, the power balance equations can be written in the matrices below.

$$\begin{bmatrix} P_1 \\ P_2 \\ \vdots \\ P_m \end{bmatrix} = \begin{bmatrix} n_1\eta_1 + \sum_{j\neq 1}^{m} n_1\eta_{1j} & -n_1\eta_{12} & \cdots & -n_1\eta_{1m} \\ -n_2\eta_{21} & n_2\eta_2 + \sum_{j\neq 2}^{m} n_2\eta_{2j} & \ddots & -n_2\eta_{2m} \\ \vdots & \ddots & \ddots & \vdots \\ n_m\eta_{m1} & n_m\eta_{m2} & \cdots & n_m\eta_m + \sum_{j\neq m}^{m} n_m\eta_{mj} \end{bmatrix} * \begin{bmatrix} E_1/n_1 \\ E_2/n_2 \\ \vdots \\ E_m/n_m \end{bmatrix} \tag{3.9}$$

For most systems, many subsystems are not related to each other, meaning no coupling among them, thus, the loss factor matrix has many zero off-diagonal terms being a sparse matrix. This makes the computation much faster than FEM models with the smaller matrices in size and reduction occurs in the required memory for data storage.

3.3 SEA Parameters

In order to obtain the energies of subsystems of a large system, the specification of damping loss factors η_i, coupling loss factors η_{ij}, modal densities n_i, modal overlap factor M_i and the center frequency of the analysis band ω is to be done.

3.3.1 Damping Loss Factor

The damping loss factor of a subsystem can be described as;

$$\eta_i = \eta_{is} + \eta_{ir} + \eta_{ib} \tag{3.10}$$

According to equation (3.10), η_{is} is the structural or material damping of the element. It is a function of the properties of the materials forming the subsystem i, and corresponds to the internal conversions to heat. η_{ir} is related with acoustic radiation losses from the subsystem to the surrounding fluid medium corresponding the conversion of vibrations to sound. η_{ib} is the loss factor associated with the energy dissipation at the boundaries of the structural element.

In most cases, the influence of η_{ib} may be neglected due to rigid connection of structures [33]. Thus, the damping loss factor includes the material damping and radiation damping.

The damping loss factor can be obtained by analytical and empirical approaches in addition to experimental methods.

3.3.2 Coupling Loss Factor

Coupling loss factor η_{ij} is a parameter that provides the relationship of connected subsystems. The efficiency of energy transmission from one subsystem to another subsystem is related with coupling loss factor. It is used in SEA modeling and coupling loss factor depends on the type of junctions and the properties of the connected subsystems. It is difficult to obtain coupling loss factor for connected subsystems because coupling loss factors are dependent on geometric details of the connection and different wave types are generated in structures and at connections.

In order to obtain coupling loss factors, theoretical, experimental and numerical procedures can be used. For theoretical expressions, there are two ways for evaluating coupling loss factor of connected structures namely modal approach and wave approach. In modal approach, the coupling between the individual modes is evaluated and an average is taken for each mode individually in the frequency range. In wave approach, transmission of the waves crossing across junctions is used to reach a coupling loss factor. The coupling loss factor is related with the power transmissibility for the semi-infinite structures. The wave approach is easier to apply since the modal approach involves evaluating complex integrals to evaluate the average values over space and frequency.

The wave approach includes several assumptions. First, the coherence between direct fields of different connections to the same subsystem is neglected. Secondly, it is assumed that the coupled elements are semi-infinite in extent therefore width and length properties of subsystems are not taken into account for the calculations. Thirdly, no power dissipation in the coupling occurs in the calculation of transmission efficiencies. The last assumption is that SEA modeling is generally held to be accurate for “weak” rather than “strong” coupling. Although the exact definition of “weak” and “strong” coupling is not exactly made, the method is applied for “weak” coupled systems. These assumptions simplify the calculation of coupling loss factor.

In order to calculate the coupling loss factor between two subsystems the size and orientation of the coupling, the impedances (or mobilities) of the subsystems at the location of the coupling and the average frequency spacing of the modes of the subsystems are the needed parameters.

It is also possible to relate the coupling loss factor with the transmission coefficient. The fraction of incident power which is transmitted is called transmission coefficient τ. It is given in the equation (3.11) where P_{trans} is the power transmitted to subsystem j across the junction and P_{inc} is the incident power on junction from subsystem i, l_f is the mean-free path length between incidences on the junction and c_g is the group velocity, the velocity of energy propagation.

$$\tau_{ij} = \frac{P_{trans}}{P_{inc}} = \frac{\omega\eta_{ij}E_{tot}}{E_{tot}\left(c_g / l_f\right)} = \frac{\omega\eta_{ij}l_f}{c_g} \tag{3.11}$$

For derivations and formulas of coupling loss factors for different subsystems and different junction types can be found in the literature [19], [24], [27] and [28].

3.3.3 Modal Density and Mode Count

The modal density n_i, is a parameter that defines the number of natural frequencies or modes per unit frequency. The capacity of energy storage of a structure can be described with its modal density. As used in SEA equations, modal density is needed to define consistency relationship which provides the energy balance matrix be symmetric. There are three parameters related with modal density of a subsystem: mode count N_i, ΔN_i indicating the number of modes within a frequency band $\Delta\omega$ and δf_i representing the average frequency spacing between modes. The relation between the modal parameters of a subsystem is given in the following equations.

$$n_i = \frac{\Delta N_i}{\Delta\omega} \tag{3.12}$$

$$\delta f_i = \frac{1}{n_i} \tag{3.13}$$

The evaluation of modal densities and related parameters depend on the geometric information about the mode shapes of the system with the dispersion relation for free waves. In wave approach, modal density is related to group velocity, the velocity which energy propagates by a free wave, and dimension of a wave field.

Theoretical expressions and derivations for different structures are available in literature [19], [29].

3.3.4 Modal Overlap

Another important property of a subsystem in SEA method is modal overlap factor *M(ω)*, indicating the ratio of the modal bandwidth to the average frequency spacing. This parameter can be written in terms of the damping loss factor and the modal density as;

$$M_i = \omega \eta_i n_i \tag{3.14}$$

Large values of the product of the modal overlap parameter and the analysis bandwidth result in low variance and a narrow confidence interval whereas small values of the product bring about high variance and wide confidence intervals. The large values of the product imply that the mean of the response is a good estimate but for the case of the small values of the product the mean does not give a good estimate and needs an upper bound in predictions.

Modal overlap may not directly appear in the developed SEA equations. It is often used to justify the appropriateness of employing the wave approach for evaluating coupling loss factors within the system. In addition, this parameter indicates whether the resonance occurring in a subsystem reaches a level for SEA to be able to give reasonable solutions [1].

3.4 Power Input Evaluation

Responses predicted by SEA method are proportional to the power input. Power inputs inject energy into one or more subsystems of a system. External sources can be characterized as forces, moments, pressures or motions. The proportionality of the real part of the subsystem response function, which is called the admittance, can be valid at the point of excitation.

Evaluation of power input can be done by experiments or analytical calculations. In addition, the power input can be found by modal parameters, which are defined in the previous sections.

The mechanical point excitations can be described by the time averaged product of applied force F and the velocity υ;

$$P_{in} = \langle F \nu \rangle \tag{3.15}$$

The applied force F and the velocity υ are related by the point mobility, which indicates the dependent of the power input on the motion of the structure at the excitation point. This relation gives the ratio of the structural velocity to the magnitude of the point excitation load. The real part of the point mobility represents the component of the velocity that is in phase with the load in the frequency domain and thus, transmits energy to the system.

With the relations explained, for a beam or plate driven by a point force power input can be written as follows;

$$P_{in} = \frac{1}{2}\left(\frac{\pi n_i(\omega)}{2m_i}\right)F^2 \tag{3.16}$$

where m_i indicates total mass of a subsystem.

Power input for an acoustical space can be obtained by the averaged product of pressure p and volume velocity U, similarly.

$$P_{in} = \langle pU \rangle \tag{3.17}$$

The input power values are determined by:

1. identifying the component mode group that are excited by the external sources of power,
2. determining the magnitudes of the external sources,
3. calculating the input power to each subsystem using the frequency averaged response function of the subsystem.

3.5 Energy

Vibrational energy is a primary variable in SEA method. This variable can be expressed in terms of more common variables for structural and acoustic subsystems. The total energy of a vibrating structural subsystem is given by;

$$E_i = M_i \langle v_i^2 \rangle = M_i v_{i,RMS}^2 \tag{3.18}$$

where M_i is uniformly distributed mass and $<v_i^2>$ is the mean square vibration velocity averaged over spatial extent of the subsystem.

For an acoustic subsystem, the total energy can be written as;

$$E_o = \frac{V_o \langle p_o^2 \rangle}{\rho_o c_o^2} = \frac{p_o^2 V_o}{\rho_o c_o^2} \tag{3.19}$$

In the equation, V_o indicates the volume, ρ_o is the density and c_o is propagation velocity of sound. $<p_o^2>$ is the mean square acoustic pressure averaged over the spatial extent of the subsystem.

3.6 General SEA Procedure

Statistical Energy Analysis provides an estimate of the equilibrium energies in a network of subsystems that are subjected to a distribution of stationary external forces over time. The general procedure of SEA modeling can be listed as follows:

1. Define the system and its relations with the surroundings for the analysis.
2. Specify the frequency bands for the analysis.
3. Break the system into subsystems.
4. Calculate the properties of the subsystems namely damping loss factor η_i, modal density n_i and coupling loss factor η_{ij}.
5. Determine the external power input $P_{i,in}$ for each subsystem.
6. Formulate the energy balance equations and form matrix given in the equation (3.9).
7. Solve the equations to obtain the average energy in each subsystem.
8. Transform the average energies of subsystems to desired response quantities.

In this study, a sound transmission model for radiated vibration and noise due to propeller, machinery auxiliaries and hydrodynamic effects is represented as “the system”. The model is tried to be as extensive as necessary for reliable predictions. Since the model can be named as a “fuzzy structure” combined with its complexity internal structures, the predictions may be limited [32].

The system is broken into substructures which are group of similar modes within the physical components of the system. The groups of similar modes in physical element are identified as those which have resonance frequencies in the specified frequency band and have similar mode shapes meaning that they have similar values of damping, excitation and coupling parameters. Subsystems represent group of modes corresponding to different wave types in a physical element. For example, an isotropic plate is usually treated as a system having three subsystems separated due to flexural, in-plane quasi-longitudinal an in-plane shear waves. The isotropic plate is characterized by its area, thickness, material properties and damping factor. In another example, a fluid volume namely acoustic cavity can be treated as a single subsystem due to the fact that longitudinal waves are responsible for the sound transmission. Other examples of subsystems are beams, shells, non-isotropic plates, etc.

In subdivision of the system, one can utilize some guidelines in order to make a proper selection for the subsystems which are;

1. For each frequency band, each subsystem should include a minimum number of modes with the fundamental frequency within the band.
2. There should be equi-partition of vibrational energy between modes. This points out that each mode should contribute more or less the same to the overall energy of the subsystem.
3. The subsystems should ideally be weakly coupled. This implies that if a subsystem is subjected to an excitation, the response of the subsystem should be significantly greater than the response of any other subsystem.

Addition to these, some guidelines can be listed for the wave approach analysis.

1. Structural wavelength must be remarkably less than the dimensions of the subsystem.
2. The wave-field in a subsystem should be diffuse, so that waves propagate equally in all directions.
3. The coefficient of wave transmission should be small at a subsystem boundary so that the reflection of most of the energy in an incident wave

occurs or the damping should be sufficiently high to ensure that the dissipation of most of the energy input to a particular subsystem occurs within that subsystem.

Providing that there are no specific rules for making selection of subsystems, following the stated guidelines and conditions provides statistically accurate results. Definition of subsystems is done according to the SEA parameters, which are just previously mentioned in the section, namely;

1. the average frequency spacing between the modal resonances, i.e. the inverse of modal density,
2. the average modal damping loss factor,
3. the total input power to the modes from the external excitations,
4. the average coupling loss factor between mode groups [19].

For subsystems, size criteria can be represented. In terms of a lower limit on the subsystem size it is not necessary to define it. When the subsystem is reasonably well coupled to a complete system with overlapping modal response, the subsystem may have arbitrarily small modal density and still give good results. On the other hand, an upper limit for the size of the subsystem, it is necessary to prevent too much decay in waves propagating a length of a subsystem with relatively high damping. The energy in a travelling wave decays with distance *x*;

$$e^{-2\pi f \eta_d x / c_g} \tag{3.20}$$

where η_d is the damping loss factor and c_g is the group velocity. The upper limit for the maximum dimension L of a subsystem can be expressed as;

$$L < \frac{c_g}{2\pi f \eta_d} \tag{3.21}$$

CHAPTER 4

MODEL DEVELOPMENT

4.1 Introduction

In structural modeling, it is important to reduce the complicated submarine structure to an ensemble of simpler elements. A submarine can be treated as a thin, rectangular shell reinforced with transverse stiffeners and subdivided internally by reinforced panels, bulkheads. It is crucial that the simplified model has still the characteristics of internal structures and the effect of fluid loading.

In this study, an underwater structure is modeled as a rectangular box structure and a cylindrical shell structure for two different analyses. Two different software on SEA are used for the two analyses. In order to predict noise transmission in an underwater structure, FreeSEA and VA One products are used. Due to the difficulty of modeling the geometry of axisymmetric bodies, cylindrical shell structure is not used for the analysis in FreeSEA. It is a demanding task to develop the equations and parameters of SEA calculations for cylindrical shells. The axisymmetric geometry and axisymmetric loading cannot be introduced to the analysis simultaneously. In SEA, the cylinder-plate coupled structures give less accurate results than those with plate-plate coupling [34]. Moreover, the cylindrical structures at high frequency vibrations act as plate-like structures [4]. Based on those aspects, the analysis is restricted to the vibration transmission through plates for the first analysis. The rectangular model can be seen in Figure 4-1 and a section view of a ring stiffened submarine can be found in Figure 4-2.

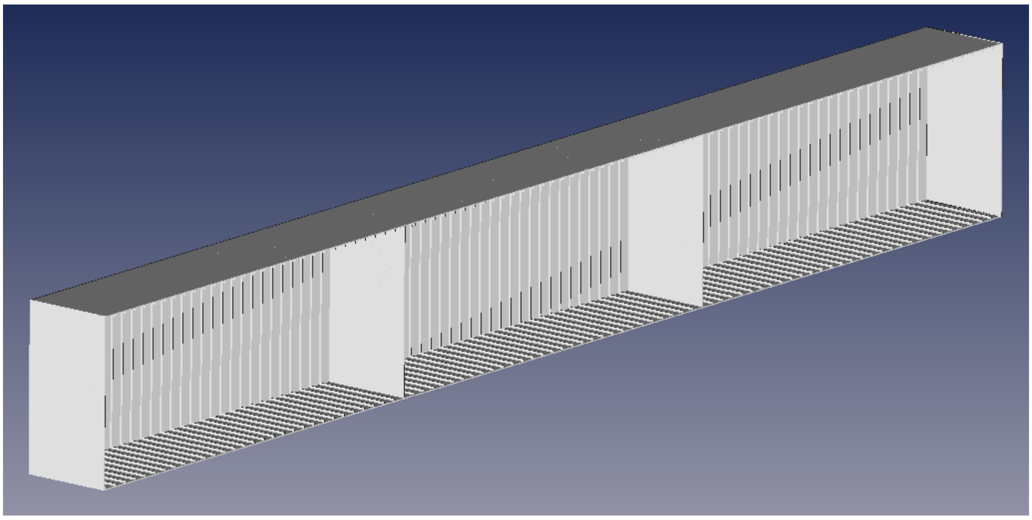

Figure 4-1 Ring stiffened rectangular model used in FreeSEA software

Figure 4-2 Ring stiffened submarine view [36]

4.2 Application of SEA Method

The submarine model in this study is the one used in the study of Caresta and Kessissoglou on low frequency vibration of a submerged hull [13]. Their study examines the structural and acoustic responses of the submarine under harmonic force excitation. The analytical model of a submerged vessel in the low frequency range is presented in the study. The submarine is composed of a 45 m length and 40 mm thick cylindrical shell stiffened with 90 stiffeners, 2 bulkheads and 2 end caps (Table 4-1). The stiffeners have a rectangular cross section 0.08 m x 0.15 m and their spacing is 0.5 m. The properties of the submarine hull and the other physical properties of the model used in the study can be found in the table below.

Table 4-1 Dimensions and properties of the Kessissoglou's model [13]

Parameter	Cylindrical Shell	Conical Shell
Length (L) [m]	45	8.9
Radius (R) [m]	3.25	3.25 / 0.50
Thickness (h) [m]	0.040	0.014
Material	Steel	Steel

For the simplicity of the analysis in FreeSEA, conversion of cylindrical shell into a rectangular shell is needed. The transformation of the radius and length of the cylindrical shell is achieved with the equilibrium of the areas of the two shells (Table 4-2).

Table 4-2 Conversion of cylindrical shell to rectangular shell

Parameter			Cylindrical Shell	Rectangular Shell
Length	(L)	[m]	45	45
Diameter	(D)	[m]	6.5	-
Thickness	(h)	[m]	0.040	0.040
Width	(w)	[m]	-	5.76
Height	(z)	[m]	-	5.76

Table 4-3 Material properties used in the study

Material Parameter			Air	Sea Water	Steel
Density	(ρ)	[kg/m^3]	1.21	1026	7800
Speed of sound	(c)	[m/s]	343	1500	-
Young's Modulus	(E)	[N/m^2]	-	-	2.10E+11
Poisson’s Ratio	(ν)		-	-	0.3

For this study, the structural and acoustical responses of the same structure are predicted for high frequency range. Due to the fact that vibrations in the pressure hull are characterized by a very large number of modes over the frequency range of interest and by high modal overlap, the analysis of prediction of noise and vibration is conducted with software products using SEA method. SEA method provides a probabilistic model for estimating high-frequency response of complex systems. The “high” term used in the analysis implies that the frequency range of concern extends to many times the natural frequency of the system, which is confronted in our model, the pressure hull with its transmission path, its irregularities and discontinuities.

Although the natural frequencies and mode shapes of the high-order modes of complex system cannot be evaluated theoretically with great precision, high frequency dynamic properties of the structure and associated vibration fields can be obtained in energetic and statistical way. The effects of irregularities, making the propagating wave to reflect, diffract and scatter, can be involved with SEA method in the frequency range of interest.

The first step in the study is the definition of the system and its sources, which is the basic of the technique of SEA. As mentioned before, the system is the submerged vessel with its internal structures and its source is the machinery mounted on the back compartment of the vessel and the turbulent excitation of the vessel. The physical system elements are the regularly framed underwater structure, fluid around the structure and the machinery sources. The transmission path is the hull of the structure. The underwater structure can be divided into three regions namely forward compartment, middle compartment and aft compartment with simplification. The center region is an infinite regularly framed rotationally symmetric thin rectangular shell with internal bulkheads and ring stiffeners.

Moreover, the model has machinery compartment in the backside of the structure. The input excitation is introduced into the system from the aft side of the structure. Due to having no information about the machinery and its environment of a submarine, the input excitation is implemented into the structure in terms energy level. This study focuses on the application of SEA to the submarine plate structure. Fluid loading effects are included and the coupling parameters are presented in the calculations. In order to investigate the effects of internal structural elements, bulkheads as vibration isolators, frames as stiffening elements, stop/pass characteristics of the structure and complex geometry are taken into account. The high frequency analysis is to be conducted for the prediction of structural and acoustical dynamics of the submarine. That is, the air-borne noise and structure-borne noise levels are provided with the developed model.

For simplicity, the shell is considered as homogeneous structure. There is no remarkably change in shell thickness or spacing of ribs. The shell can be broken down into physical sections or “chunks”. A chunk is a frame whose boundary is placed at a significant reflector, bulkhead (Figure 4-3). The dispersion characteristics of the infinite cylinder are examined within a chunk.

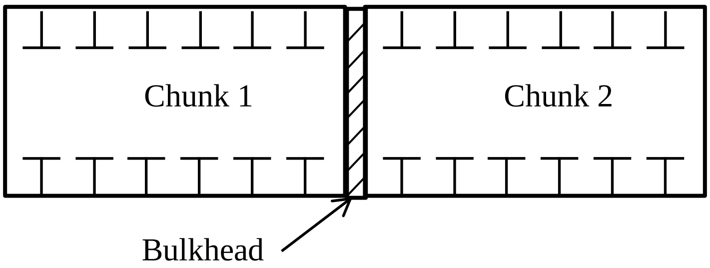

Figure 4-3 Schematic diagram of the model

These chunks can also be named as the subsystems of the models in the analyses. In FreeSEA analysis, compartments are considered as rooms and bulkheads of the submarine are introduced as rectangular plates. The connections of the room and the plate bulkheads are implemented in the analysis through area junctions. The stiffeners are modeled as beams connected to the surfaces through line junctions (Figure 4-4 and Figure 4-5).

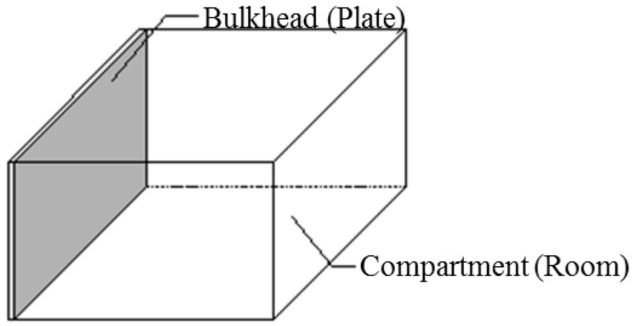

Figure 4-4 Area connection of compartment and bulkhead

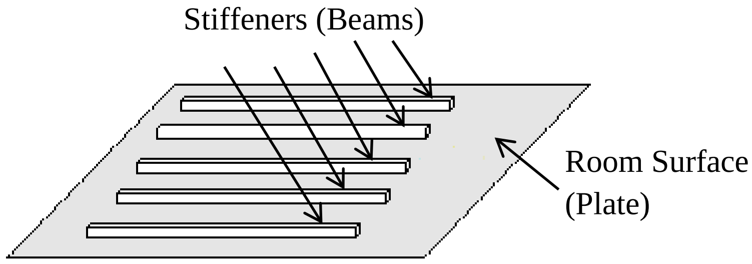

Figure 4-5 Line connections of the room plates and stiffeners

In modeling of the plates of the compartments, line connections of the L-shaped plates are considered. The Figure 4-6 shows the technique how the power is transmitted through the line junction in the compartments.

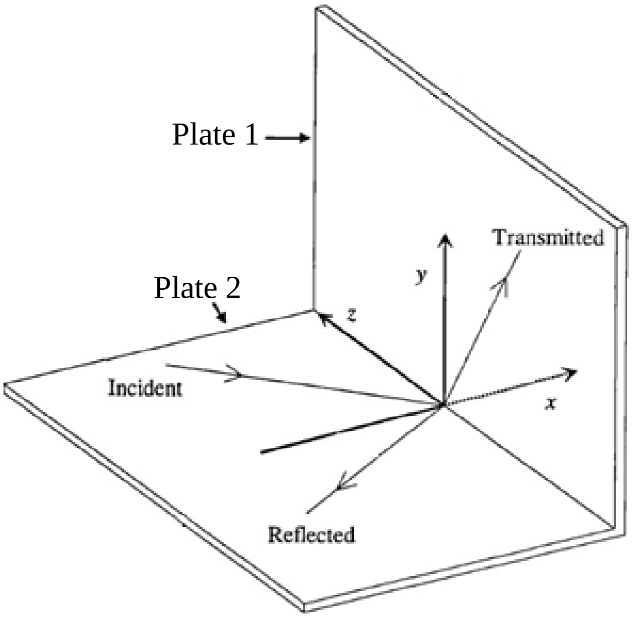

Figure 4-6 Right angled (L-shaped) plate line joint showing incident, reflected and transmitted bending waves

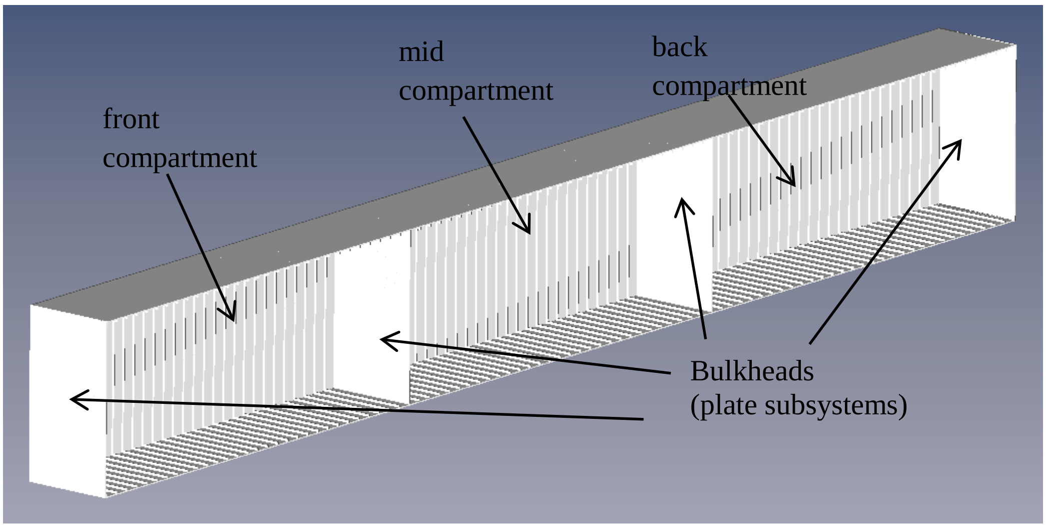

Figure 4-7 Complete SEA model used in FreeSEA

The schematic representation of the cylindrical shell modeled in VA One is given in Figure 4-8.

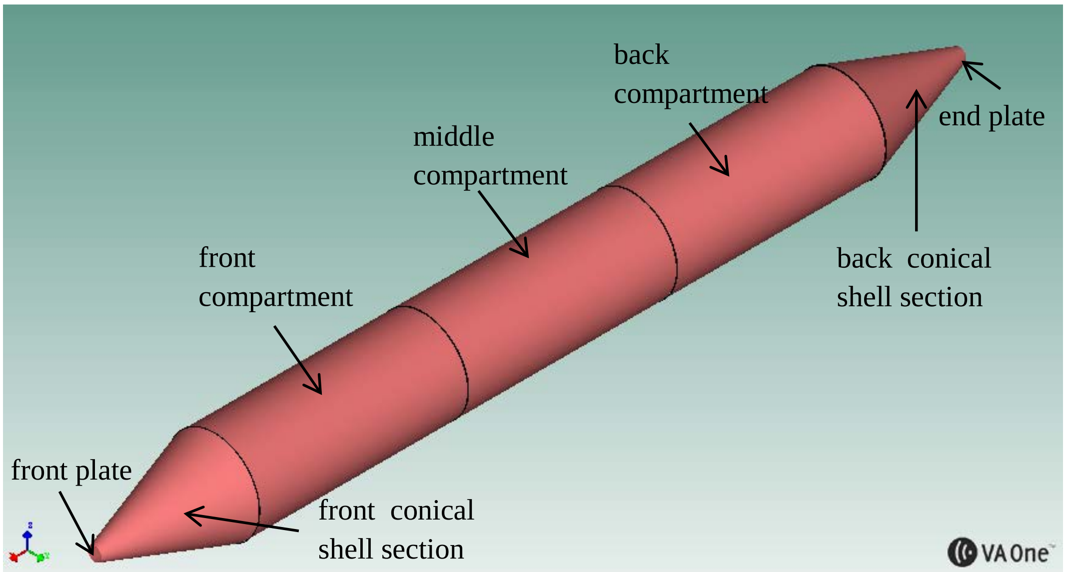

Figure 4-8 Subsystems defined in the analysis

In the same manner, for VA One 2010 analysis, all compartments are presented as cylindrical shell subsystems whereas the bulkheads of the structure are plate subsystems. For shell and plate subsystems, the wave propagates in two dimensions. The ring stiffeners are defined in the software product, which enables the users characterize the subsystem as ribbed plate in the property selection. The fluid medium (air) in the compartments is modeled as acoustic cavities of the system. The acoustic cavities are used to represent wave propagation in three dimensions. The external section of the structure is subjected to turbulent flow by sea water. The blue color indicates the skin of the compartments subjected to sea water, the pink color shows the internal surface of the compartments, which are ribbed shell subsystems, the light blue color points out the bulkheads, plate subsystems in Figure 4-9.

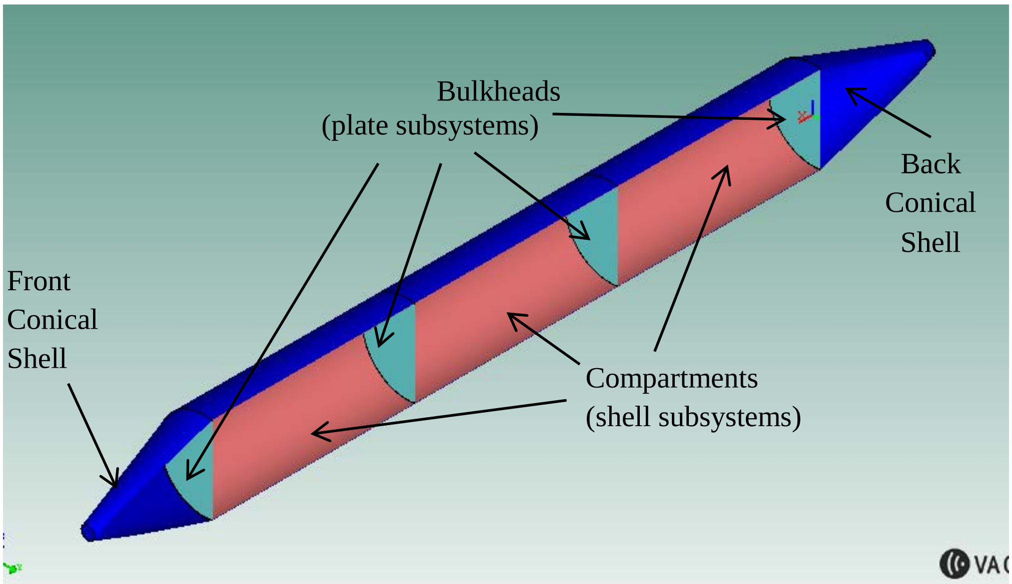

Figure 4-9 Subsystems of the submarine in VA One

In the model, there are 11 subsystems, 5 acoustic cavity subsystems, 11 area junctions and 4 line junctions. The construction of these subsystems and junctions is achieved through the guidelines mentioned in Chapter 3. The each subsystem has the same number of modes in the specified band and the same wavelengths. In addition, the same energy levels are contributed through the subsystem. A general view of the model developed in VA One can be seen in Figure 4-10.

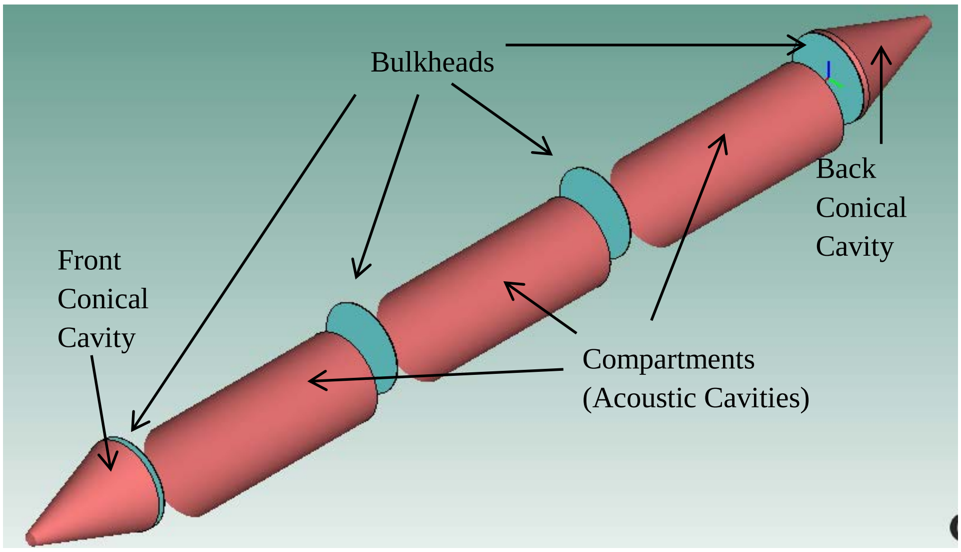

Figure 4-10 Internal section shrink view of the model

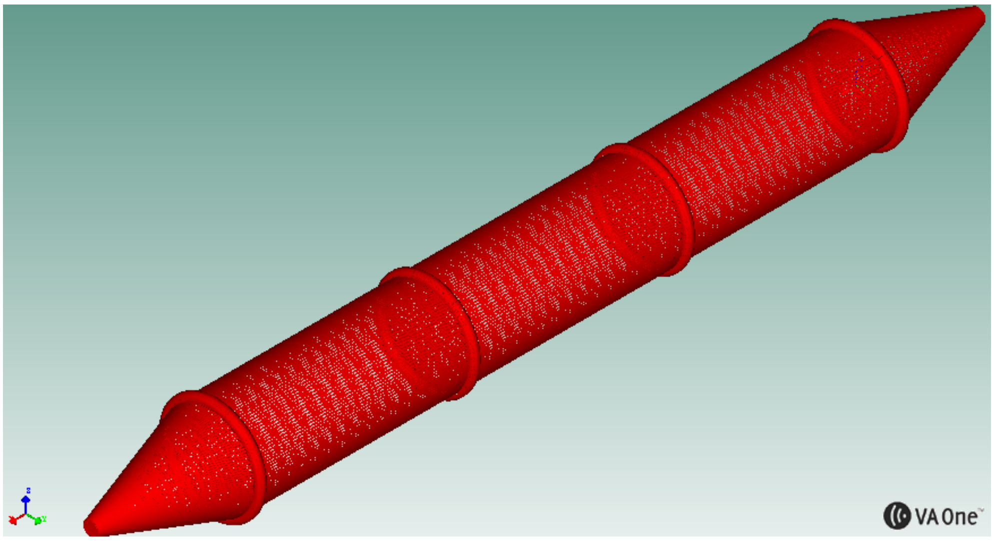

Figure 4-11 Line and area junctions connecting the subsystems

The connections of the subsystems are created by *auto connect* feature in VA One. The view of the submarine model with its junctions is given in Figure 4-11. The plate and shell subsystems are coupled with each other through line junctions, the coupling of the acoustic cavities and shell and plate subsystems is provided by area junctions.

The “ribbed” property of the plates is identified with the capability of the product. In order to model the rib stiffening of the plates, which are created as subsystems, the cross sectional properties of the ribs are utilized in the SEA analysis. This table gives the required properties which are implemented into the software product for the stiffeners having 80 mm x 150 mm rectangular cross section.

Table 4-4 Beam properties used for the ribbed plate subsystem in VA One

Parameter			Rectangular Beam
Perimeter	-	[mm]	460
Area	(A)	[mm^2]	12000
Centroid – Panel Center Distance	(d_1)	[mm]	95
Shear Center – Centroid Distance	(d_2)	[mm]	0
Second moment of area of the beam's cross-section about the beam's X axis	(I_{xx})	[mm^4]	2250 x 10^4
Second moment of area of the beam's cross-section about the beam's Y axis	(I_{yy})	[mm^4]	640 x 10^4
Polar moment of area of the beam's cross-section about the beam's Z axis	(J_{zz})	[mm^4]	2890 x 10^4
Torsional rigidity of the beam	(Q_{zz})	[mm^4]	16932480
Spacing of the beam stiffeners	-	[mm]	500

These properties of the stiffeners having rectangular cross section are introduced into the model. All compartments are modeled as ribbed plate by using the feature of VA One.

After the creation of the subsystems and junctions of the system in the software VA One, modal densities of the subsystems can be evaluated. The modal density of a subsystem gives the number of modes per unit frequency which is described in the previous chapters. The modal densities of the structure's subsystems are given in Figure 4-12, Figure 4-13, Figure 4-14, Figure 4-15, Figure 4-16 and Figure 4-17.

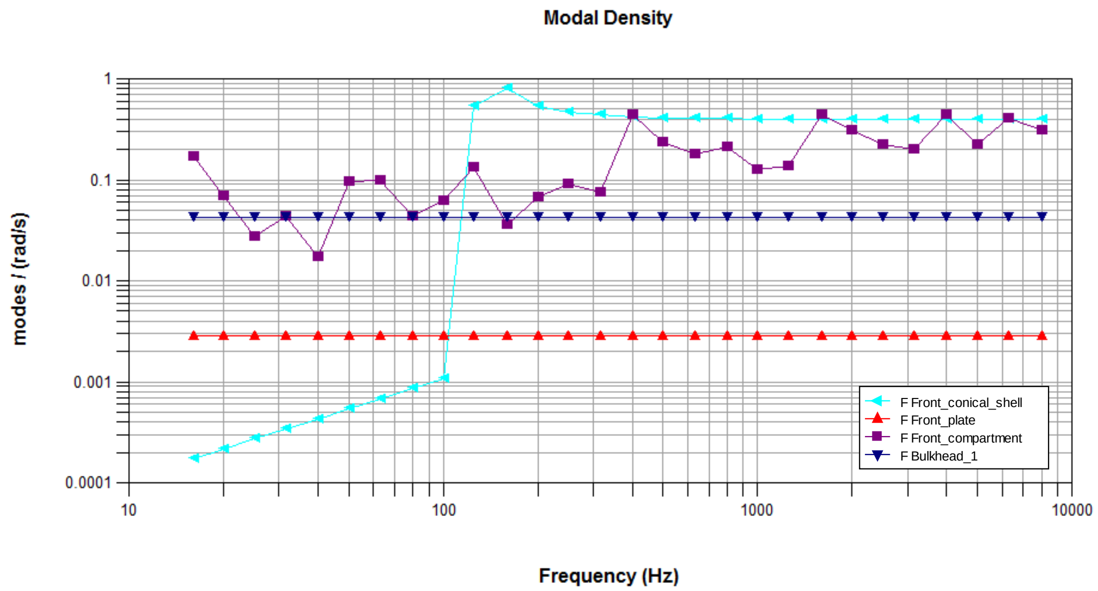

Figure 4-12 Modal densities of front section of the structure

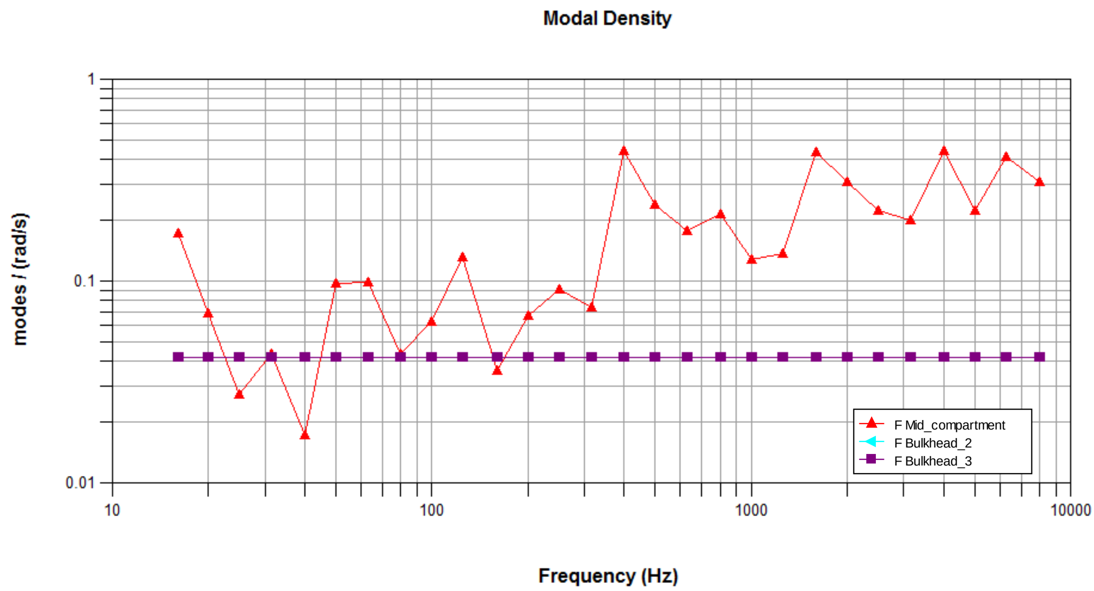

Figure 4-13 Modal densities of mid-section of the structure

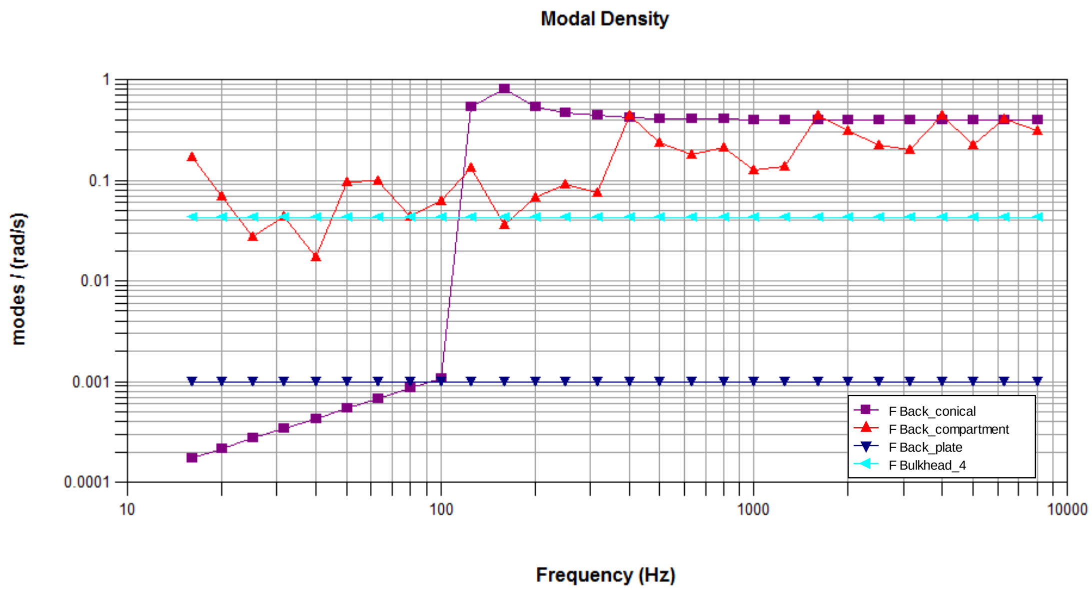

Figure 4-14 Modal densities of back section of the structure

In the first three figures, it can be observed the bulkheads of the system with their same areas and thicknesses have the constant and same values for the modal densities. The biggest values for the modal densities are noticed for the conical shells at higher frequencies. The thickness of the conical shells differs from the thickness of the cylindrical shell. The irregularities of the curves are the reasons of the ribbed properties of the compartments.

Front, mid and back compartments have the same values for their modal densities. The modal density concept is highly dependent on the geometry, material and frequency of the structure. For flat plates, the modal density is independent of frequency and the values for the bulkheads are constant for all the frequencies. At high frequencies, the subsystems' modal densities reach to constant values except the compartments'. This is because the modal density of the shell approaches the modal density of a flat plate [22]. The modal densities of all the subsystems have determined behavior except the conical shells. For low frequencies front and back conical shells have smaller modal densities and at some frequency (100 Hz) the values show a remarkable jump. This is because of the geometry of the shells. The tip areas of the conical shells are very small when compared the areas that connecting with the bulkheads. This results in appreciably alternation in the modal density values. The SEA method is applied when the modal density values of the subsystems are high enough. Thus, above 100 Hz the developed model has reliable results and the frequency range of this problem can be noted as between 100 Hz and 10 kHz.

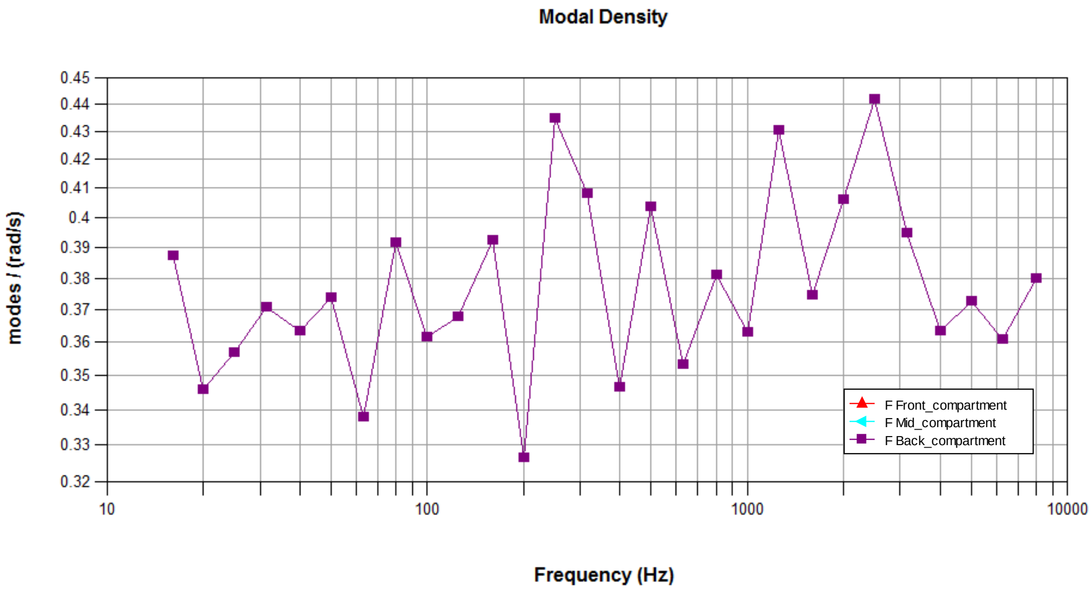

Figure 4-15 Modal densities of all the compartments without ribs (unribbed)

The existence of stiffeners affects the modal density visibly. In Figure 4-15 modal densities' behavior of all the compartments without ribs are presented. It can be concluded that the compartments without stiffeners have generally higher number of modes per unit frequency. Figure 4-16 represents that the cylindrical subsystems with ribs have scattered values of number of modes per unit frequency. At some frequencies (400 Hz, 1600 Hz, 4000 Hz and 6300 Hz), the number of modes per unit frequency for ribbed cylinders has peak values and these values are very close to each other. The unribbed subsystem has relatively high and constant values for modal density.

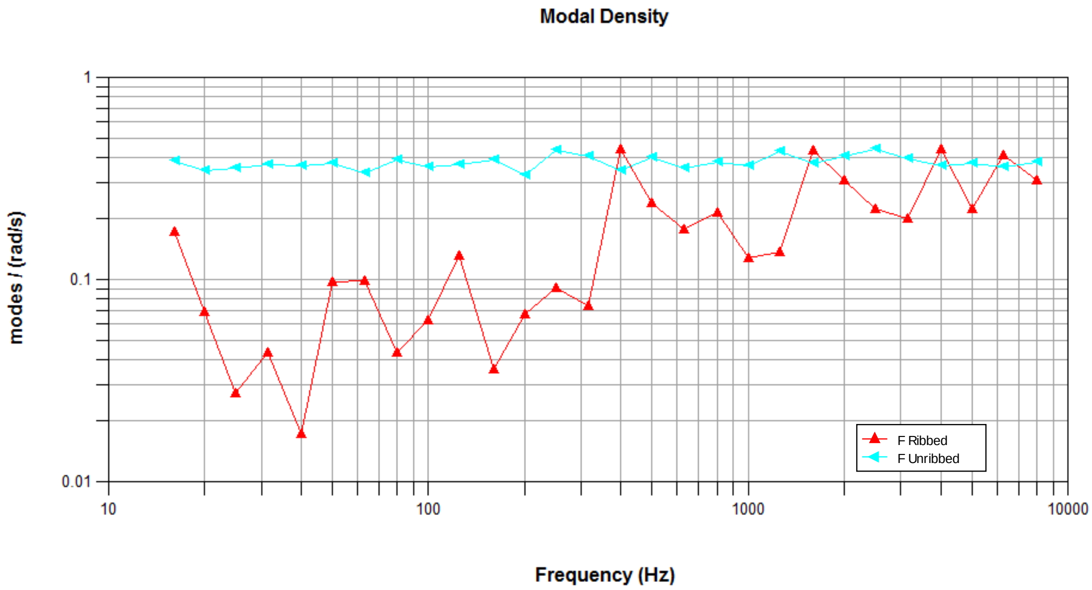

Figure 4-16 Comparison of ribbed and unribbed case for front compartment

Additionally, the five acoustic cavities of the system, which are filled with air, have remarkably high modal densities (Figure 4-17). The conical shape has smaller values for modal density whereas cylindrical geometry indicates bigger values at the same frequency. The differential in the area is the responsible for this dissimilarity. The mid compartment has larger number of modes per unit frequency when compared to other compartments. The location of the mid compartment makes the element to interact with the other compartments. The subsystem becomes limited at the boundaries which provide a high number of modes.

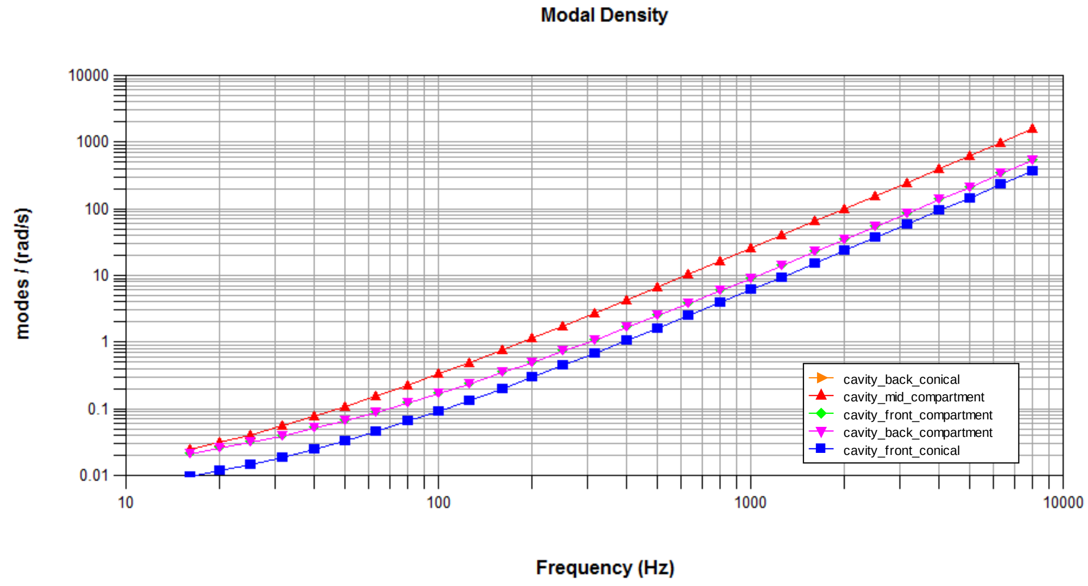

Figure 4-17 Modal densities of the acoustic cavities of the structure

It can be noted that there is no obvious change in modal densities except the conical subsystems. A sharp change in modal densities is observable for the subsystems undergoing area changes. In addition, SEA method is applicable when the modal densities of the subsystems are high enough. For this problem, the results are expected to be valid above 100 Hz.

For the simulation of the structure to be in water with a specified depth, the compartments and conical shells are connected to a free field propagation model called Semi-Infinite Fluid (SIF). The elements being underwater should be introduced to the model due to the fact that fluid loading adds a frequency dependent fluid mass loading to each subsystem. All external subsystems are connected to the SIF subsystem to describe the fluid loading on the system.

The frequency range of applicability of the developed model is checked and the results are presented for frequencies from 31.5 Hz to 10000 Hz.

CHAPTER 5

SEA ANALYSIS

5.1 General

In this chapter, the analyses and results are given in detail. The models described in the previous chapter are analyzed with the specified software FreeSEA and VA One.

There are numerous sources acting on the submarine as explained in earlier chapters. Engine noise, propeller noise and turbulent boundary layer noise among those that can be namely specified. From standpoint of the frequency bands of interest, power plant or engine originated noise and turbulent boundary layer noise are studied for the developed models. Propeller noise excitation is not treated in this study due to its tonal character.

In this study, two types of analyses are conducted. In FreeSEA, engine noise is utilized for the excitation whereas in VA One, both engine noise and turbulent boundary layer noise are introduced into the system.

For FreeSEA analysis, due to the lack of ability of the software product, the realization of the submerged structure is limited. The fluid loading effects and the flow excitation of the sea water on the structure are excluded. The surrounding medium is introduced as sea water but the static and hydrodynamic affects cannot be taken into the account. The focus of the FreeSEA analysis is to note the behavior of the thin, finite length rectangular shell structure, divided into rooms, when the excitation is applied onto the floor plate.

On the other hand, VA One can handle both the system engine noise and turbulent boundary layer noise excitations. Different conditions for the input loadings and speed of the submerged vessel are developed by considering the submerged condition of a submarine. Table 5-1 shows the conditions used in the analyses.

Table 5-1 Configurations for VA One analysis

First Condition TBL noise	Depth	5 m
	Speed	5 m/s
		10 m/s
Second Condition Engine noise (ABN) TBL noise	Depth	5 m
	Speed	5 m/s
		10 m/s
Third Condition Engine noise (ABN & SBN) TBL noise	Depth	5 m
	Speed	5 m/s
		10 m/s

The damping loss factors for all the subsystems are set to 1% for the analyses.

5.2 Analysis by FreeSEA

With the explained model, the analysis is conducted for unribbed and ribbed cases in order to observe the effects of ring stiffening. The input excitation is 1 W applied on the floor plate on the back compartment. The structure is model as surrounded by sea water. However, the turbulent flow excitation is not contributed to the analysis. The hydrodynamic effects on the submarine are not accounted for and excluded in the analysis done by FreeSEA. The developed model is introduced in the program code, which is given in Appendix A

5.3 Results by FreeSEA

The results for FreeSEA product are given in 1/3-octave band frequencies and in the frequency range between 100 Hz and 3150 Hz. The tabulated results are presented in Appendix B.

For the unribbed case, the expected sound levels for air-borne noise of the three compartments are obtained (Figure 5-1). The maximum sound level is observed in the back compartment and around its critical frequency. Critical frequency, also namely coincidence frequency, occurs when the flexural wavelength of the plate is equal to the wavelength of acoustic waves in the air. It can be calculated as;

$$f_c = \sqrt{\frac{3\rho}{E}}\frac{c_0^2}{\pi h} \tag{5.1}$$

For the ribbed case, the same observation can be done (Figure 5-2). Around the coincidence frequency, which is the same as the unribbed case, the air-borne noise levels are high for the three compartments. When the ribbed structure is analyzed, the sound pressure levels within the compartments decrease when compared with the values of the unribbed cases. The difference of the predicted sound levels shows that the coupling of plate-beam causes the vibration level to decrease and transmission loss to increase.

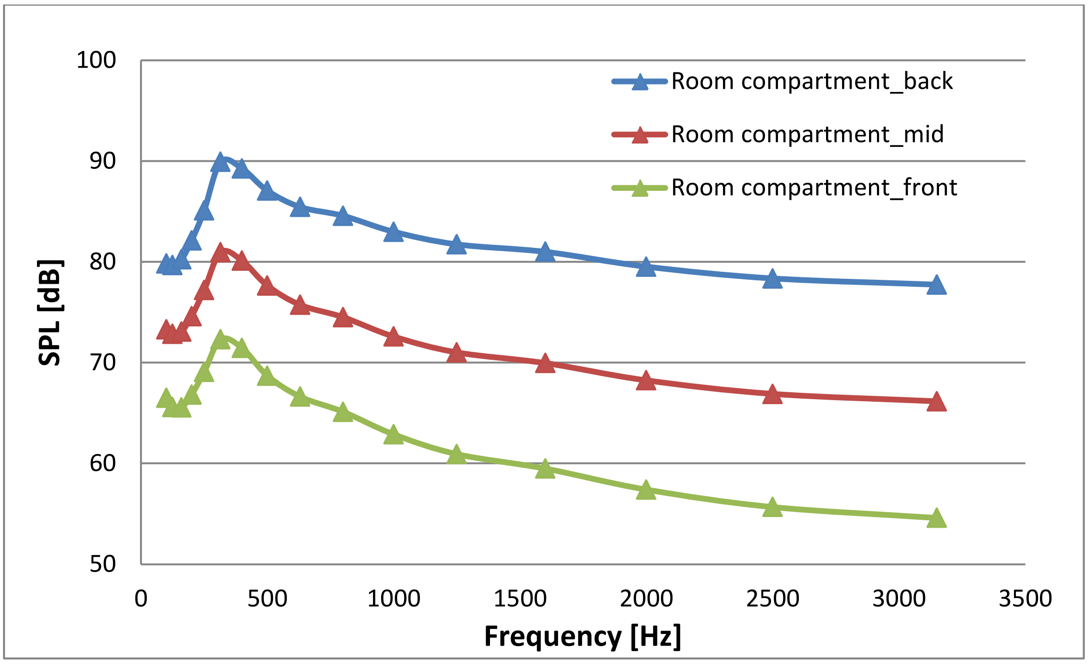

Figure 5-1 Interior noise contributions of the three compartments by excitation from the back compartment (unribbed case)

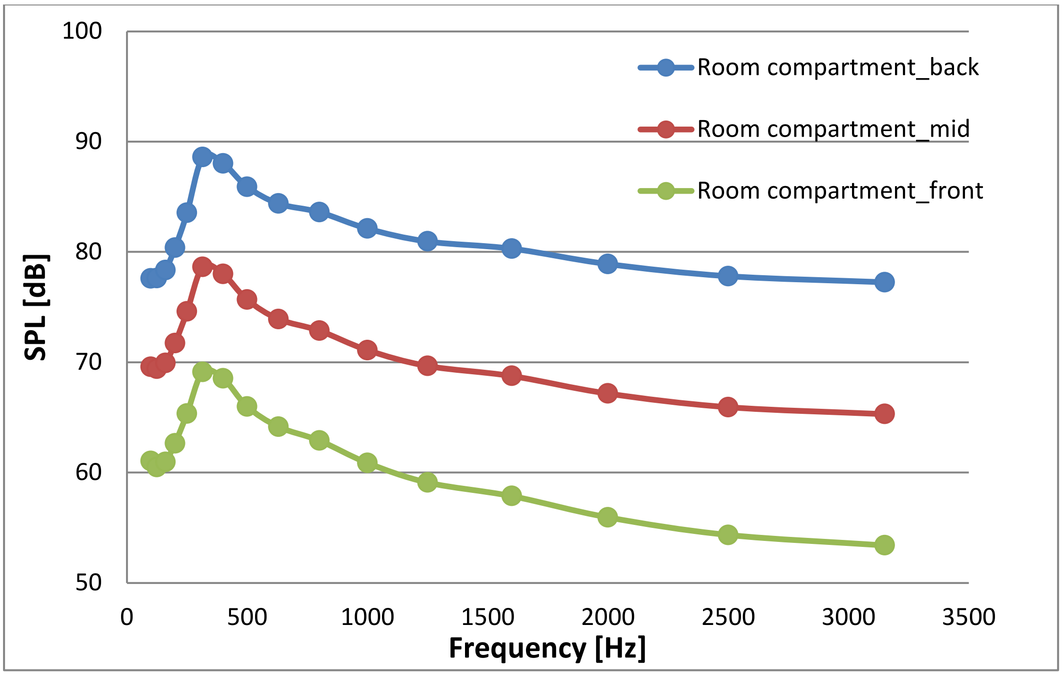

Figure 5-2 Interior noise contributions of the three compartments by excitation from the back compartment (ribbed case)

When the velocity profiles of the top plate for the two cases are examined in Figure 5-3 and Figure 5-4, it is seen that there is no remarkably change in velocity values when the coupling effects are presence. The maximum obtained velocity values are the same for both cases and they are observed around 315 Hz.

The vibrational velocity curves of the top plates are expected to behave like a decreasing trend as the frequency increases. However, the values get higher around 1600 Hz. This unexpected occurrence for the analyses occurs when the fluid around the structure is introduced as water in the analyses. The fluid around the structure causes the magnitude of the mass of the structure to increase and the added mass makes the vibration velocity values descend. In this analysis, the effect of the added mass cannot be observed for the values around 1600 Hz. The reason for the case encountered might be related with the damping characteristic of the structure, or it also might happen due to the incapability of SEA method for this problem in FreeSEA software.

It can be concluded that little change occurs in the velocity values however the plate-beam coupling effect brings out observable decrease in the displacements of the top plates.

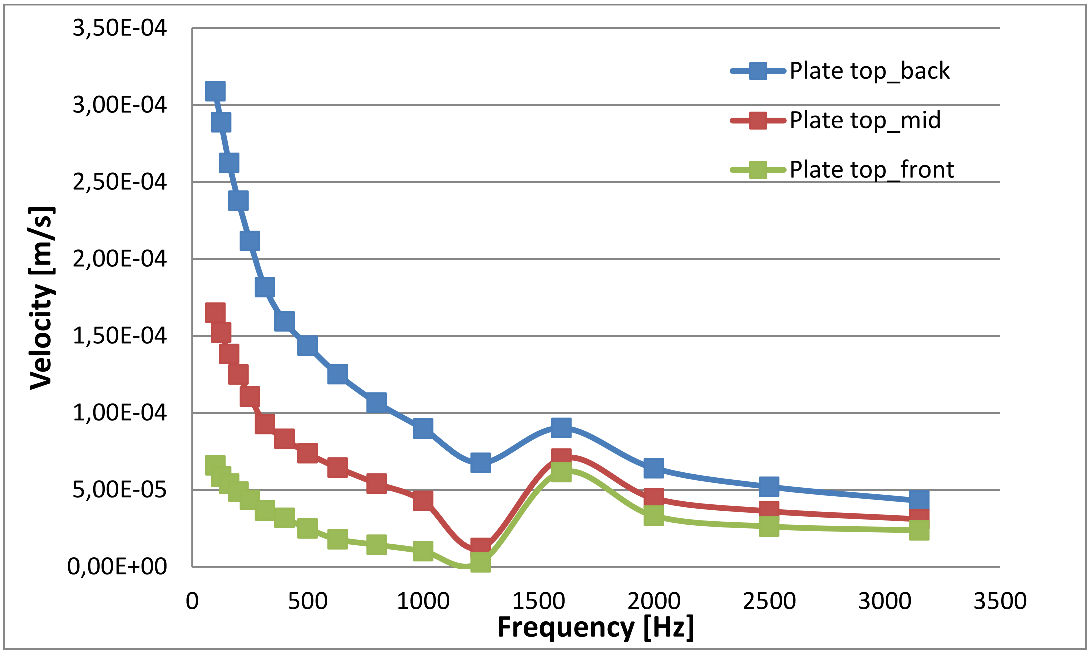

Figure 5-3 Velocity profiles of the upper plates of the three compartments by excitation from the back compartment (unribbed case)

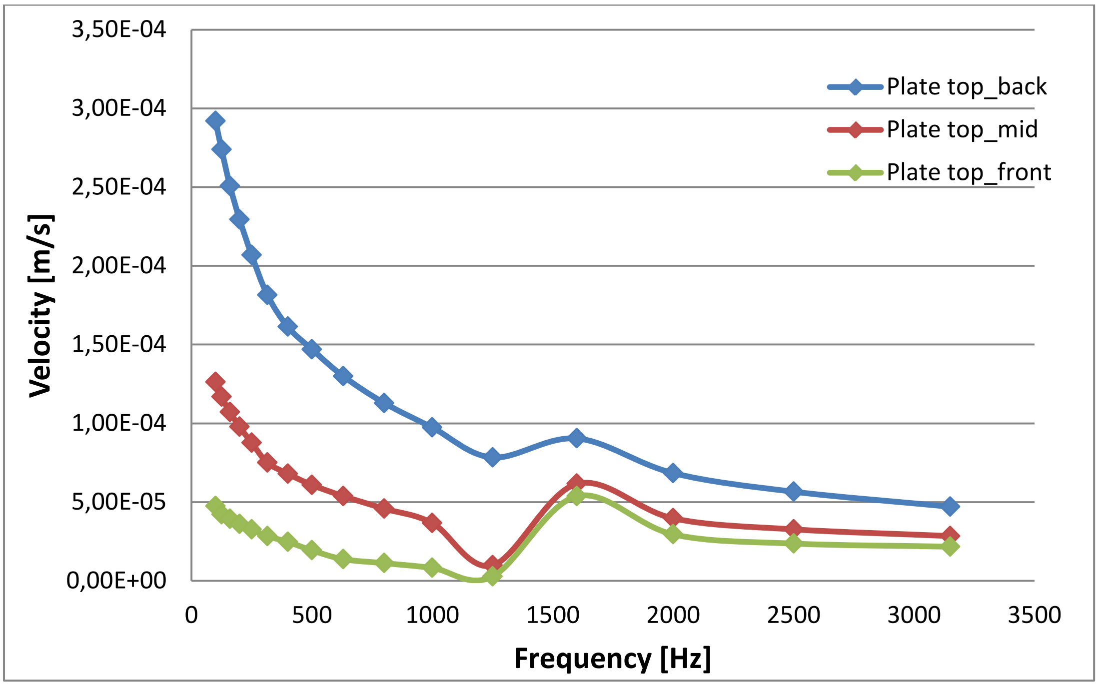

Figure 5-4 Velocity profiles of the upper plates of the three compartments with the excitation from the back compartment (ribbed case)

5.4 Analysis by VA One

In the previous chapter, the model development is explained. The plate and shell subsystems are obtained, the acoustic cavities are introduced, the surrounding semi-infinite fluid is modeled, all the connections between the subsystems are done and the modal densities of all the subsystems are presented. The damping loss factors of the subsystems are chosen as 1% in the analyses. At this point, the loading is applied to the developed model. The input excitations are engine noise and turbulent boundary layer noise. The analyses for the two excitations are conducted for each operating condition.

5.4.1 Engine Noise

In the study, the submarine model has an engine in the back compartment, in other words back compartment is engine room. It was not possible to find a submarine model with specified engine but the measured noise and vibration levels for a diesel engine appropriate for submarines are obtained. In Table 5-2, 1/1 octave band noise and vibration levels are provided. Noise level can be named as air-borne noise (ABN) whereas vibration level can be associated with structure-borne noise (SBN).

Table 5-2 Measured air-borne and structure-borne noise levels of the engine

1/1 Octave Noise Levels dB(A) re 20 µPa		**1/1 Octave Vibration Levels dB(A) re 1 µG**	
Frequency (Hz)	**Noise Level dB(A)**	**Frequency (Hz)**	**Vibration Level dB(A)**
31.5	109	31.5	95
63	90	63	114
125	100	125	115
250	101	250	109
500	101	500	119
1000	103	1000	118
2000	102	2000	119
4000	97	4000	120
8000	93	8000	117

The data is given in 1/1 octave band and the alteration gives insufficient information therefore the measured ABN and SBN levels are not turned into 1/3 octave band levels. The SBN levels are directly used in the analyses, meaning that no transfer function is implied in the foundation of the engine. This is because of the lack information about the diesel engine used in the model.

In VA One, the ABN levels are introduced into the acoustic cavity subsystem of the back compartments with "Acoustic Diffuse Field" function of the software product. The SBN levels are entered into the system as vibrational power levels for the specified frequencies.

5.4.2 TBL Noise

Turbulent boundary layer noise is an important noise source due to change in profiles may bring out high excitation levels on the surface of the body while fluid flows over the structure at high velocity levels. The motion of the submarine through the water makes the outer surface of the submarine subjected to turbulent pressure spectrum. This pressure is dependent on the dynamics of pressure and the moving speed of the structure within the medium.

The TBL noise generated by the fluid flow over the surface of the compartments and conical shells is calculated by the software. The TBL is applied to all the external SEA elements of the model. The software utilizes an empirical model in order to define the fluctuating pressure spectrum. In the application of the loading, the selection of the flow characteristics in terms of attached or separated should be done.

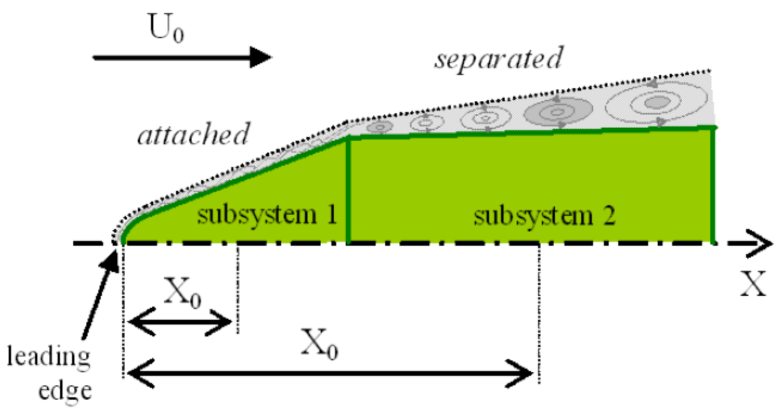

Figure 5-5 TBL characterizations: attached or separated [37]

The parameters used in calculating the TBL can be defined as:

- free stream flow velocity, U_0
- fluid density, ρ_0
- fluid kinematic viscosity, υ
- fluid speed of sound, c_f
- distance from the leading edge of turbulent boundary layer to the center of pressure load on the surface of the subsystem, X_0 [37].

The distance from the leading edge, X_0 is the distance from the leading edge of the boundary layer to the center of the pressure load on the surface. The distances for the subsystems used in TBL calculation can be found in Figure 5-6.

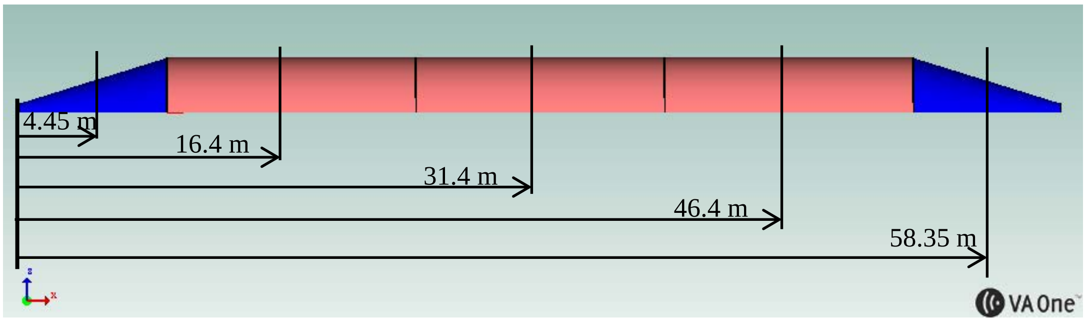

Figure 5-6 TBL distances for the model

In the analyses, the separated TBL field is used because the separated TBL field gives higher pressure levels than the attached TBL field.

5.4.3 Results by VA One

The analyses are performed for the first and the second conditions with different speed and depth configuration of the submarine. The estimations sound levels inside the compartments are provided in A-weighted sound levels and can be seen in the following figures.

The first analysis includes TBL noise with speed 5 m/s and 10 m/s in a depth of 5 m. The results are presented between 31.5 Hz and 8000 Hz, which is the frequency band for the given input, in 1/1 octave band center frequencies. In the second analysis, engine ABN noise and TBL noise are applied to the structure which is moving with speeds of 5 m/s and 10 m/s in a depth of 5 m.

In Figure 5-7, the TBL noise contribution is analyzed. In terms of the magnitude, the TBL noise results in low noise levels within the compartments. When the results of the sound levels within the compartments are examined, at low frequencies the values are close to each other. For mid and high frequencies the values within the conical compartments are getting higher compared to cylindrical shell compartments. At some frequencies distinct peak values can be observed for the compartments with conical shapes. In addition, above 1000 Hz, the front conical compartment is the noisiest region and has higher values for the vibration velocity for the submarine model. This is related with the flow direction of the structure (Figure 5-8).

Once the velocity curves of the interior elements are examined, it is surprising that all bulkheads show nearly the same characteristics. The fourth bulkhead, which is located at the region between the back compartment and back conical shell, has different vibration velocity trends for 5 m/s and 10 m/s (Figure 5-9 and Figure 5-15). At high frequencies, the magnitude of the vibration velocity of the fourth bulkhead is the bigger than the others. This occurs because the connected subsystems of the fourth bulkhead vibrate more at higher frequencies. The vibration characteristic of the bulkhead is much related with the connected subsystems.

As the submarine goes faster, the sound levels are getting observably higher values and the velocity levels also increase. It can be noted that the sound levels increase nearly 15 dB at the speed of 10 m/s for all the frequency range.

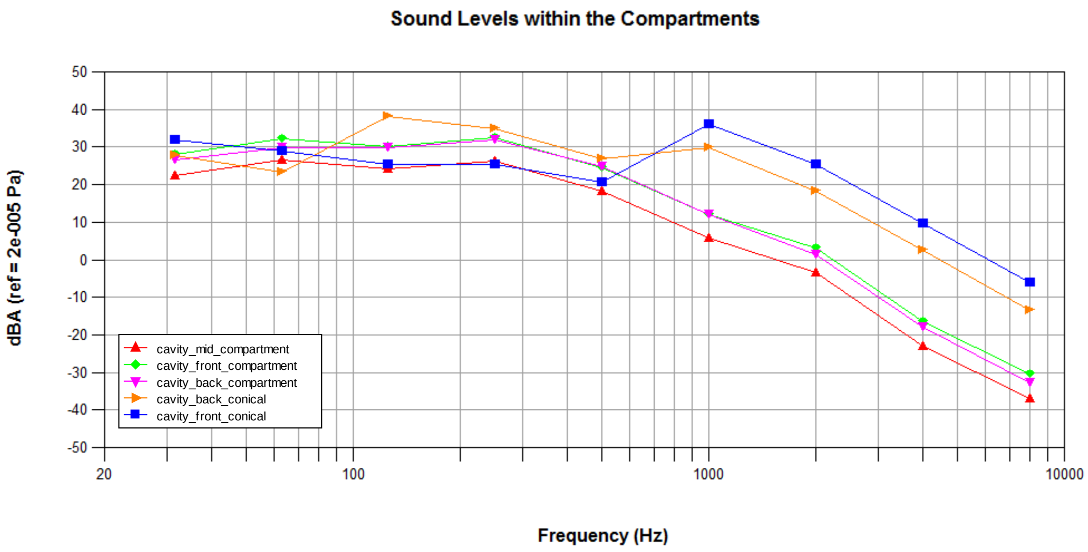

Figure 5-7 SEA Estimation for the sound levels within the compartments (First Condition (TBL noise) – 5 m/s & 5 m)

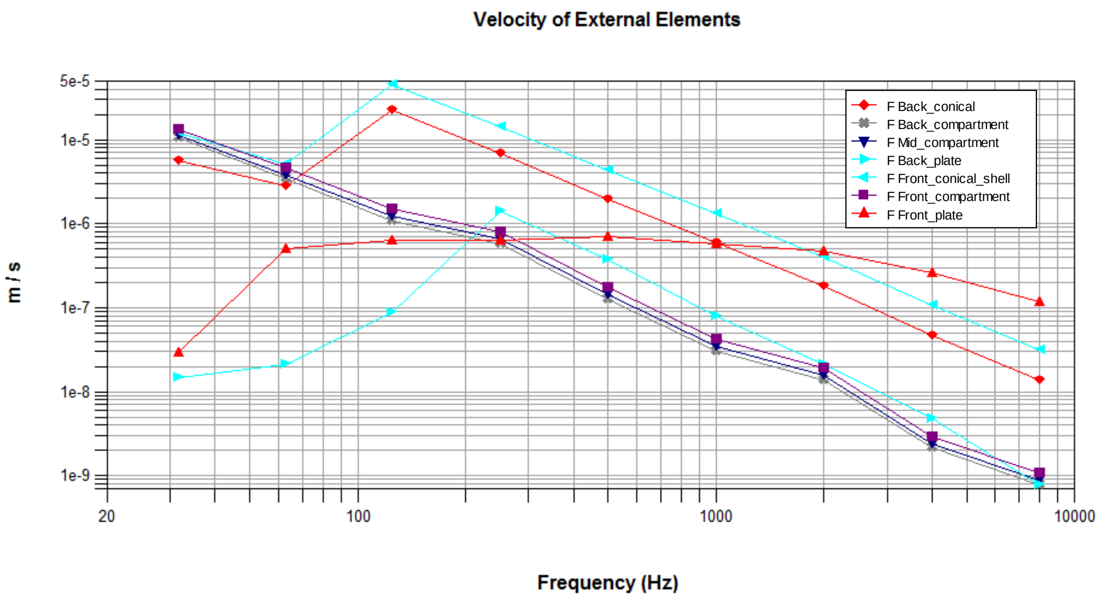

Figure 5-8 SEA Estimation for the velocity levels of the external elements (First Condition (TBL noise) – 5 m/s & 5 m)

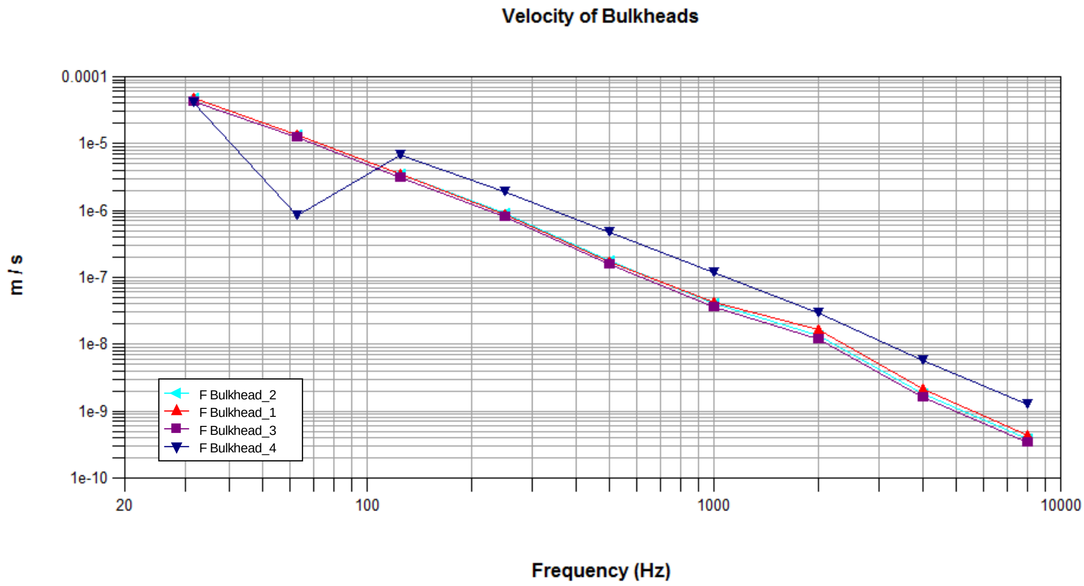

Figure 5-9 SEA Estimation for the velocity levels of the bulkheads (internal elements) (First Condition (TBL noise) – 5 m/s & 5 m)

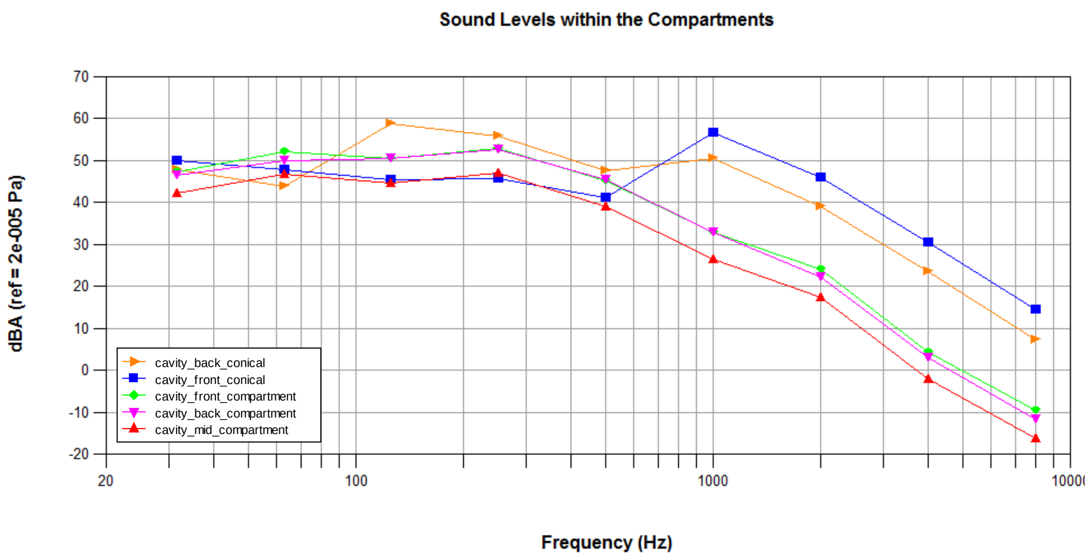

Figure 5-10 SEA Estimation for the sound levels within the compartments (First Condition (TBL noise) – 10 m/s & 5 m)

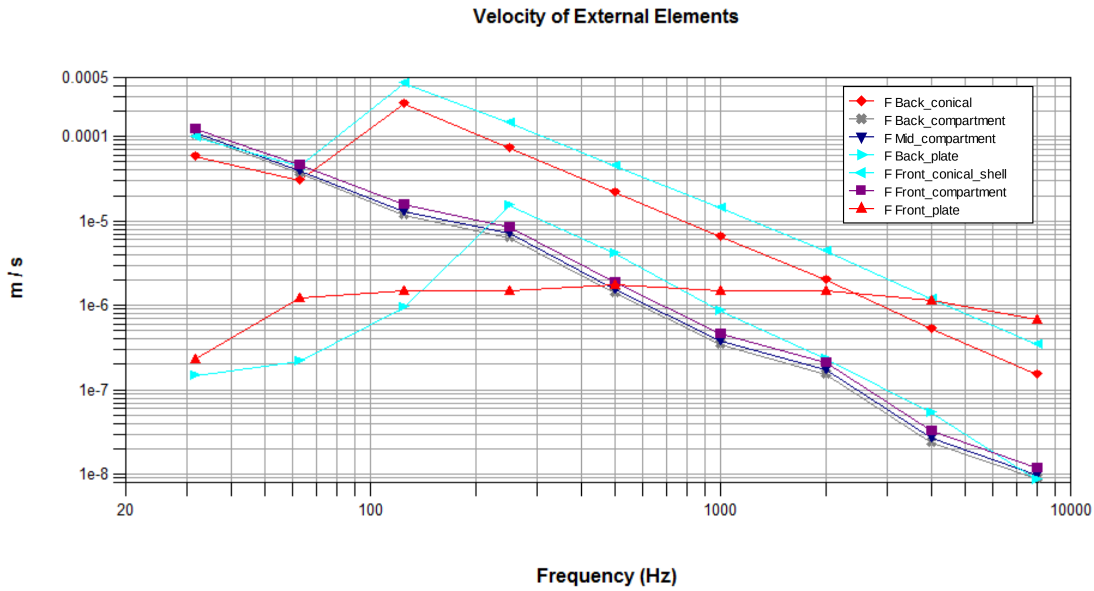

Figure 5-11 SEA Estimation for the velocity levels of the external elements (First Condition (TBL noise) – 10 m/s & 5 m)

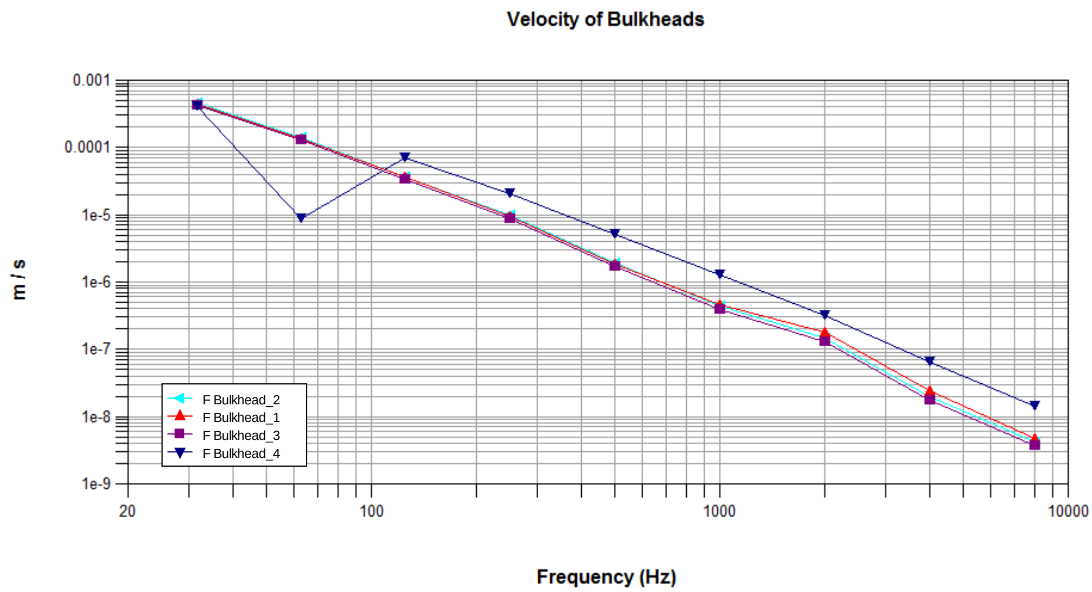

Figure 5-12 SEA Estimation for the velocity levels of the bulkheads (internal elements) (First Condition (TBL noise) – 10 m/s & 5 m)

In second analyses, the air-borne noise levels of the machine are added. It is obvious that the sound levels of the back compartment increase appreciably. Moreover, the values for the sound pressure levels within the compartment increase visibly (Figure 5-13). The curves of the sound levels within the compartments are parallel to each other and are arranged in descending order, from back compartment to front compartment.

As the frequency increases, the noise and vibration velocity levels are in a descending behavior. The curves are parallel to each other and settled separately. For high frequencies, the sound levels of the compartments and velocity values of the external elements are getting closer (Figure 5-13 and Figure 5-14). The velocity profiles of the bulkheads can be grouped among themselves. The bulkheads located at the back side of the structure have higher values than the ones located at the front side of the structure. They have the same slope for the vibration velocity curves in Figure 5-15.

For mid and high frequencies, the compartments except the back compartment at the speed of 10 m/s have nearly 2 dB higher values than the values related with the speed of 5 m/s (Figure 5-16). When the engine ABN is introduced to the submarine, the effects of the change in the speed due to TBL excitation are not noticeable (Figure 5-16, Figure 5-17 and Figure 5-18).

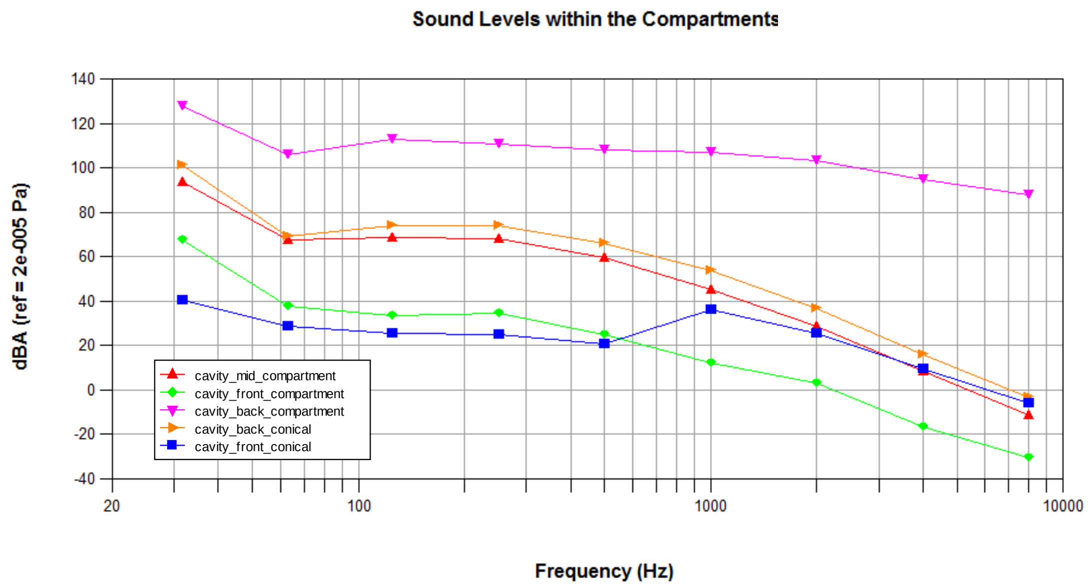

Figure 5-13 SEA Estimation for the sound levels within the compartments (Second Condition (ABN & TBL noise) – 5 m/s & 5 m)

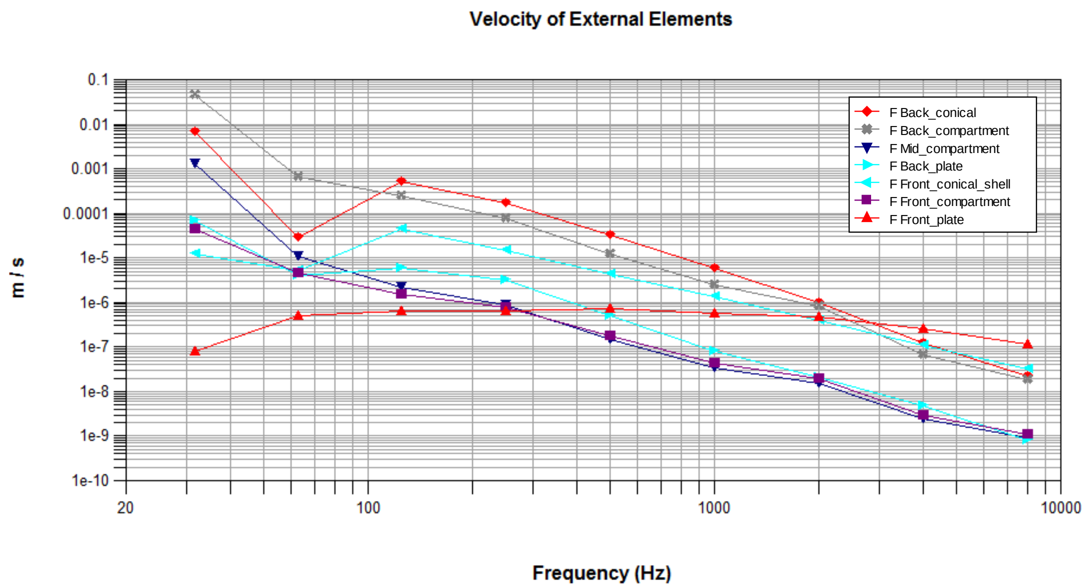

Figure 5-14 SEA Estimation for the velocity levels of the external elements (Second Condition (ABN & TBL noise) – 5 m/s & 5 m)

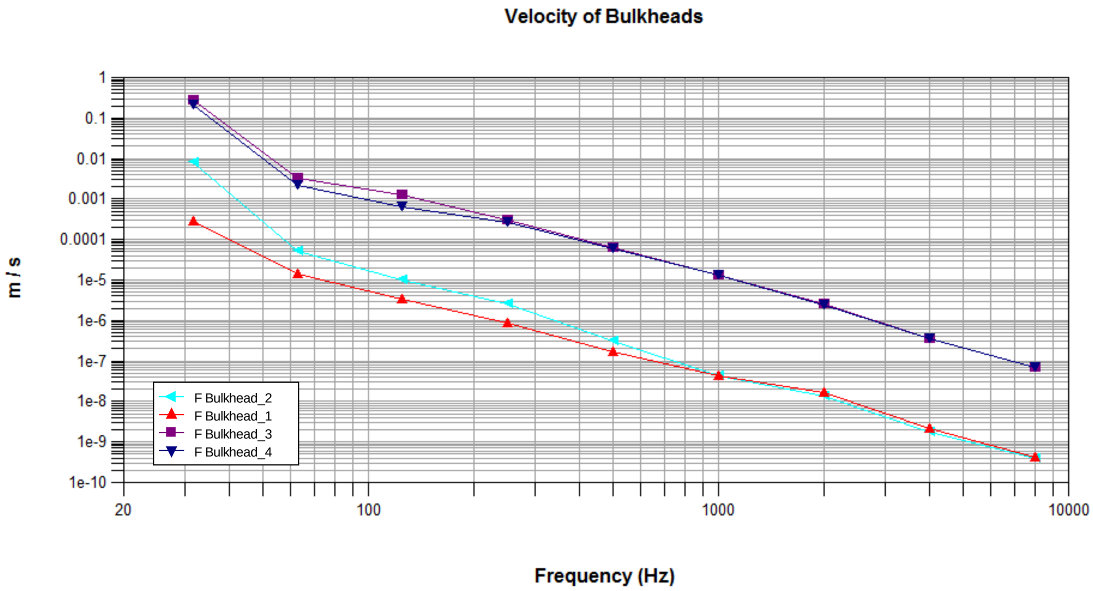

Figure 5-15 SEA Estimation for the velocity levels of the bulkheads (internal elements) (Second Condition (ABN & TBL noise) – 5 m/s & 5 m)

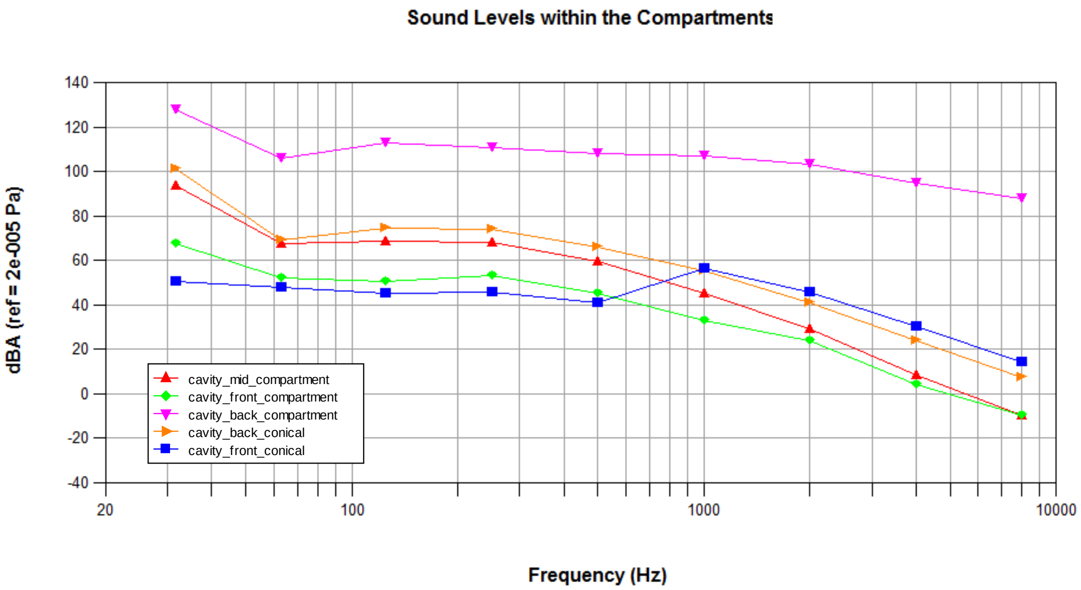

Figure 5-16 SEA Estimation for the sound levels within the compartments (Second Condition (ABN & TBL noise) – 10 m/s & 5 m)

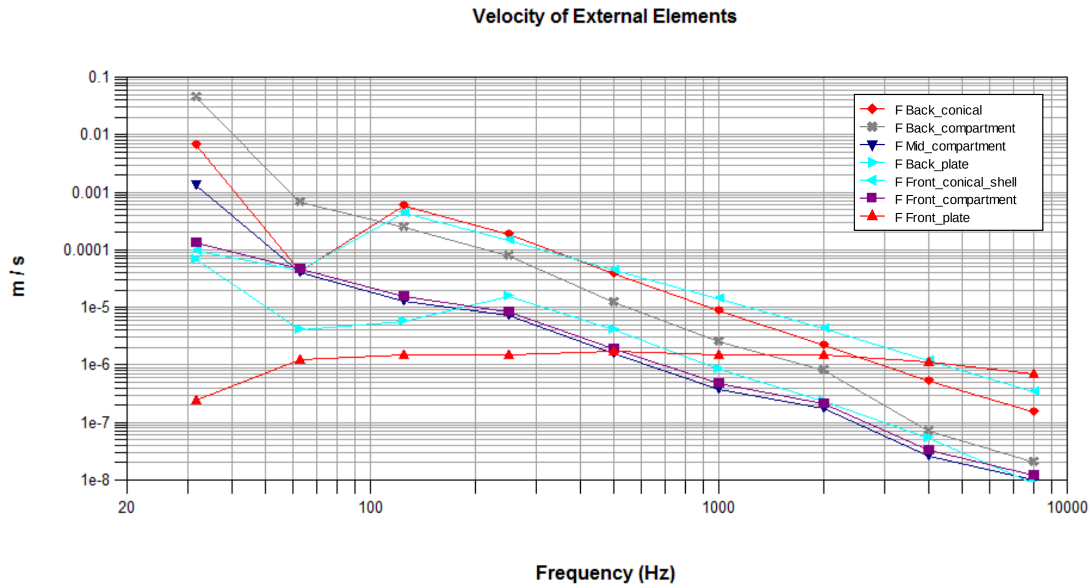

Figure 5-17 SEA Estimation for the velocity levels of the external elements (Second Condition (ABN & TBL noise) – 10 m/s & 5 m)

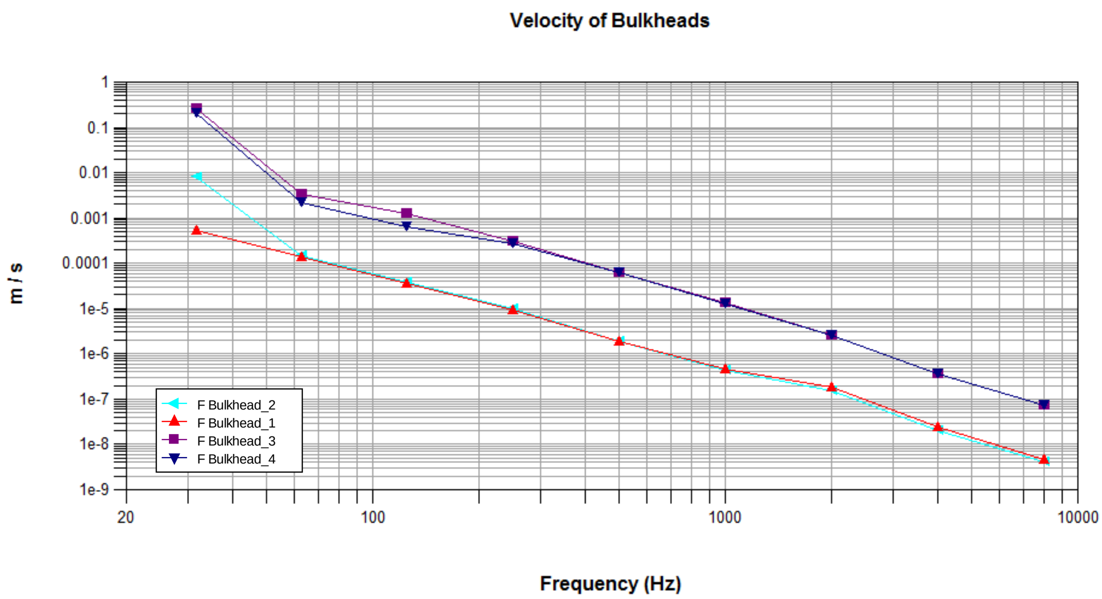

Figure 5-18 SEA Estimation for the velocity levels of the bulkheads (internal elements) (Second Condition (ABN & TBL noise) – 10 m/s & 5 m)

For the third operating condition, a plate subsystem is added for the foundation of the engine and the back compartment is divided into two parts. Upper and lower parts of the back compartment are modeled as curved shell subsystem. The vibration input of the engine is introduced into the system at the plate subsystem where the engine is placed. In this study, the focus is on the effects of the loadings on cylindrical shell modeled submarine; hence the details of the submarine compartments, engine foundation are not included in the analyses.

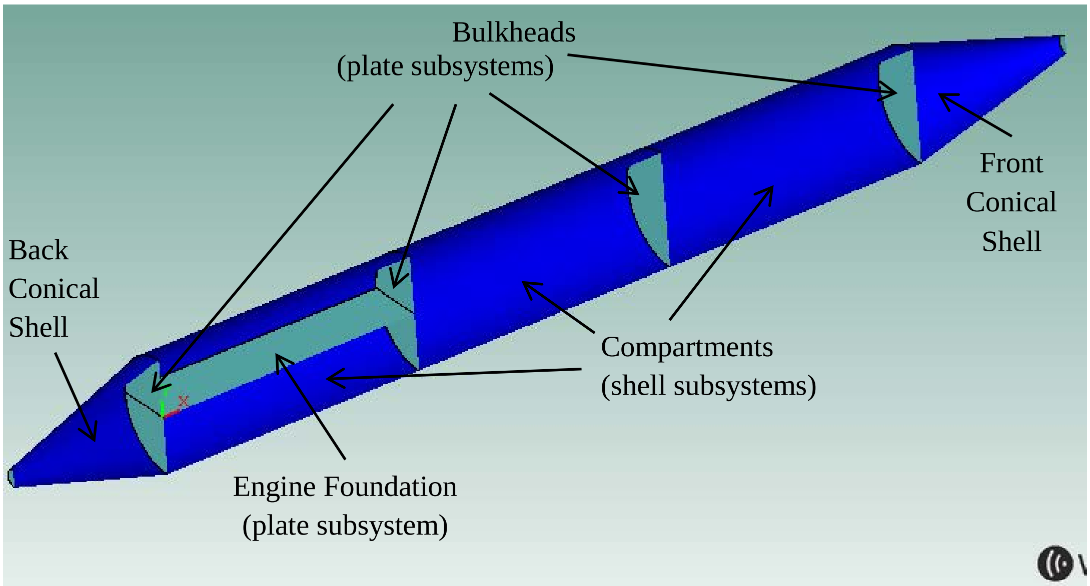

Figure 5-19 SEA model with engine foundation

In the new SEA model, 15 subsystems, 6 acoustic cavity subsystems, 8 point junctions, 9 line junctions and 15 area junctions are generated. The new modal densities are provided in order to observe the effect of the engine's plate.

It can be observed that the modal densities of the new model are similar to the previous ones in Figure 5-20, Figure 5-21 and Figure 5-22. The cylindrical shells have the highest modal density values. The modal density curves of the conical shells converge to the cylindrical curves as the frequency increases. At low frequencies, the modal density values of the conical shells are close to the front and back plates. Due to dependency of the modal density of a subsystem on geometry, thickness and material, the half cylindrical shells have smaller values for modal density than the cylindrical shells.

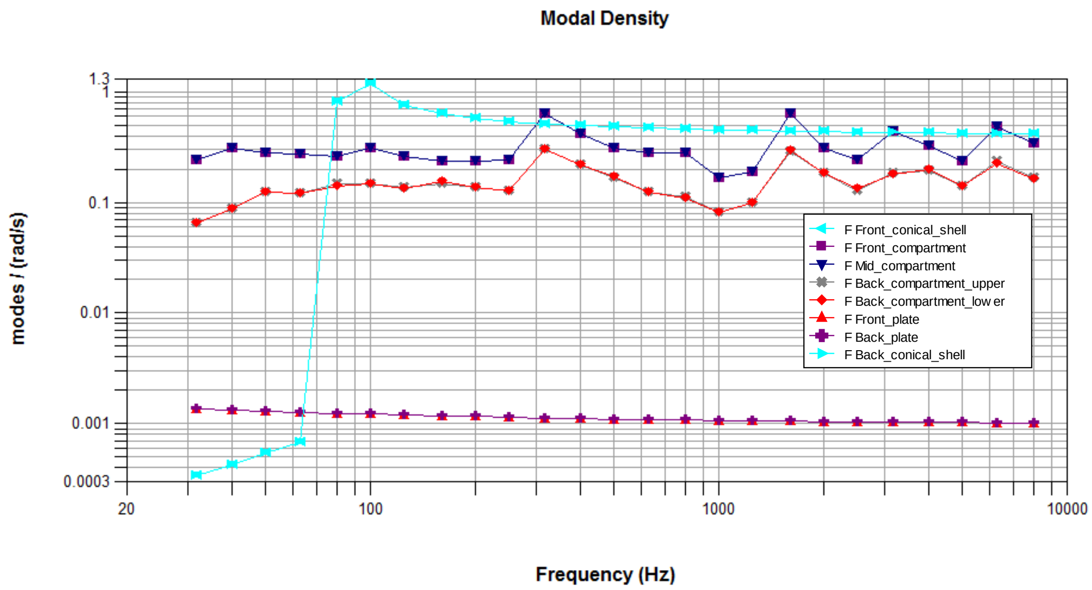

Figure 5-20 Modal densities of the external elements in the new model

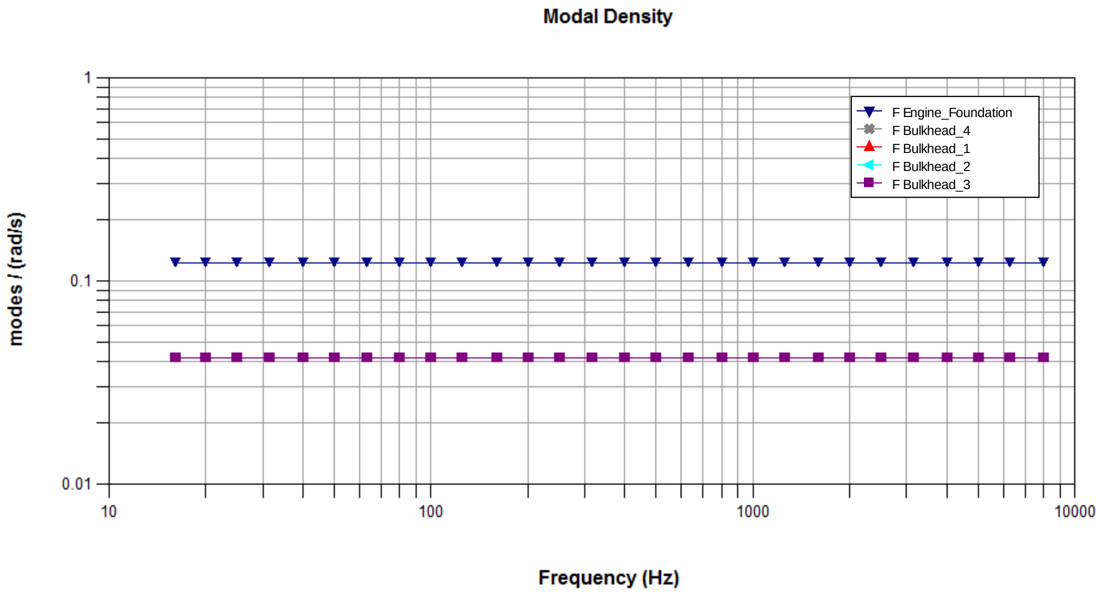

Figure 5-21 Modal densities of the bulkheads and the engine plate in the new model

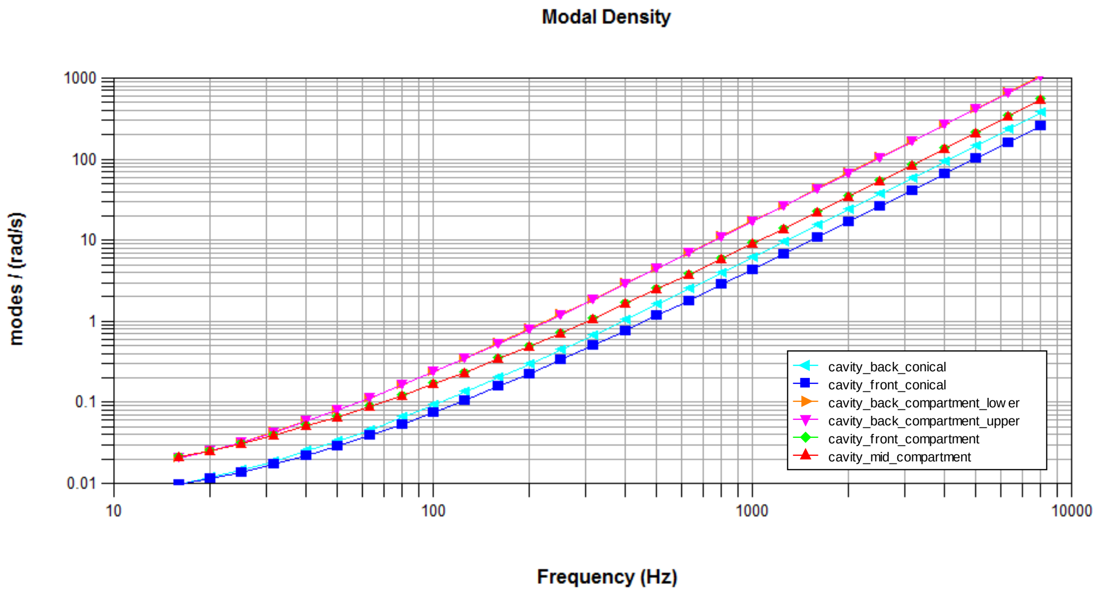

Figure 5-22 Modal densities of the acoustic cavities in the new model

The analyses with the new model are performed in the same manner. The estimations of sound levels in the compartments and the velocity of the subsystems are presented in the following figures.

The behavior of the curves of the third condition shows similarity with the ones of the second condition. There are similarities in the behavior but also the differences in the noise levels within the compartments (Figure 5-23). The sound levels in the front sections decrease for the third condition. It is unexpected that the back conical shell has lower sound values at mid frequencies for the third condition. This might happen due to the mistake in the connections of the model. As the number of subsystems increases, the connections of the subsystems cannot be achieved successfully in the developed model.

The compartments of the velocity values are increased, on the contrary the front and back plates’ velocity values are decreased. Their coupling effect is observed in the velocity graphs (Figure 5-24).

In Figure 5-25, the velocity values for the engine foundation are higher than the bulkheads. The curves are parallel to each other and as the frequency increases the values get closer.

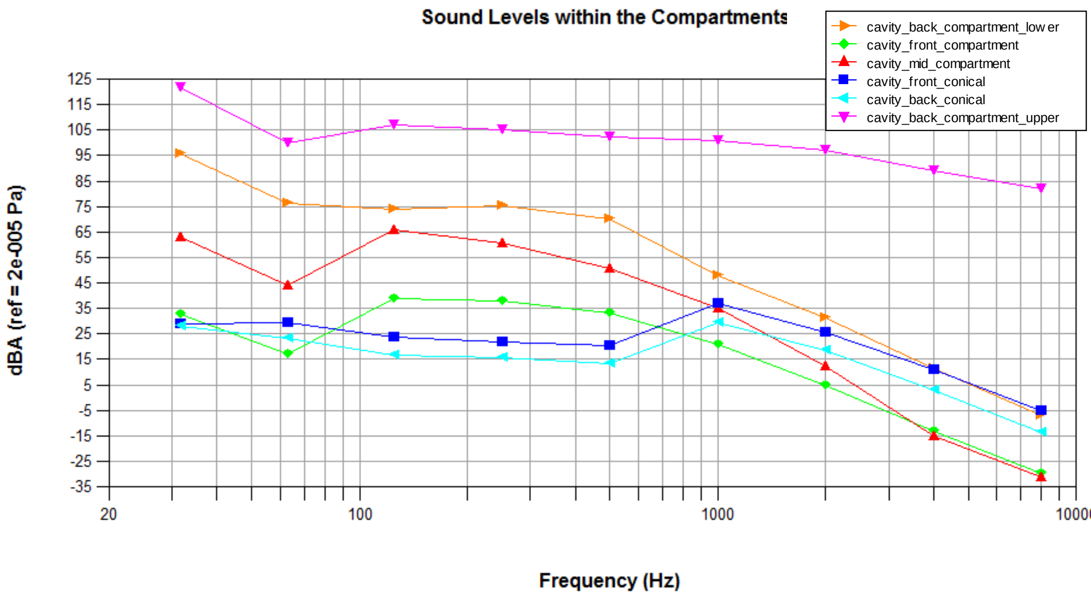

Figure 5-23 SEA Estimation for the sound levels within the compartments (Third Condition (ABN, SBN & TBL noise) – 5 m/s & 5 m)

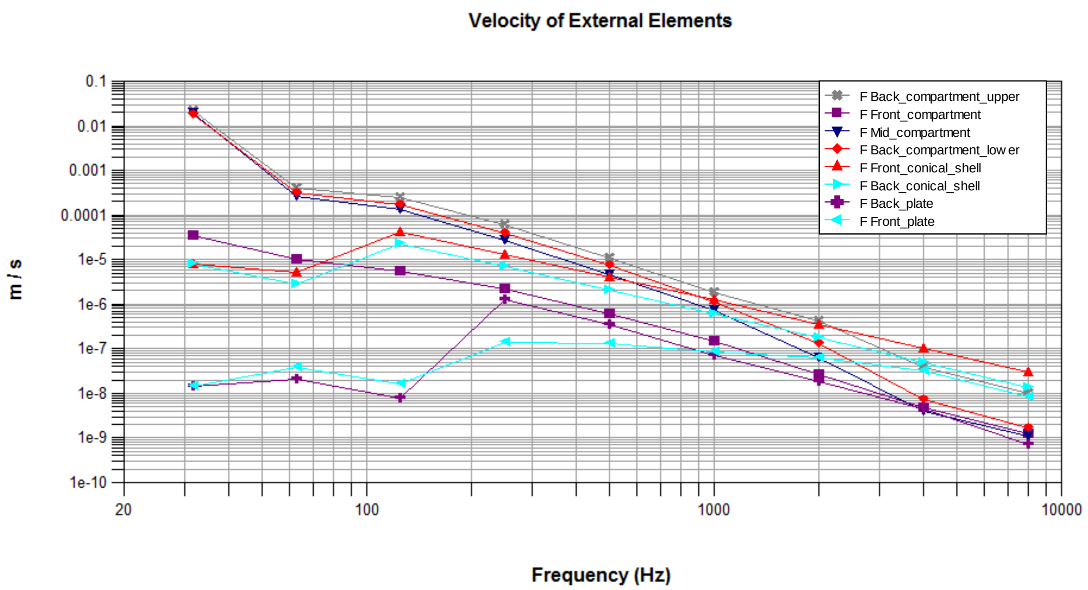

Figure 5-24 SEA Estimation for the velocity levels of the external elements (Third Condition (ABN, SBN & TBL noise) – 5 m/s & 5 m)

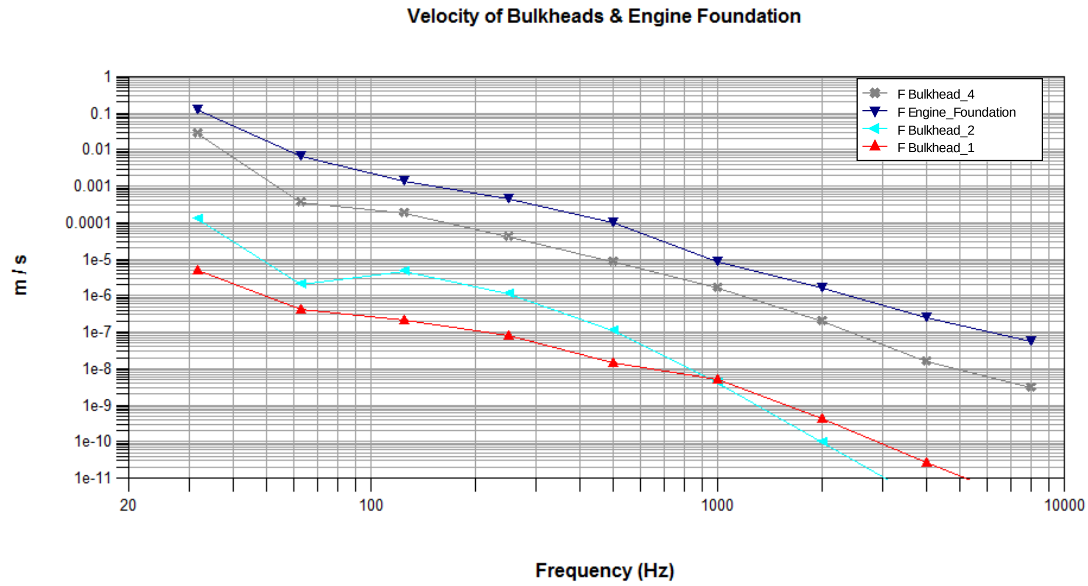

Figure 5-25 SEA Estimation for the velocity levels of the bulkheads and the engine foundation (Third Condition (ABN, SBN & TBL noise) – 5 m/s & 5 m)

In the second part of the third condition analysis, the estimations of sound levels within the compartments and velocity levels of the elements of the structure are done for 10 m/s excited by the engine and TBL excitations. When compared with the results of the analysis with 5 m/s, the increased speed of the submarine brings in high sound levels for the compartments at mid and high frequency ranges. Especially, for front and back conical shells, the sound levels are increased in sensible grade above 1000 Hz (Figure 5-26). The similar occurrences are observed for the velocity curves of the external elements and internal elements in Figure 5-27 and Figure 5-28. TBL noise contribution meaning change in the speed can easily be investigated for high frequency ranges. The values for the front section of the submarine are increased more than the ones for the back section. The velocity values for the engine foundation are higher than those for bulkheads. At higher frequencies, the values get closer for the bulkhead of the back compartment and the engine foundation.

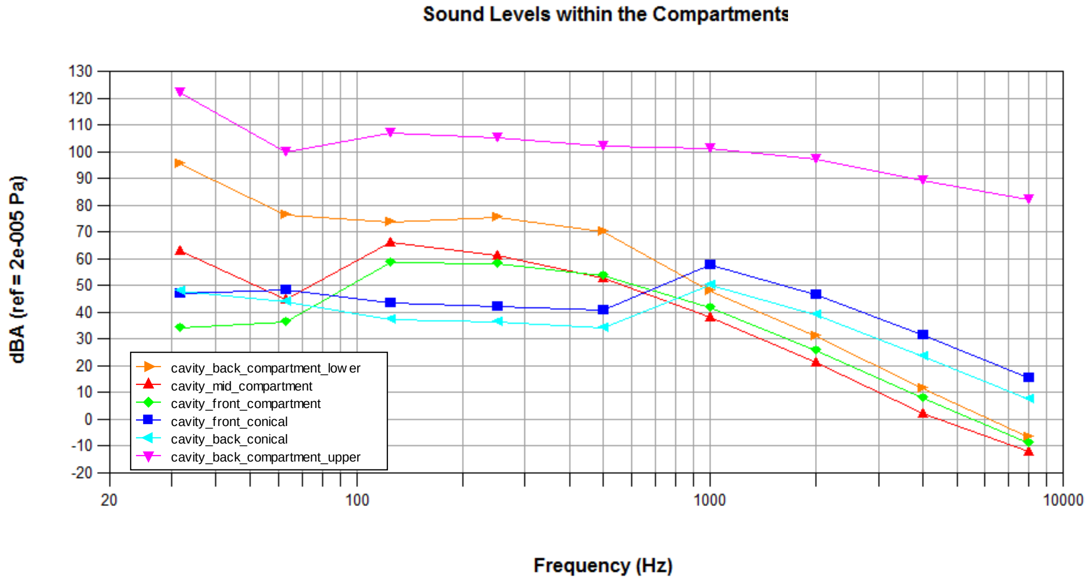

Figure 5-26 SEA Estimation for the sound levels within the compartments (Third Condition (ABN, SBN & TBL noise) – 10 m/s & 5 m)

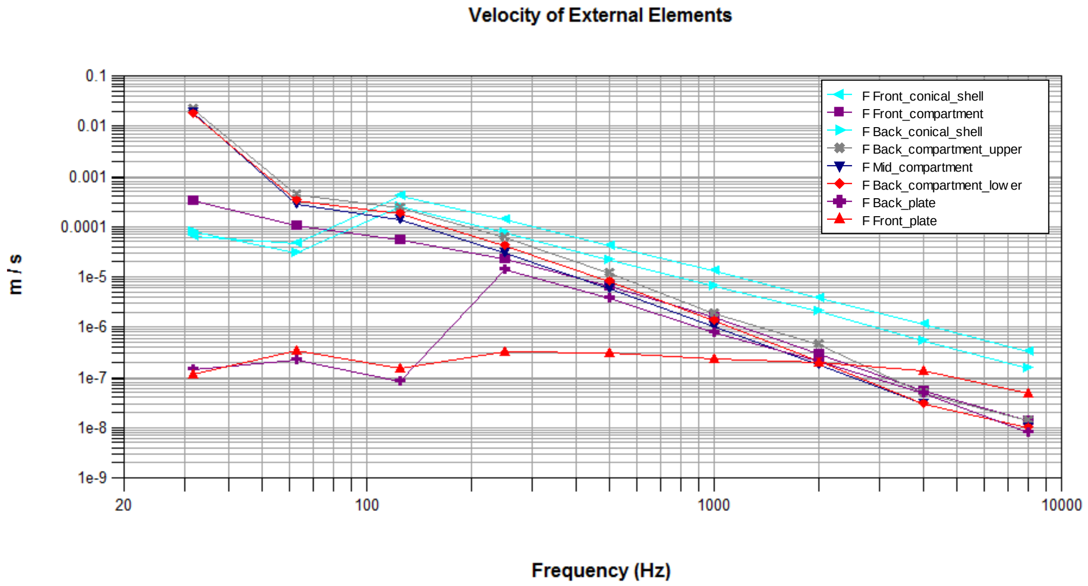

Figure 5-27 SEA Estimation for the velocity levels of the external elements (Third Condition (ABN, SBN & TBL noise) – 10 m/s & 5 m)

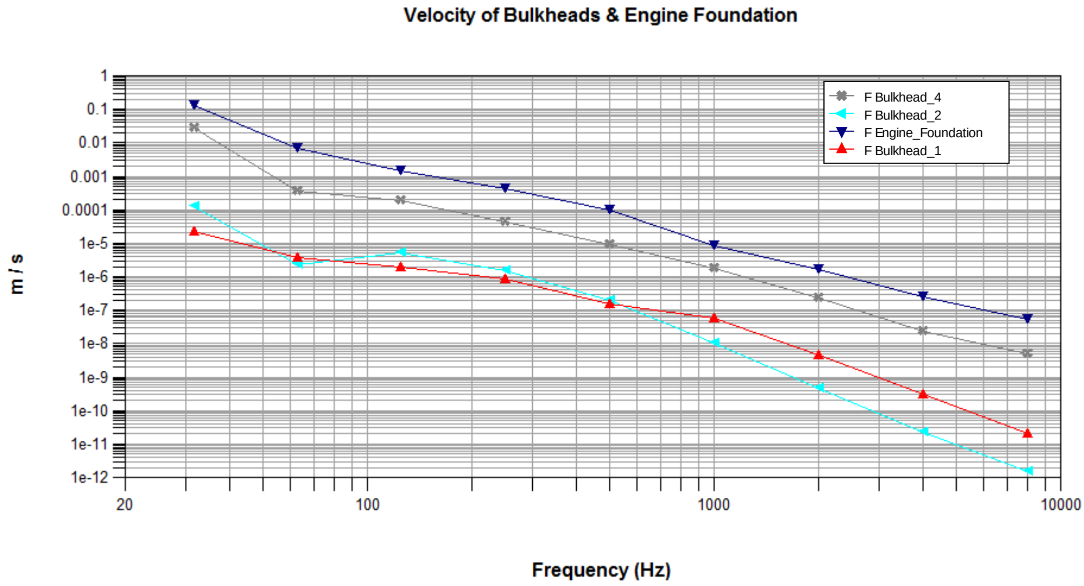

Figure 5-28 SEA Estimation for the velocity levels of the bulkheads and the engine foundation (Third Condition (ABN, SBN & TBL noise) – 10 m/s & 5 m)

CHAPTER 6

CONCLUSIONS AND FUTURE WORK

6.1 Summary and Conclusions

In this study, the application of SEA to the prediction of air-borne and structure-borne noise in a submerged structure is investigated. In order to fairly present the analysis, the surrounding elements and the inputs are carefully investigated.

On the FreeSEA, the submarine hull is modeled as a rectangular box with its ring stiffeners, bulkheads and end caps. Plate elements are the subsystems of the model and plate-plate and plate-beam couplings are assumed in the analysis. The ring stiffeners are simulated as beam elements and the coupling of the beam and the plate elements are applied into the system. FreeSEA software, which involves computer codes solving SEA problems, is used in the analysis. In the program, the physical and geometric properties of the system and the subsystems, the coupling properties of the junctions are provided as input. Then, the sound pressure levels of the three compartments and the displacements of the top plates are obtained from the analysis.

Sound pressure levels are obtained for the three compartments. The air-borne noise levels for the interior of the submarine are presented in the results section. It can be concluded that the sound due to the machinery compartment is transmitted through the bulkheads and the structure. The presence of the beams coupled with plates show pass characteristics for the transmission of the sound in the structure.

For cases of the plate elements coupled with and without beams, the results are comparable. In the case of the plates without beams, meaning unribbed case, the values of the velocity profiles of the upper plates of the model are larger than those in the case of the plates with beams. This is an expected and intended result stemming from the addition of the stiffeners to the system. The stiffeners again bring rigidity in the plates, to which they are attached.

When the fluid loading is applied to the analyses, there occurs a jump around 1600 Hz in the results for vibrations of the plates. This is an unexpected behavior for the vibrations of the plates. The vibrational velocity values of the top plates are expected to decrease drastically as the frequency due to the added mass (apparent mass or mass loading) effect on the structure.

In the analysis of FreeSEA, the coupling loss factors between plates and beams are calculated with the presented computer code. However, in order to get the real coupling values, measurements should be done on the structure. In addition, a hybrid model can be introduced to get the coupling loss factors.

The second model covers cylindrical and conical shells and SEA estimations are provided for the three afore mentioned configurations in Chapter 5. In order to observe the different loading conditions, controlled numerical experiments are designed where noise sources are applied separately and together. The analyses are performed on VA One software and the sound levels in the compartments and the vibration velocity levels of the structure subsystems are obtained.

In general, it can be noted that the effects of the axisymmetricity of the structure can be seen appreciably in the analyses. The front and back compartments reveal similar characteristics in terms of noise and vibration.

By examining the figures and results, the TBL noise brings in low sound levels in the cavities. The movement of the submarine affects the vibration and noise characteristics of the structure. When the submarine goes faster with a speed 10 m/s, the sound levels are getting remarkably higher and the velocity levels rise appreciably as well. The trends of the curves of the velocity and sound levels do not change. This is an expected result for the turbulent boundary layer excitation. At very low frequencies, below 63 Hz, the sound levels of the compartments are close to each other. The air-borne noise levels in the conical shells are in an unpredictable form and they have the highest noise levels. Distinct peak values for the conical shells are observed at some frequencies. These are due to the geometry of the subsystem and the thickness of the structure. It is evident at high frequencies that the front conical shell is the noisier compartment than the back conical. This is related with the direction of the flow. Moreover, the compartment in the middle is the quietest compartment and this is an unforeseen result. The explanation of this result is the axisymmetricity of the structure.

The addition of air-borne noise of the engine into the system at the back compartment cavity yields different results than the results of the first condition. The noise levels in the compartments increase visibly and the levels get lower as moving away from the back compartment where the noise originates from. This occurrence can be observable in the sound levels of the compartments and velocity levels of the subsystems. As the frequency increases, the noise and velocity levels tend to be in a descending behavior. The sharp changes or jump values cannot be observed with the contribution of the engine noise. When the engine ABN is introduced to the submarine, the effects of the change in the speed are not noticeable. The acceleration of the submarine makes the noise level higher in the front conical cavity and front compartment at the high frequencies, above 1000 Hz. This is also valid for the velocity values of the external subsystems and the bulkheads. This tendency can be attributed to the high frequency spectral content of TBL excitation. The values are ascending in an adequate amount for the high frequencies. The velocity curves of the front and back bulkheads look parallel to each other. Their slopes seem to be the same. It can be concluded that the ABN noise is dominant so the vibration characteristics of the structure is not remarkable.

A new model is constructed in order to apply the vibration levels of the engine. There are similarities in the trends of the sound level curves, but noise level differences between the compartments are observed. The sound levels of the compartments are not affected remarkably by the structural excitation. The structural excitation mainly results in vibration in the elements of the submerged structure. The velocity behavior of the compartments with respect to frequency is the same for all of them and also the conical shells show the same behavior because of the similarity in geometry. The velocity values of the compartments are increased; on the contrary, the front and back plates' velocity values are decreased for low frequency ranges. Their coupling effect is observed in the velocity graphs. In this low frequency range, the developed model cannot yield expected results due to the applicability of SEA to the higher frequency ranges. The vibration velocity values for the engine foundation are higher than the bulkheads. At higher frequencies, these values get closer for the bulkhead of the back compartment and the engine foundation.

It is observed that the noise levels in the compartments increase when the submarine goes with a speed of 10 m/s when compared with the ones at a speed of 5 m/s. In particular, except the engine room, high sound levels are obtained in all the compartments. The velocity levels for the external elements are not affected with the increased speed as well as the velocity of the bulkheads and the engine foundation.

The analyses conducted in VA One software also include the secondary structure-borne noise effects in the results of the sound levels within the compartments. The transmission of vibration from the engine to the structure occurs directly. Moreover, the transmitted sound from the engine to the air inside the compartments moves again through the structure and the bulkheads. This occurrence is named as secondary structure-borne noise. The secondary structure-borne noise levels are included in the sound levels within the compartments and vibration levels of the internal elements of the structure.

The obtained results should be checked through measurements if the estimated sound levels in the compartments are acceptable. There are regulations to limit noise levels in the compartments. Submarine crew shall not be exposed to continuous noise over limiting levels. The engine room is the most critical place for the submarine crew.

Two different approaches are provided with this study for the submarine noise and vibration problem. The first approach can be used to have an idea of behavior of a basic structure. With VA One, complex geometries can be solved and all the properties of the structure and the details of the input loadings can be introduced into the calculations. VA One provides users with complete solution for the vibro-acoustic problems.

Sound transmission analysis of structure-borne noise is rigorous task. On this account the analysis is restricted to the specific area and the application of SEA is undertaken through the limited area. The identification of transmission paths is also complex problem because reflection and absorption always occurs in the structure. The results of the study can be used for predictions of the real case situation with measurements done and then comparison can be made through this analysis. This study can be regarded as the beginning of a detailed sound transmission analysis. Also, the noise transmission through the equipment foundations should be taken into account for a reliable analysis. The apparent stiffness and damping characteristics of the mounting system definitely affect the system response.

In order to apply SEA, the parameters for the analysis must be determined with the conducted experiments. The internal loss factor of each plate is taken as constant in the analyses carried in this study. It is one of the crucial factors for obtaining inaccurate sound levels. In the frequency range of interest, damping values should be measured and can be averaged for the frequency range or if possible for the program to implement as a function of the frequency.

In the light of this study, the following conclusions can be drawn:

1. Effect of TBL excitation on interior noise levels of the submerged structure depends on the speed of the submerged structure, as anticipated. When only TBL noise excitation on the structure is applied, lower sound pressure levels within the compartments are obtained. The effect of TBL originated noise can be seen notably for mid and high frequencies as the submerged structure is analyzed under TBL and engine noise excitations.
2. Results obtained applying engine noise and vibration levels show that air-borne noise is dominant and vibrational power effects of the engine can be observable in the results of vibration levels of the exterior elements of the submerged structure.
3. It can be concluded that high sound and vibration levels are obtained in this study due to the fact that low damping and low absorption values are used in the analyses. Another reason can be attributed to the direct introduction of sound and vibrational power levels into the structure. The damping and absorption factors are important parameters in aspects of noise and vibration transmission.
4. Noise transmission due to the machinery and hydrodynamics are not the only transmission mechanisms in submarines. As mentioned before, there are a lot of mechanisms that must be taken into account for the acoustic analysis of submarines. Propeller, engine exhaust, auxiliary machinery, generators, heating, ventilating and air-conditioning systems can be added into the analysis in order to simulate the submarine's case. After each of the mechanisms are identified and calculated or measured, superposition is imposed in order to get the total acoustic behavior of the structure. Some of these excitations cannot be truly investigated by SEA methods due to their low frequency tonal character.

6.2 Recommendations for Future Work

The following recommendations for future work can be listed:

1. Emphasis should be given to determine reliable loss factors for the analysis in order to get better accuracy from SEA application.
2. Sound transmission model should also include other noise sources and mechanisms (propeller, engine exhaust and auxiliary machinery, generators, heating, ventilating and air-conditioning systems).
3. Complexity of the submerged structure can be introduced in the analysis and the materials of the compartments and the hull of the submarine can be given in a manner to visualize the real case.
4. Analysis can be performed with different operating conditions and a determination of the critical operating conditions can be obtained.
5. Noise control treatments can be applied to the engine room and engine foundation in the analyses.

REFERENCES

[1] Ross D., *Mechanics of Underwater Noise*, Pergamon Press, Oxford, 1976.

[2] Carlton J.S., *Marine Propellers and Propulsion*, Butterworth-Heinemann, Oxford, Second Edition, 2007.

[3] Xavier Lurton, *An Introduction to Underwater Acoustics: Principles and Applications*. Springer, 2002.

[4] Feit D. and Arveson P., *Mechanics of Underwater Noise*, Applied Technology Instiute (ATI) Course Notes, May, 2011.

[5] Hodges C.H., Power J. and Woodhouse J., “The low frequency vibration of a ribbed cylinder, part 1: theory”, *Journal of Sound and Vibration*, 101 (2), pp. 219-235, 1985a.

[6] Hodges C.H., Power J. and Woodhouse J., “The low frequency vibration of a ribbed cylinder, part 2: observations and interpretation”, *Journal of Sound and Vibration*, 101 (2), pp. 219-235, 1985b.

[7] Leissa A., Vibration of Shells, *Acoustical Society of America*, New York, 1993.

[8] Solaroli G., Gu Z., Baz A. and Ruzzene M., “Wave propagation in periodic stiffened shells: spectral finite element modeling and experiments”, *Journal of Vibration and Control*, 9 (9), pp. 1057-1081, 2003.

[9] Sandman B.E., “Fluid-loading coefficients for a finite cylindrical shell”, *Journal of the Acoustical Society of America*, 111 (1), pt 1, pp 75-79, 1976.

[10] Liao C-J., Jiang W-K., Duan J. and Wang Y., "Vibrational and acoustic radiation from a submerged periodic cylindrical shell with axial stiffening", *Journal of Mechanical Engineering Science*, Proc. IMechE, Vol. 223 Part C., 2009.

[11] Liao C-J., Jiang W-K., Duan J. and Wang Y., "Vibration and sound radiation from submerged finite cylindrical shells reinforced with axially periodic stiffeners", *Journal of Vibration and Control*, 17 (10), pp. 1472-1480, 2011.

[12] Caresta M., Kessissoglou N., "Acoustic signature of a submarine hull", *14th International Congress on Sound and Vibration*, 9-12 July 2007.

[13] Caresta M., Kessissoglou N., "Acoustic signature of a submarine hull under harmonic excitation", *Applied Acoustics*, 71, pp. 17-31, 2010.

[14] Burroughs C. B., Hallander J. E., "Acoustic radiation from fluid-loaded, ribbed cylindrical shells excited by different types of concentrated mechanical drives", *Applied Research Laboratory, Penn State University*, Technical Memorandum, 90-320, 1991.

[15] Ai S., Sun L., "Fluid-structure coupled analysis of underwater cylindrical shells", *Journal of Marine Science and Application,* 7 (2), pp. 77-81, 2008.

[16] Junger M.C., "Acoustic fluid-elastic structure interactions: Basic concepts", *Computers & Structures*, 65 (3), pp. 287-293, 1997.

[17] Khambaswadkar R., "Acoustic Optimization of an Underwater Vehicle", Master Thesis, Wright State University, 2005.

[18] Choi S.-H., Igusa T. and Achenbach J. D., "Nonaxisymmetric vibration and acoustic radiation of a submerged cylindrical shell of finite length containing internal substructures", *Journal of Acoustical Society of America*, 98 (1), pp. 353-362, 1995.

[19] Lyon R. and DeJong R., *Theory and Application of Statistical Energy Analysis*, Butterworth-Heinemann, Second Edition, 1995.

[20] Lyon R., Maidanik G., "Power flow between linearly coupled oscillators", *Journal of Acoustical Society of America*, 34, pp. 640-647, 1962.

[21] Fahy F. J., "Statistical Energy Analysis: A Critical Overview", from the collection "Statistical Energy Analysis, An Overview, With Applications in Structural Dynamics" by A. J. Keane and W. G. Price, Cambridge University Press, 1994.

[22] Langley R. S. and Bercin A. N., "Wave intensity analysis of high frequency vibrations", from the collection "Statistical Energy Analysis, An Overview, With Applications in Structural Dynamics" by A. J. Keane and W. G. Price, Cambridge University Press, 1994.

[23] Manning J. E., "Formulation of SEA parameters using mobility functions", from the collection "Statistical Energy Analysis, An Overview, With Applications in Structural Dynamics" by A. J. Keane and W. G. Price, Cambridge University Press, 1994.

[24] Cacciolati C. and Guyader J. L., "Measurement of SEA coupling loss factors using point mobilities" from the collection "Statistical Energy Analysis, An Overview, With Applications in Structural Dynamics" by A. J. Keane and W. G. Price, Cambridge University Press, 1994.

[25] Keane A. J., Price W. G., "Statistical energy analysis of strongly coupled systems", *Journal of Sound and Vibration*, Vol. 117(2), pp. 363-386, 1987.

[26] Keane A. J., Price W. G., “Statistical Energy Analysis of Periodic Structures”, *Proceedings of the Royal Society of London*. Series A, Mathematical and Physical Sciences, 423 (1865), pp. 331-360, 1989.
[27] Kurtoğlu İ., “Acoustically Induced Stress Analysis of Center Fuselage Skin Panels of a Basic Training Aircraft Using Statistical Energy Analysis”, Master Thesis, Middle East Technical University, 2009.

[28] Kiremitçi U., “Interior and Exterior Noise Analysis of a Single Engine Propeller Aircraft Using Statistical Energy Analysis Method”, Master Thesis, Middle East Technical University, 2009.

[29] Heckl M., “Vibrations of point driven cylindrical shells”, *Journal of Acoustical Society of America*, 34, pp. 1553-1557, 1962.

[30] Miller D. K., Hart F. D., “Modal density of thin circular cylinders”, *NASA Contractors Report*, CR-897, 1967.

[31] Szechenyi E., “Modal densities and radiation efficiencies of unstiffened cylinders using statistical methods”, *Journal of Sound and Vibration*, 19, pp. 65-81, 1971.

[32] Liu M., Keane A.J. and Eatock Taylor R., “Modeling liquid-structure interactions within the framework of statistical energy analysis”, *Journal of Sound and Vibration*, 238 (4), pp. 547-574, 2000.

[33] Blakemore M., Woodhouse J. and Hardie D.J.W., “Statistical power-flow analysis of an imperfect ribbed cylinder”, *Journal of Sound and Vibration*, 222 (5), pp. 831-832, 1999.

[34] Tso Y. K. and Hansen C. H., “The prediction of structure-borne noise transmission in ships using statistical energy analysis”, Acoustics Australia, 25 (3), pp. 97-102, 1997.

[35] Bron-van der Jagt, S., “Sound transmission through pipe systems and into buildings”, Doctor of Philsophy Thesis, Technische Universiteit Eindhoven, 2007.

[36] BAE Systems, Maritime Systems – Submarine Presentation, 2006.

[37] ESI Group, VA One 2010.5 User’s Guide, April 2011.

APPENDIX A

TUTORIAL ON FreeSEA SOFTWARE

Sample input codes for FreeSEA software program;

```
! Material definition
material label air
 young 344
 rho 1.19
 poisson 0.0
enter

! Plate definition
plate label wall_ref1
  material steel
  area 33.18
  peri 23.04
   damping 0.01
  thick 0.040
enter

!Room definition
room label compartment_back
  damping reverb
  material air
  volu 497.66
  area 411.96
  peri 106.08
enter

!Beam definition
beam
  label stiffener1
  damping 0.01
  material steel
  length 5.76
```

```
  ixx 2.25e-5
  iyy 6.40e-6
  itt 1.71e-5
  ipp 2.89e-5
  cross 0.012
enter



!**plates-stiffener connections
connect
  sub left_back
  sub stiffener1 90
  sub stiffener2 90
  sub stiffener3 90
  sub stiffener4 90
  sub stiffener5 90
  sub stiffener6 90
  sub stiffener7 90
  sub stiffener8 90
  sub stiffener9 90
  sub stiffener10 90
  sub stiffener11 90
  sub stiffener12 90
  sub stiffener13 90
  sub stiffener14 90
  sub stiffener15 90
  sub stiffener16 90
  sub stiffener17 90
  sub stiffener18 90
  sub stiffener19 90
  sub stiffener20 90
  sub stiffener21 90
  sub stiffener22 90
  sub stiffener23 90
  sub stiffener24 90
  sub stiffener25 90
  sub stiffener26 90
  sub stiffener27 90
  sub stiffener28 90
  sub stiffener29 90
  sub stiffener30 90
enter

!**plate connections
connect
```

```
  sub wall_ref1
  sub bottom_back -90
  sub bottom_mid 90
  length 5.76
enter

!** Resonant coupling
connect
  sub compartment_back
  sub back_back
  sub wall_ref1
  sub left_back
  sub right_back
  sub bottom_back
  sub top_back
enter

!** Non-Resonant coupling
connect
  sub wall_ref1
  sub compartment_back
  sub compartment_mid
  area 33.18
enter

! Power input 1 Watt into compartment_back
! at bottom plate assumed
source
 sub bottom_back flex
 power 1
enter
```

APPENDIX B

RESULTS FOR FreeSEA SOFTWARE

The results for FreeSEA product are given in 1/3-octave band frequencies. The results are tabulated through Table B-1 to Table B-4 and they are also presented in the Figure 5-1, Figure 5-2, Figure 5-3 and Figure 5-4.

Table B-1 The SPL values for the case of the unribbed plates

Frequency	**Sound Power Level (SPL)**		
	Back Compartment	Middle Compartment	Front Compartment
100	79.8	73.3	66.5
125	79.7	72.8	65.6
160	80.2	73.1	65.6
200	82.1	74.6	66.8
250	85.1	77.2	69.1
315	89.9	80.9	72.3
400	89.3	80.1	71.4
500	87.1	77.6	68.7
630	85.4	75.7	66.6
800	84.6	74.5	65.1
1000	83.0	72.6	62.9
1250	81.7	71.0	60.9
1600	81.0	70.0	59.5
2000	79.5	68.2	57.4
2500	78.4	66.9	55.7
3150	77.7	66.2	54.6

Table B-2 The SPL values for the case of the ribbed plates

Frequency	**Sound Power Level (SPL)**		
	Back Compartment	Middle Compartment	Front Compartment
[Hz]	[dB]	[dB]	[dB]
100	77.6	69.6	61.0
125	77.6	69.4	60.5
160	78.3	69.9	61.0
200	80.4	71.7	62.6
250	83.5	74.6	65.3
315	88.6	78.6	69.1
400	88.0	78.0	68.5
500	85.9	75.7	66.0
630	84.4	73.9	64.2
800	83.6	72.9	62.9
1000	82.1	71.1	60.9
1250	80.9	69.7	59.1
1600	80.3	68.8	57.9
2000	78.9	67.2	55.9
2500	77.8	65.9	54.3
3150	77.3	65.3	53.4

Table B-3 The velocity values of the upper plates of the model for the case of the unribbed plates

Frequency	**Top plate velocity profile**		
	back	middle	front
[Hz]	[m/s]	[m/s]	[m/s]
100	3,088780E-04	1,649370E-04	6,588530E-05
125	2,886460E-04	1,520420E-04	5,854010E-05
160	2,622710E-04	1,381900E-04	5,385710E-05
200	2,377270E-04	1,249390E-04	4,885410E-05
250	2,115510E-04	1,105990E-04	4,327570E-05
315	1,815650E-04	9,273530E-05	3,655550E-05
400	1,593630E-04	8,305510E-05	3,159860E-05
500	1,436540E-04	7,368370E-05	2,474500E-05
630	1,250600E-04	6,430490E-05	1,778910E-05
800	1,065930E-04	5,404500E-05	1,421974E-05
1000	8,965480E-05	4,274030E-05	1,011791E-05
1250	6,746460E-05	1,216440E-05	2,840228E-06
1600	9,016140E-05	7,027490E-05	6,150530E-05
2000	6,400640E-05	4,439350E-05	3,308790E-05
2500	5,192580E-05	3,603870E-05	2,615670E-05
3150	4,288240E-05	3,088440E-05	2,361100E-05

Table B-4 The velocity values of the upper plates of the model for the case of the ribbed plates

Frequency	**Top plate velocity profile**		
	Back	middle	front
[Hz]	[m/s]	[m/s]	[m/s]
100	2,919080E-04	1,262130E-04	4,740890E-05
125	2,738950E-04	1,168170E-04	4,212220E-05
160	2,507520E-04	1,070760E-04	3,927590E-05
200	2,294240E-04	9,775740E-05	3,613190E-05
250	2,068730E-04	8,766140E-05	3,259480E-05
315	1,815530E-04	7,509610E-05	2,839210E-05
400	1,612770E-04	6,799620E-05	2,476080E-05
500	1,470270E-04	6,090270E-05	1,937280E-05
630	1,299710E-04	5,381530E-05	1,375730E-05
800	1,129350E-04	4,583670E-05	1,123069E-05
1000	9,737860E-05	3,669200E-05	8,239674E-06
1250	7,833250E-05	9,859730E-06	2,768898E-06
1600	9,040670E-05	6,158660E-05	5,376610E-05
2000	6,840830E-05	3,968480E-05	2,950060E-05
2500	5,660220E-05	3,268700E-05	2,366450E-05
3150	4,709260E-05	2,846110E-05	2,171240E-05

APPENDIX C

RESULTS FOR VA One SOFTWARE

The overall pressure and velocity contour plots for the second condition with the submarine speed 10 m/s are provided in order to observe the pressure and velocity distribution.

The cylindrical shell with light blue color, meaning high overall pressure and velocity levels, is the back compartment.

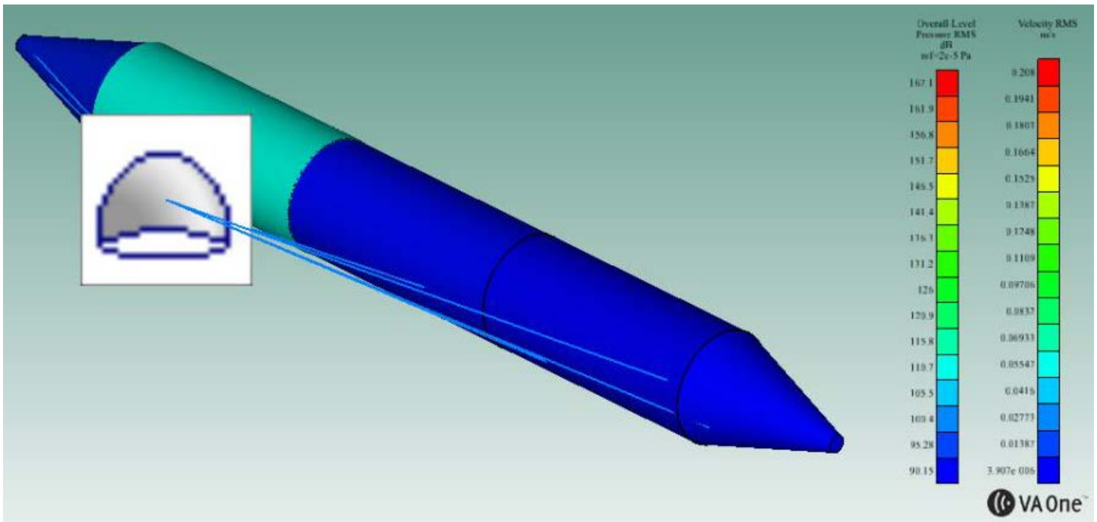

Figure C-1 Pressure and velocity contour plots of the external elements – Second Condition – 10 m/s

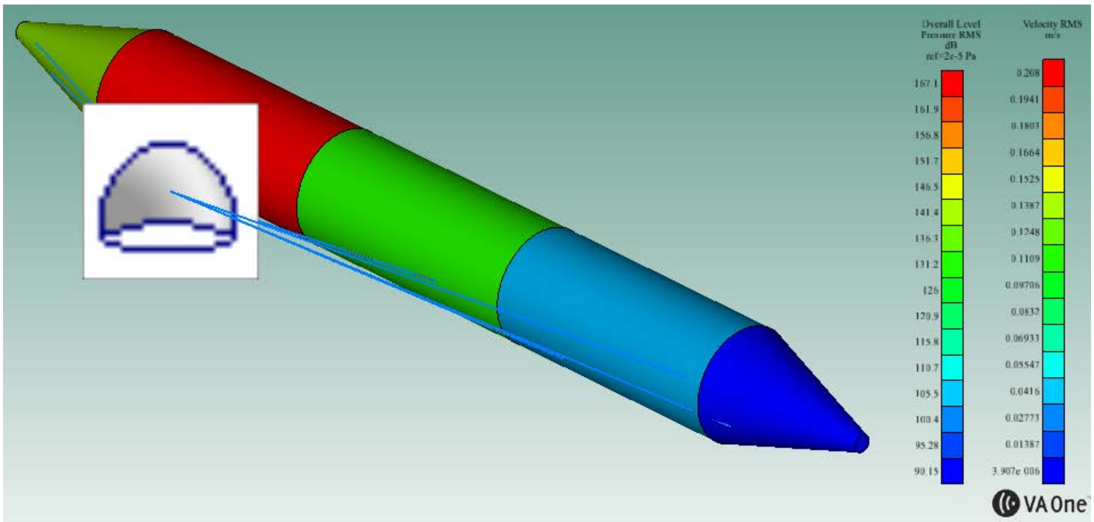

Figure C-2 Pressure and velocity contour plots of the acoustic cavities – Second Condition – 10 m/s

The cavity with red color, which means having the highest pressure and velocity levels, is the back compartment.