Noise Considerations for Propulsion System Evaluation

Muhammad Radityo Pradipta
(4505441)

Delft University of Technology

TU Delft Delft University of Technology

Challenge the future

Noise Considerations for Propulsion System Evaluation

By

Muhammad Radityo Pradipta

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Supervisor	: Rear Admiral Ir. K. Visser	TU Delft, Chairman
	Ir. Ioana Georgescu	TU Delft, Daily Supervisor
	Ir. A. Blanken	IHC MTI, Daily Supervisor
	Dr. ir. P.R. Wellens	TU Delft, External Committee Member

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Summary

The laws and regulations regarding the radiated noise and the on-board noise of the ship are expected to be stricter in the future. There is a big concern in this field, especially for the underwater noise. The radiated noise from the ship is harming the life of the underwater mammals while the on-board noise is threatening the health of the on-board crew. One of the main contributors to the noise generated by a vessel is the propulsion system. It is true that the noise reduction can be achieved by doing the corrective action in the later design stage, such as installing mounting and noise absorption material. Nevertheless, the decision in early-design stage often gives the highest impact on the noise generated by the propulsion system where the level of uncertainty is high. This thesis has two main goals; to determine the design choices of propulsion systems which affect the noise excitation, and to develop an evaluation methodology to assess a certain power configuration from the perspective of generated noise.

First, the aspects of the propulsion system that affect its noise are determined. Those are transmission types, number of engines, number of shafts and number of compartments. The loading point is also included as one of the parameters. Although it is not a design choice, it has a significant role in the noise generated by the propulsion system. Second, the selection of the significant noise source in the propulsion system needs to be done. There is much equipment inside a propulsion plant, but not all of them give sufficient contribution to the overall noise level. Based on the literature review, the equipment that is considered as the main noise sources are the diesel engine, the diesel generator set, the reduction gear and the electric motor.

A noise model from SNAME is implemented in this project to predict the airborne and structure-borne noise source levels of the equipment and the transmission losses to the receiver location. An engine room-sizing model is developed in this project since the transmission loss is a function of the compartment dimensions. Furthermore, the room dimensions depend on the equipment dimensions. Therefore, it is necessary to develop a model to predict the size of the equipment too. Afterwards, the evaluation methodology is established to quantify the effect of a certain design choice towards the noise of a propulsion system.

The effects of varying ship requirements are also investigated to see the behaviour of the model with a different input. These requirements are the ship installed power, the ship propulsion power and the ship auxiliary power. It is not possible to do this analysis along with the evaluation methodology due to the feasibility of the simulation.

This project provides a general guideline for the marine engineer to evaluate the propulsion system based on the noise considerations in the early design phase. The evaluation methodology proves to be applicable to a wide range of propulsion plant type. It is possible to extend the application of this method for a ship with prime mover other than the diesel engine and the electric motor. However, the model and the method are limited to the noise on-board of the ship. If the proposed model and methodology are used for the radiated noise considerations, there should be several adjustments to the design choice.

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1

Introduction

1.1 Background Information

Noise is becoming an increasingly important issue in the maritime world. Global shipping activity is one of the significant contributors to ocean noise, and it does not show any decreasing trends towards the future. A study predicted that in 2030 the maximum noise capacity of the global shipping fleet would escalate by up to a factor of 1.9. The increase in noise is caused by the increasing number of container ships, bulk carriers, and oil tankers [1].

A wide range of noise emissions produced by a vessel can be summarised into the following three categories; (1) noise and vibration on board, (2) underwater noise radiation, and (3) airborne noise emissions from the ship [2]. These unwanted sounds pollute and have consequences for different environments. Firstly, the noise and vibration on board are threatening the health of the crew and the passengers. A survey on comfort on board of a ship shows that the acoustic noise is the primary source of complaints [3]. Secondly, the underwater noise radiation generates a disturbance in communication between marine mammals. The damage from underwater noise is still the subject of on-going research [4]. It is expected to have more strict regulations shortly [5]. Lastly, airborne noise usually becomes a disturbance to the environment around the mooring location of ships.

Noise pollution is coming from various sources of noise inside the ship. For instance, the HVAC ducting system, pumping system, hydraulic system and another type of systems which consist of thousand parts and instruments. Generally speaking, the noise sources in a vessel can be split into four categories, i.e., [6]

1. Machinery Vibration: propulsion machinery, ships' services, and auxiliary machinery.
2. Propellers: jet, super-cavitating propeller, azimuth propeller and other forms of in water propulsion.
3. Acoustic noise within compartment below the waterline
4. Hydroacoustic noise generated external to the hull by flow interaction with appendages, cavities, and other discontinuities.

Sometimes, the noise created by these sources not only damages the environment around the vessel but also interferes with the ship's sonar system.

One of the significant sources of noise is the machinery of the propulsion system: either the noise produced by the machinery and transmitted to the water through the hull or the noise induced by the propellers. Not surprisingly, this means that the sound generated from the propulsion system can be reduced through ship design. When selecting the configuration, one needs to consider the noise generated by the individual components and the possibilities a particular arrangement offers for noise reduction strategies.

The propellers radiate noise both directly from the blades and through vibrations transmitted to the hull. One of the phenomena in which the propeller radiates noise is cavitation [6]. One way to prevent this is by choosing a propeller design that has better performance in cavitation than others. There is much research going on to reduce the cavitation by improving the design of the propeller.

1.2 Boundaries

Due to the complex nature of the subject, clear boundaries need to be determined to define what is inside and outside of the scope of this thesis.

- Since the prevention of cavitation is more a matter of improving a certain design than a trade-off between different criteria, it will not be considered further in this study.
- Diesel engines and electric motors are always considered as the prime movers of the propulsion system. Therefore, gas turbines, steam turbines and other types of prime movers are not considered in this project.
- One of the main sources of radiated noise is the propulsion system machinery. The level of radiated noise depends on the transmission path from on-board noise to the underwater noise. This path is affected by the high level of ship structure details and subsequently should be addressed in later design stages. Therefore, modelling the underwater noise is excluded in this project.
- The compartment design used in the analysis is a simplified shape, a block with the equipment inside it. Therefore, details of the structure compartment per arrangement are not considered here. All the configurations have a typical compartment.

1.3 Research Objectives

The main goal of this project is to help the marine engineer to understand better the acoustical consequences by choosing a certain option when selecting the propulsion system. It is important to find the aspects of the propulsion system configuration which are relevant to noise pollution and a way to quantify their influence. The term propulsion system here includes the propulsion driver system and the auxiliary system. In practice, noise reduction is more to the corrective action rather than consideration in the early design phase. Often, the decision in the early stages of design gives the greatest impact when the information is still minimum, and much uncertainty is present [7]. The impact of choosing a specific configuration for the on-board power system on future noise concerns (either pollution or costs and extent of necessary corrective action) is not yet known. This project aims to offer at least a partial solution to this problem.

In this research, the evaluation method for analysing the acoustical performance of a propulsion system is developed. The procedure has to be applicable to a wide range of ship drive configurations with different power ranges and loading point. The propulsion systems that are investigated in this project are the diesel-mechanical plant, the diesel-electrical plant, the diesel-hybrid plant and the battery plant. The chosen parameters that affect the noise levels are based on the early design choices. The investigated parameters are the transmission types, the number of engines, the number of shafts, the number of compartments and the loading factor. Some constraints are going to be imposed due to the high number of possible combinations.

To summarise, the research objectives are,

"Determine the aspects of propulsion systems which affect the noise excitation."

"Develop a method to evaluate a certain propulsion system from the perspective of generated noise."

The research questions that need to be answered in this project are,

1. What is the most effective way to model the noise pollution generated by a propulsion system?

2. How will the early design choices affect the noise excitation of a propulsion system?

2.1. How do different propulsion system configurations that have similar transmission type perform in the area of noise pollution?

2.2. How do different transmission types of a propulsion system perform in the area of noise pollution?

1.4 Structure of the Report

The thesis is divided into six chapters. The first chapter consists of background information, the boundaries of the project and the research objectives.

The second chapter of the thesis is used to explain the theoretical foundation of the topic and introduce the reader to the subject of noise generation and prediction on board ships. It starts by presenting the fundamental theories of airborne and structure-borne noise in the propulsion system and explains how the noise is generated. The focus then shifts to the individual noise sources considered, thus establishing the project boundaries in more detail. Lastly, the chapter presents an overview of existing noise models and concludes with the motivation for the selection of the model used in this project.

The third chapter of the thesis describes the research approach employed in this project. Various models and methods are being implemented and developed for this topic. Firstly, the SNAME noise model is discussed comprehensively. It is used to predict the noise source level for the propulsion equipment and the transmission loss from the source to the receiver location. Secondly, the design choices that are used as the input parameters are determined here. These parameters are expected to affect the noise level of a propulsion system. The constraints of these parameters are also developed to limit the possible combinations of the propulsion system layout. Thirdly, the engine sizing model is being developed in the third chapter. It is required to build this model since the noise model is a function of the engine room dimensions. Lastly, the chapter also shows the evaluation methodology that used to analyse the effect of each parameter/design choice. These parameters are related to each other, thus the relations between them are defined too. The methodology used to choose an operation condition for a part load operation is discussed. It shows how to get the minimum noise level by playing with the operation point of the propulsion equipment.

The fourth chapter of the thesis presents the simulation results and an example of the proposed evaluation methodology. Results of the airborne and structure-borne noise are obtained and analysed thoroughly to see what are the influences of certain parameters on propulsion system noise levels. Apart from the individual parameters, the airborne and structure-borne noise levels are compared between four main transmission types in order to assess their relative performance from an acoustical point of view. The example is carried out for an imaginary vessel. The impact of all the assumed values regarding this vessel is investigated in the following chapter.

The fifth chapter of the thesis analyses the sensitivity of the obtained results to several assumptions made throughout this work. Firstly, the impact of the model used for the sizing of the components on the end results is quantified. Afterwards, the focus changes to the impact of the assumptions about the vessel used in the example; namely the installed power and the split between propulsion and auxiliary power demands.

The final chapter gives the answers to the initial research questions, thus providing the overall conclusions of the presented work. Additionally, recommendations of further study are made regarding both specific aspects of the current approach, and its applicability in the field of early ship design.

2

Noise in Propulsion System

2.1 Introduction

When looking at the ship propulsion system, there are three types of noise generated from it, i.e., (1) Airborne Noise (2) Structure-borne Noise, and (3) Underwater Noise [6] [8] [9]. In this project, the focus is to analyse the first two types of noise since the underwater sound is heavily affected by the propeller design rather than choices of the propulsion system.

All the noise produced by the propulsion system is the result of vibrating machinery. The vibration energy from the equipment is transmitted to the surrounding air and the ship structure with a complicated mechanism. The energy is transferred to the atmosphere generates airborne noise while the structure-borne noise is the reaction of ship structure being excited by the machinery vibration. Furthermore, the airborne and the structure-borne noise will travel along the structure and subsequently generate the underwater noise [9]. There are three primary acoustic transmission paths for underwater noise, i.e., (1) Airborne path (2) First Structure-borne path, and (3) Secondary Structure-borne path. The latter one is the incident airborne noise diffuses on the boundary surface creating vibration in the hull structure [8]. A detailed illustration of these mechanisms can be seen in the figure below.

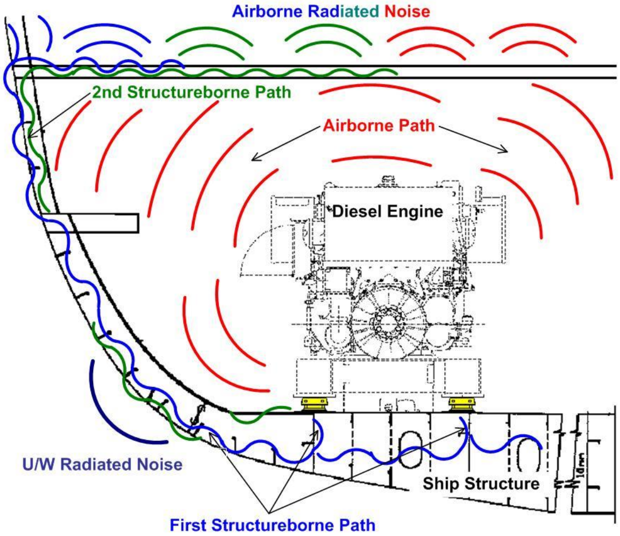


Figure 2.1 Acoustic Path in Ship Structure [9]

The next sub-chapter will be divided based on the types of noise that are being investigated in this thesis. First, the theoretical backgrounds of airborne and structure-borne noise are discussed. Second, there is a comprehensive explanation of how to predict the source level of the airborne and the structure-borne noise from previous studies. Last, the acoustic transmission paths are discussed to analyse the sound levels received by the observer at a certain location.

2.2 Fundamentals of Airborne Noise for Propulsion System

Airborne noise is one of the significant acoustical discomforts experienced by the crews inside a ship [3], especially in the working area that is adjacent to the machinery room. The long-term exposure to a certain level of airborne noise can cause permanent damage to the human hearing ability [2]. Much effort has been made to reduce the airborne noise inside the ship [10] [11] [12].

2.2.1 Basic Acoustics

Sound Pressure Level

The sound of the machinery is the result of the surrounding air excitation due to the transfer of vibration energy from the equipment. This excitation creates a fluctuation in the air pressure above and below atmospheric pressure which is known as sound pressure. The variation in the pressure creates a sound wave that can be represented by the equation as follows [13],

$$p(t) = p_0 \sin(2\pi f)\, t \tag{2.1}$$

Where $p(t)$ is the instantaneous sound pressure in the measure of pascals (Pa), p_0 is the maximum sound pressure, f is the frequency in hertz (Hz), and t is the time in seconds. A normal human hearing can perceive sound in the frequency range of 15 Hz to 16000 Hz.

The airborne noise observed by the receiver at a certain distance from the source can be defined by the sound pressure level [13].

$$L_p = 20 * log10 \left(\frac{p(t)}{p_{ref}}\right)^2 \; db\; re\; p_{ref} \tag{2.2}$$

Where, L_p is the sound pressure level and p_{ref} is the reference pressure. The latter variable is equal to $2 * 10^{-5}\ N/m^2$ when the sound wave travels in the air. If the medium is water the value decrease to $1 * 10^{-6}\ N/m^2$ since water has a higher density compare to air [13].

Another way to describe the sound pressure level is to use the sound intensity. It is defined as the power carried by a sound wave per unit area. The formula to calculate it is as follows [13],

$$I = \frac{p_{rms}^2}{\rho_0\, c_0} \tag{2.3}$$

In which ρ_0 is the density of air, c_0 is the speed of sound in the air and p_{rms} is the root mean square pressure that can be calculated by Equation (2.4).

$$p_{rms} = \frac{1}{T - T_0} \int_{T_0}^{T} p(t)^2 dt \tag{2.4}$$

Where T is the period of the sound wave. Thus, the equation in (2.2) can be rewritten as [13],

$$L_p = 10 * log10 \left(\frac{I}{I_{ref}}\right) db\ re\ I_{ref} \tag{2.5}$$

where, I_{ref} is equal to $10^{-12}\ W/m^2$.

Sound Power Level

The sound power level is an important measure to describe a noise source. The main difference between the sound power level and the sound pressure level is that the first one is independent of the receiver distance and the room condition. It merely quantifies the amount of power a source has. It can be used to calculate the sound pressure level for the sound propagation model when the locational directivity factor, the room condition, and the distance to the receiver are known.

The sound power level can be expressed as [13],

$$L_w = 10 * log10 \frac{W}{W_0}\ dB\ re\ W_0 \tag{2.6}$$

Where W is the sound power in watts (W) and the W_0 is reference sound power, standardized at $10^{-12}\ W$.

Source Directivity Factor

The location of a sound source will determine its directional characteristics because the sound directivity factor is profoundly influenced by reflecting surfaces. For instance, a hanging speaker from the ceiling radiates sound equally in all directions (sphere distribution). Nevertheless, when the speaker is placed on the floor, the noise is constrained within half the space (hemisphere distribution) increasing the sound intensity twice as high as the one from the hanging configuration.

The source directivity factor is a dimensionless parameter. One can find its value by using the formula as follows [14],

$$Q = 2^{n_s} \tag{2.7}$$

n_s is the number of surfaces touching the source. Therefore,

Source(s) Location	Surface(s)	Q
Hanging	0	1
Middle of the floor	1	2
The Floor meets the walls	2	4
Corner, the floor meets two walls	3	8

Table 2.1 Sound Directivity Factor value as a function of source(s) location

Frequency Band Levels

The sound that perceived by a typical human is not only characterised by the loudness level but also by its frequency. There are two types of noise based on frequency classification (1) tonal noise, and (2) broadband noise. A sound source that can only be heard at a single frequency is called tonal noise. For instance, the sound wave created by a vibrating tuning fork. If the sound source has signals in some range of frequencies it is defined as broadband noise [6].

In this thesis work, the broadband noise will be investigated. Therefore, one should choose the frequency bands to characterise the noise spectrum. Practically, the 1/1 octave band and 1/3 octave band are the most widely used for engineering application. The first one can be determined as,

$$f_n = f_{n-1} * 2 \tag{2.8}$$

While for the latter one,

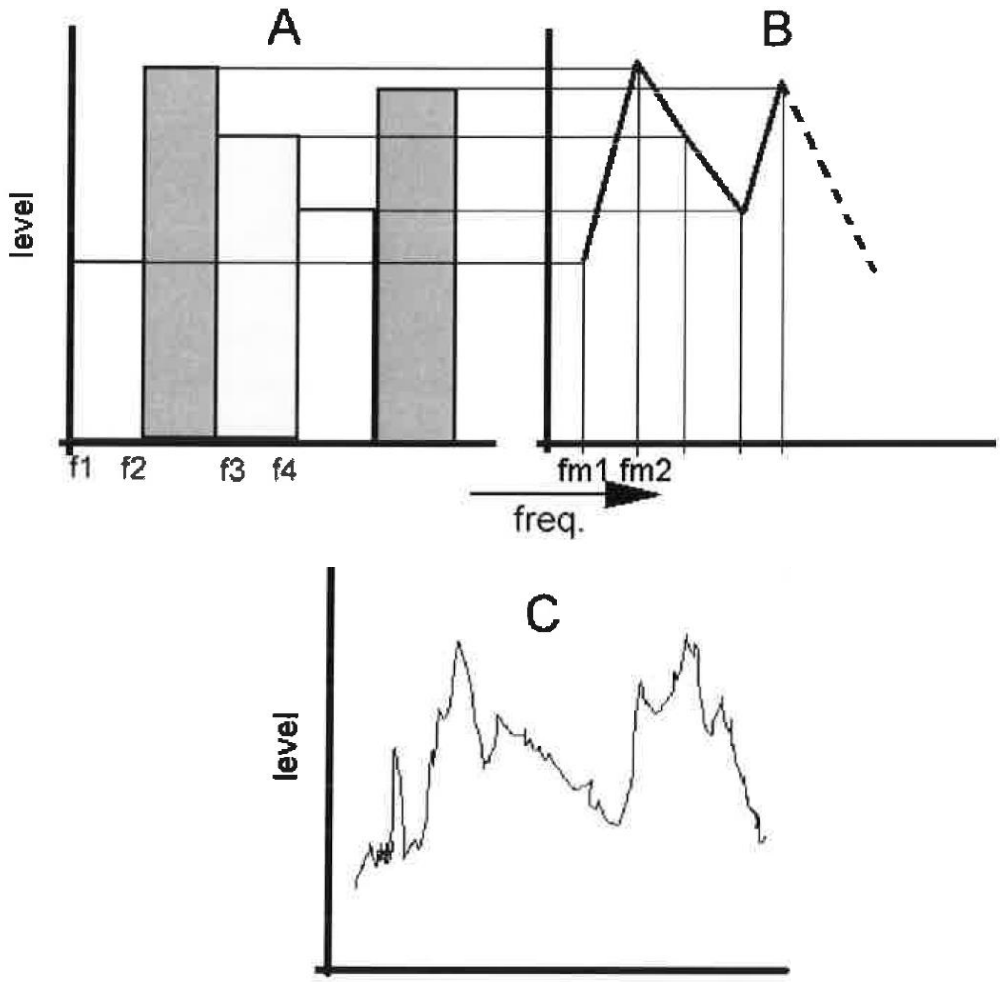
$$f_n = f_{n-1} * 2^{\frac{1}{3}} \tag{2.9}$$

Thus,

Lower Limit 1/1-octave [Hz]	Lower Limit 1/3-octave [Hz]	Centre Frequency [Hz]	Upper-Limit 1/3-octave [Hz]	Upper limit 1/1-octave [Hz]
11.2	11.2	12.5	14.1	22.4
	14.1	16	17.8	
	17.8	20	22.4	
22.4	22.4	25	28.2	44.7
	28.2	31.5	35.5	
	35.5	40	44.7	
44.7	44.7	50	56.2	89.1
	56.2	63	70.8	
	70.8	80	89.1	
89.1	89.1	100	112.2	177.8
	112.2	125	141.3	
	141.3	160	177.8	
177.8	177.8	200	224	355
	224	250	282	
	282	315	355	
355	355	400	447	708
	447	500	562	
	562	630	708	
708	708	800	891	1413
	891	1000	1122	
	1122	1250	1413	

Table 2.2 Standardized 1/1 Octave Band and 1/3 Octave Band

Aside from octave band, a narrow band frequency is also used to characterise the sound spectrum. It gives more detailed information about the sound wave rather than octave band. However, it provides some drawbacks due to more data processing and is not directly linked to the perception of sound by a human.



A = Sound spectra in 1/1 or 1/3 octave bands using bar diagram form
B = Sound spectra in 1/1 or 1/3 octave bands by connecting each level at the centre frequencies
C = Sound spectra in narrow-band frequencies

Figure 2.2 Comparison of various frequency bands presenting sound spectra information [15]

2.3 Fundamentals of Structure-Borne Noise for Propulsion System

The vibration energy from the sources is not only transmitted to the surrounding air but also through the ship structure. When it is propagated through the solid material, it is termed as structure-borne noise. Unlike airborne noise, the structure-borne noise can reach the receiver not only adjacent to the sources but also far away from it. One of the most effective ways to reduce the transmitted structure-borne noise level is to install a resilient mounting. The drawbacks of it are that some of the vibrational energy is reflected back to the engine which can cause damages to the equipment [16]. Many studies have been done to reduce the structure-borne noise level of a diesel engine and its transmission on board of the ship [16] [17].

2.3.1 Basic Vibro-Acoustics

Vibration Levels

In practice, the structure-borne noise source level can be expressed as velocity, acceleration and displacement level. It depends on the tool uses to measure the vibration level. The most common one is to use accelerometers because it is available in all types and there is a very broad choice of it. The velocity and displacement measurements are quite rare to use nowadays because one could directly derive those two values from the acceleration levels. The acceleration level can be written as,

$$L_a = 20 * \log\left(\frac{a(t)}{a_{ref}}\right) \; dB \; re \; a_{ref} \tag{10}$$

Where $a(t)$ is the instantaneous acceleration and a_{ref} is the reference acceleration. According to ISO the a_{ref} is equal to $10^{-6}\, m/s^2$ but there is also another common reference value which is equal to a_{ref} is equal to $10^{-5}\, m/s^2$. The velocity level is as follows,

$$L_v = 20 * \log\left(\frac{v(t)}{v_{ref}}\right) \; dB \; re \; v_{ref} \tag{11}$$

The v_{ref} is equal to $10^{-8}\, m/s$ based on ISO but there is also another common reference value which is equal to v_{ref} is equal to $10^{-9}\, m/s$. The displacement levels can be expressed as,

$$L_d = 20 * \log\left(\frac{d(t)}{d_{ref}}\right) \; dB \; re \; d_{ref} \tag{12}$$

The d_{ref} is equal to $10^{-11}\, m$ but it is not common to represent the vibration level in the displacement measurement. The laws and regulations typically use both the acceleration and velocity levels.

2.4 Propulsion Systems Noise

2.4.1 Noise Sources

In general, the propulsion system consists of the prime movers, the reduction gears, the shafts, the electric motors and the propellers. These principal components produce a high level of noise when operating. Vibratory forces from the equipment induce the air pressure fluctuation.

The figure below shows the overview of the frequency range of the main contributors to the ship airborne and structure-borne noise. Most of the sources are part of the propulsion system. Every component has its unique frequency band because they work at different speeds and generate sound with different mechanisms. The propulsion machinery produces noise ranging from 10 to 1000 Hz while the cavitation and the turbines are the significant sources above 10000 Hz.

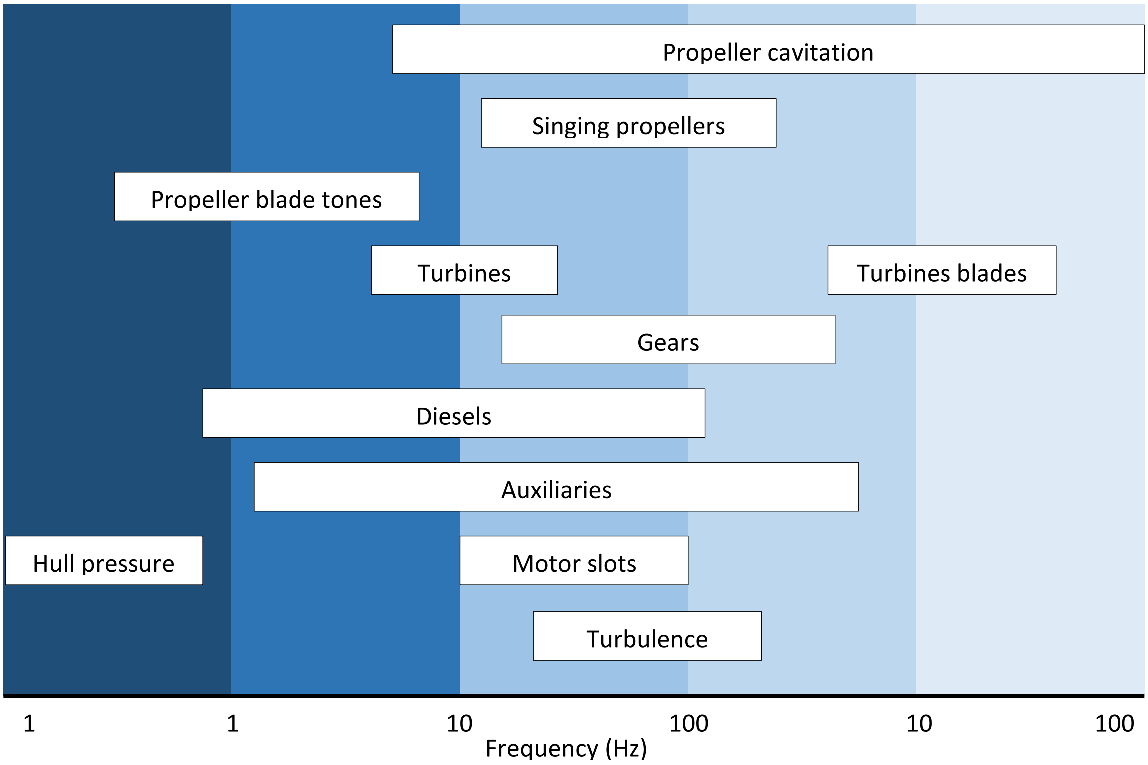


Figure 2.3 Frequency ranges from radiated noise by ship noise sources [6]

2.4.2 Airborne and Structure-borne Noise Mechanism

The simplified view of the machinery components from diesel-electric propulsion and the causes of sound generated by the equipment is provided in figure below. Those mechanisms trigger vibration which leads to the airborne noise and the structure-borne noise. The detailed explanation regarding them are given as follows,

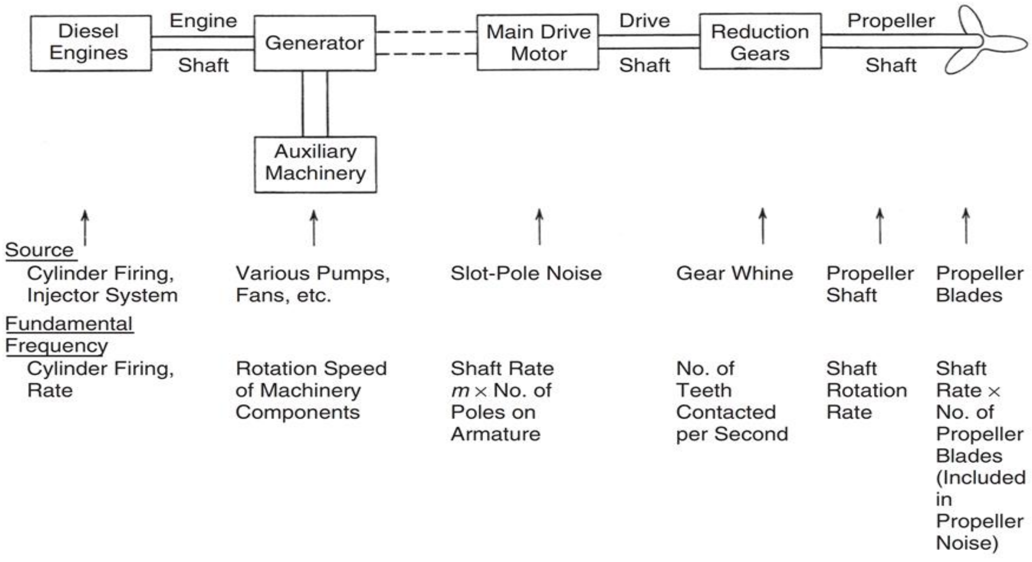


Figure 2.4 Overview of noise sources related to ship propulsion system [6]

- Diesel engine

The driving factor of noise from a reciprocating engine is the rapid changes in cylinder pressure. The changes are responsible for the direct and indirect excitation of the diesel structure. The direct excitation vibrates the piston and cylinder head, typically known as combustion noise. The indirect one result in mechanical noise [18].

There are lots of types of mechanical noise, for instance, piston slap, connecting rod impact, crankshaft bearing impact and so on. The dominant noise mechanism from mechanical noise is piston slap. It is caused by the collision between the piston and cylinder inner wall when the combustion occurs [19]. In practice, combustion noise is the most significant source compare to the other [20].

- Generator

The mechanical imbalance of the rotor is causing vibration that leads to the unwanted sound of the generator. The defects are caused by manufacturing reasons, for instance, eccentricity, blow holes in castings, clearance tolerances and distortion. The vibration that occurs is proportional to the force. Since the force is proportional to the square of angular speed, the radiated power increases the rotational speed to the power of four [6].

- Electric Motor

The working principle of the electric motor is to convert electrical energy into mechanical energy. Considering that it is not possible to create an electric motor with 100% efficiency, there will be some heat produced in the conversion energy process. In order to avoid overheating, usually, a fan will be installed in the electric motor. Unfortunately, it is very noisy and becomes the most significant source of noise in electric motor [21].

As for the electric motor without a fan, noise is still produced but will result in a lower level. The driving factors of noise in electrical motors can be categorised in three-types, mechanical, electromagnetic and aerodynamic. The detailed factors are represented in the table below.

Types	Description
Mechanical	Excessive bearing clearance
	Nonround bearings
	Rotor unbalance
	Rotor eccentricity
	Crooked shaft
	Brush and brush holder vibration
	Misalignment
	Loose laminations
Electromagnetic	Magnetostriction
	Torque pulsations
	Air gap eccentricity
	Air gap permeance variation
	Dissymmetry
	Sparking or arcing
Aerodynamic	Fan blade-passing frequency
	Turbulence
	Noise due to airflow path restrictions

Table 2.3 Electrical Motor Noise Driven Factors [21]

- Gearbox

  There are two types of gear noise, i.e., (1) gear rattle and (2) gear whine. Gear rattle is an impact-induced sound that occurs in lightly loaded gears that are externally excited by an oscillating torque. Gear whine is sound due to the meshing of the gear teeth. It happens at the gear mesh frequency which can be expressed as [22],

$$f_m = N_{teeth} * f_s \qquad (2.13)$$

  Where N_{teeth} is the number of gear teeth and f_s is the shaft frequency.

- Auxiliary Machinery

  Components such as the auxiliary machinery also create noise but not as significant as the main components. In other words, one can say that the auxiliary equipment noise is being masked by the main machinery [23].

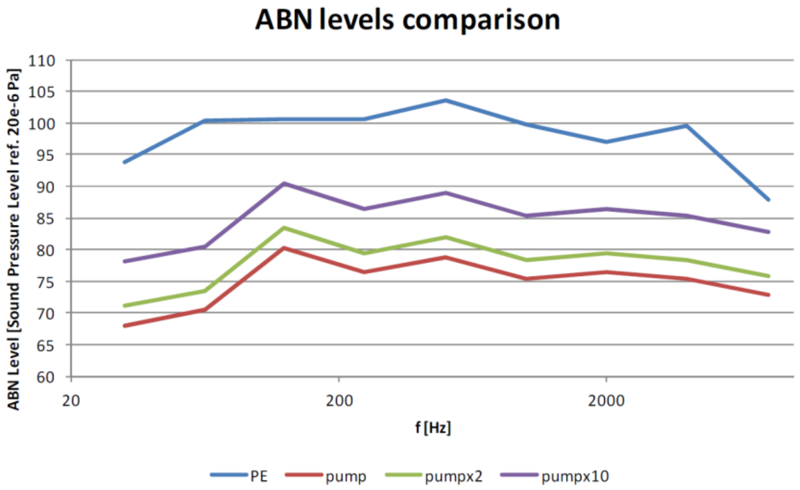


Figure 2.5 Comparison of Airborne Noise Levels of several pumps and one propulsion engine [23]

As can be seen from the chart above, the sound pressure level of one propulsion engine is significantly higher compared to one pump. If one increases the number of pumps to 10, the difference is still evident, more or less 15-20 dB for each frequency. Thus, it is possible to neglect the auxiliary machine as airborne noise sources.

2.5 Noise Models

There are a lot of existing studies on how to model the on-board and radiated noise of the ship [6] [15] [24] [25]. One can categorise it by two classifications, (1) Mathematical Models and, (2) Empirical Models.

The mathematical models provide theoretical approaches to calculate the ship on board and radiated noise. The level of ship details required is high with this method. For example, to estimate the structure-borne noise it is necessary to provide the complete definition of the ship structure, which is not convenient at the beginning of the design phase. Moreover, the calculation time can also be another big issue since the mathematical model takes a long time to finish. Nevertheless, the flexibility behaviour of mathematical models allows the user to investigate the effect of modification in the structures such as building materials, additional damping, discontinuity and so on.

The empirical method is a mixed between measurement and mathematical models. The source strengths of the propulsion system machinery determined empirically or mathematically. The variables of the equation will be represented by the parameters of the propulsion equipment such as the power of the engine, the speed of the machine, the engine weight and among others. In the early design stage phase, this method is suitable since it is fast, simple, inexpensive and sufficiently accurate.

2.5.1 Mathematical Models

There are various ways to solve the acoustic problems with the mathematical models. The first one is Finite Element Method (FEM). It is a deterministic analysis in which the "general discretisation procedure of continuum mechanics problems posed by mathematically defined statements" [26]. It is one of the practical methods used by the engineer to solve the acoustical problems. The finite element model requires a lot of variables and details to solve an acoustical problem. These variables are local in space and time.

The second one is Statistical Energy Analysis (SEA). According to Lyon "**Statistical** emphasises that the systems being studied are presumed to be drawn from statistical populations having known distributions of their dynamical parameters. **Energy** denotes the primary variable of interest. Other dynamic variables such as displacement, pressure, and among others, are found from the energy of vibrations. The term **Analysis** is used to emphasise that SEA is a framework of study rather than a particular technique" [27].

Theoretically, SEA is the total opposite of the deterministic analysis. FEM solves the problem by only taking its conditions into account without considering another sequence. On the other hand, SEA answering the question in a general way and taking all the situation into account. It was developed because the inability of deterministic analysis to model the behaviour of the structure at higher frequencies [28]. Other advantages of SEA over FEM are it does not require as much detail as FEM and has a brief computational time. The basic principle of SEA is that the average power flow between two coupled elements proportional to the difference in the average modal

The basic principle of SEA is that the average power flow between two coupled elements proportional to the difference in the average modal energies. Power flows out of a subsystem either by transmission to another subsystem (P_{12} or P_{21}) or through dissipation ($P_{1,diss}$). Power flows into a subsystem by transmission from another subsystem (P_{12}) or excitation from the external sources (P_1 or P_2). The figure below illustrates the schematic of this method.

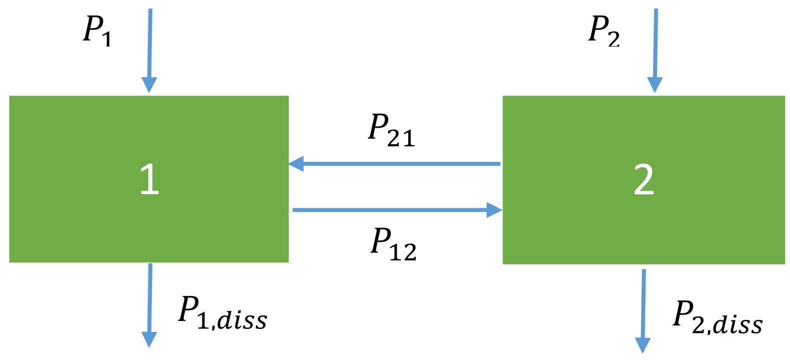


Figure 2.6 Power flow between two systems in SEA

The dissipated power in subsystem one can be calculated by the equation as follows:

$$P_{1,diss} = 2\pi f \eta_1 E_1 \tag{2.14}$$

Where η_1 is the damping loss factor of element one and E_1 is the total vibrational energy of the modes at frequency f.

The transmitted power between the two subsystems is given by the formula below:

$$P_{12} = 2\pi f \eta_{12} E_1 \tag{2.15}$$

Where η_{12} is the coupling loss factor between subsystem one and two. This coupling factor only depends on the physical properties of the two coupled subsystems.

The power balance for the two subsystems can be written as:

$$P_1 = P_{1,diss} + P_{12} = 2\pi f(\eta_1 + \eta_{12})E_1 - 2\pi f \eta_{21} E_2 \tag{2.16}$$

$$P_2 = P_{2,diss} + P_{21} = 2\pi f(\eta_1 + \eta_{12})E_1 - 2\pi f \eta_{21} E_2 \tag{2.17}$$

In a specific case for instance, if there is no input power to subsystem two, the energy flow inside the subsystem one always be higher than the subsystem two. Therefore, the power balance equation will be simplified:

$$\frac{E_2/N_2}{E_1/N_1} = \frac{\eta_{21}}{\eta_2 + \eta_{21}} \tag{2.18}$$

Where N_i is the mode count of a subsystem i.

2.5.2 Empirical Models

The empirical models are developed based on measured data from laboratory experiments, shipboard investigations and an extensive databank of the vessels. From those data, the mathematical models will be derived and combined with the acoustical theory.

SNAME Model and TNO Cabin are one of the most established empirical models for ship noise. In general, both of them use the same scheme to predict the sound in the receiver location.

Figure 2.7 Sound Transfer Path

The output from TNO cabin is the sound pressure level in a particular location. It will indicate which sound source and frequency is significant to the sound pressure levels. Typically, the TNO Cabin calculation can be written as [15]:

$$L_{p,receiver} = L_{a,sources} - \Delta L_{a,sound\ path} + \left(L_p - L_a\right)_{receiver} \tag{19}$$

Where $L_{p,receiver}$ is the sound pressure level in the receiver room, $L_{a,sources}$ is the structure-borne source level described by acceleration level at the feet of rigidly installed engines, $\Delta L_{a,sound\ path}$ is the effect of transmission path due to position and mounting of the engine, and sound transfer via hull structures (depends on ship type) from the sources location to receiver room, and lastly $\left(L_p - L_a\right)_{receiver}$ is a variable presenting the effect of acoustic treatment in the receiver room. TNO cabin uses the octave band centre frequency ranging from 63 – 2000 Hz.

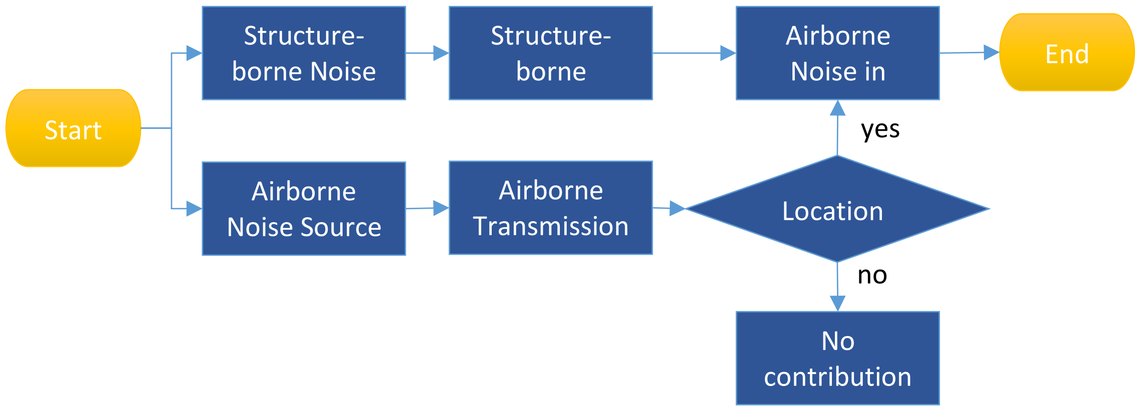


Figure 2.8 Flow Chart of TNO Cabin Model

In TNO Cabin model, the empirical formulas for structure-borne noise calculation are derived for various equipment. On the other hand, the airborne noise sources taken into account is only the diesel engine. The TNO cabin assumes that the diesel engine noise is masking other airborne noise sources inside the engine room. Moreover, it can contribute significantly to the noise level in a room adjacent to it. This contribution means that if the receiver located in another room close to it, the sound pressure level experienced by the receiver is added by the airborne noise generated from the diesel engine.

Noise prediction by SNAME also uses the three elements as shown in Figure 2.6. The airborne noise sources will be measured in sound power level instead of sound pressure level. In the airborne transmission path, the sound power level will be converted into the reverberant and the free field sound pressure level. The structure-borne noise sources are described by the free acceleration levels. The final output is similar to TNO Cabin where the sound pressure levels represent the noise at the receiver location. However, unlike TNO Cabin, the SNAME model considers various equipment for the airborne noise sources. Thus, the calculation flow chart is a bit different.

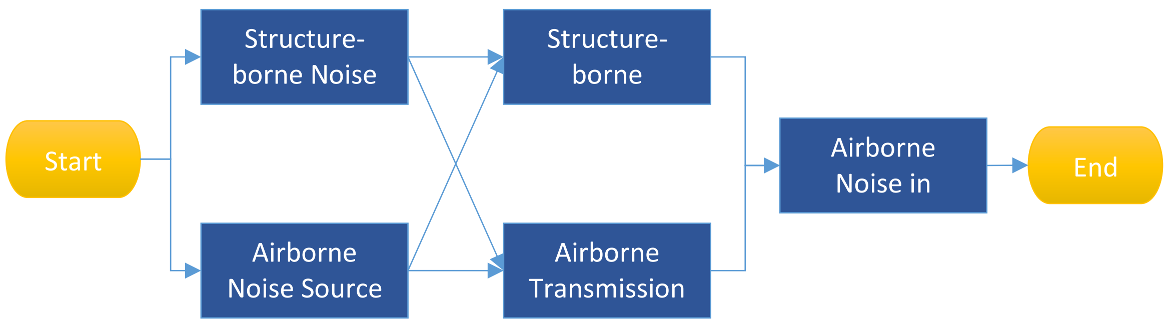


Figure 2.9 Flow Chart of SNAME Model

The range of frequencies of SNAME is the octave band centre frequency from 31.5 – 8000 Hz. SNAME model has more extensive frequency range compared to TNO Cabin.

2.6 Noise Models Selection

	Level of Details Required	Calculation Time	Accuracy	Frequency Range
FEA	High	High	High	0-250 Hz
SEA	Moderate	Fair	Fair	>500 Hz
TNO Cabin	Low	Low	Fair	31.5 – 2000 Hz
SNAME	Low	Low	Fair	31.5 – 8000 Hz

Table 2.4 Overall Comparison

The FEA has the highest accuracy compared to the other models, but it requires a very high level of details. It is not convenient to have this requirement in the early design phase. Moreover, this makes the method sensitive to a tiny change, especially in the upper frequency. Therefore, it can only be used for frequencies between 0-250 Hz [29]. This limitation is not convenient because the human ear is not so sensitive to the sound wave in this range.

Another mathematical model is SEA. It has quicker calculation time and requires fewer details than FEA. However, the result from the SEA is only accurate starting from the high-frequency region (>500 Hz) because it treats the problem in a general way with considering all the situations. It is also very sensitive to the subsystem power input. If the excitation is inaccurate, then the result will not be reliable. Therefore, whenever possible, it is advisory to use measurement results for the input [30].

TNO cabin and SNAME have the same characteristic except for the frequency range of the result. Most of the acoustic room criteria, for instance, Noise Criteria (NC), and Room Criteria (RC) requires the noise level to be in between 31.5 – 8000 Hz. The small frequency range (31.5 Hz – 20000 Hz) of TNO Cabin means that it is not suitable anymore to comply it with the regulations. Moreover, the TNO Cabin also has less variety of equipment in its empirical model compare to SNAME Model. TNO Cabin assumes that the airborne noise source level in an engine room is only the main engine, neglecting other heavy equipment such as gearbox and electrical motor.

Based on this comparison, the SNAME is the most suitable model to use in this project. It does not rely on measurement to get the input and has a standard frequency range. SNAME has validated the accuracy of the empirical formula. Moreover, it is not necessary to have a high details information about the propulsion system which is convenient in the early design phase.

3

Research Approach

3.1 Overview

The model developed in this project is expected to have the capability to predict the noise level from various propulsion system configurations. Three main components are at the core of this research, i.e., (1) The acoustical model (2) The equipment sizing model (3) The evaluation methodology. The first one is used to predict the noise of a propulsion system and has been developed outside of this project. The second one is required because the noise absorption depends on the size of the compartment, which depends on the dimensions of the equipment inside it. The last one is the heart of this research and uses the first two to determine the effect of a certain parameter/design choice on the noise characteristic of propulsion systems.

The acoustical model that is implemented here is the SNAME model based on the work of *Fischer et al.* [24] with the supplementary guide by *Fischer et al.* [31] (see subchapter 3.2). As explained in the previous chapter, the SNAME model is a complete and compact tool to predict noise on board the ship. It can calculate the sources levels which can be converted to the sound experienced by the receiver in a particular location. At the very beginning, one has to calculate the noise sources levels. It is essential to define which sources are being considered in this thesis. The focus will be on the on-board power systems of the ships; therefore, the equipment being considered is limited to that. The propulsion driver equipment includes the diesel engine, the electric motor and the reduction gear. Furthermore, the considered auxiliary machinery is only the diesel generator set because it provides a significant amount of noise and vibration from its prime mover. Afterwards, the types of structures and the types of its materials should be chosen to determine the transmission loss from the source to the receiver location. Finally, the dimensions of the room will affect the airborne noise and the structure-borne noise level in the receiver location.

The noise level at the receiver location is a function of the room dimensions. The transmission path for both the airborne and structure-borne noise depends on the length, width and height of the compartment. Therefore, it is required to develop the engine sizing model. The equipment sizing models for the diesel engine and electric motor have used the work done by *De Vos et al.* [32] (see subchapter 3.5). For the diesel engine sizing, the regression analysis to determine the primary element of the equipment is developed in this project. Furthermore, the reduction gear sizing model is also developed in this thesis based on the relation between the main features of the equipment. The method to predict the engine compartment dimensions is created here too.

Based on the noise model, the relevant early-design choices are made. These options are the transmission types, the number of engines, the number of shafts, the number of compartments and the loading points. Although the last one is not an early-design choice, it is an important parameter that affects the noise level of the propulsion system. The constraints for the early design choices also need to be defined in order to limit the possible combinations of propulsion system configurations. Furthermore, the ship requirements need to be determined as well. It consists of the total installed power, the propulsion power and the auxiliary power. The early design choices and the ship requirements will determine the sets of propulsion system configurations that are being analysed.

Lastly, the evaluation methodology is established to investigate the influence of early design choices to the noise level of the propulsion system (see subchapter 3.6). This can be done by keeping one option varies and the other constant. It is expected each of the parameters will have different behaviour in affecting the noise level. The relation between the parameters is also being defined in the methodology.

In general, this project attempts to achieve the following outputs:

1. ***Identify the characteristics of every major noise source in a propulsion system.***
2. ***Determine the Airborne and Structure-borne Noise level of a propulsion system configuration.***
3. ***Analyse the effect(s) of every parameter on the noise generated by a propulsion system.***
4. ***Investigate how the chosen parameters interact with each other in the noise calculation.***
5. ***Quantify the influence of early design choices toward the noise of a propulsion system.***

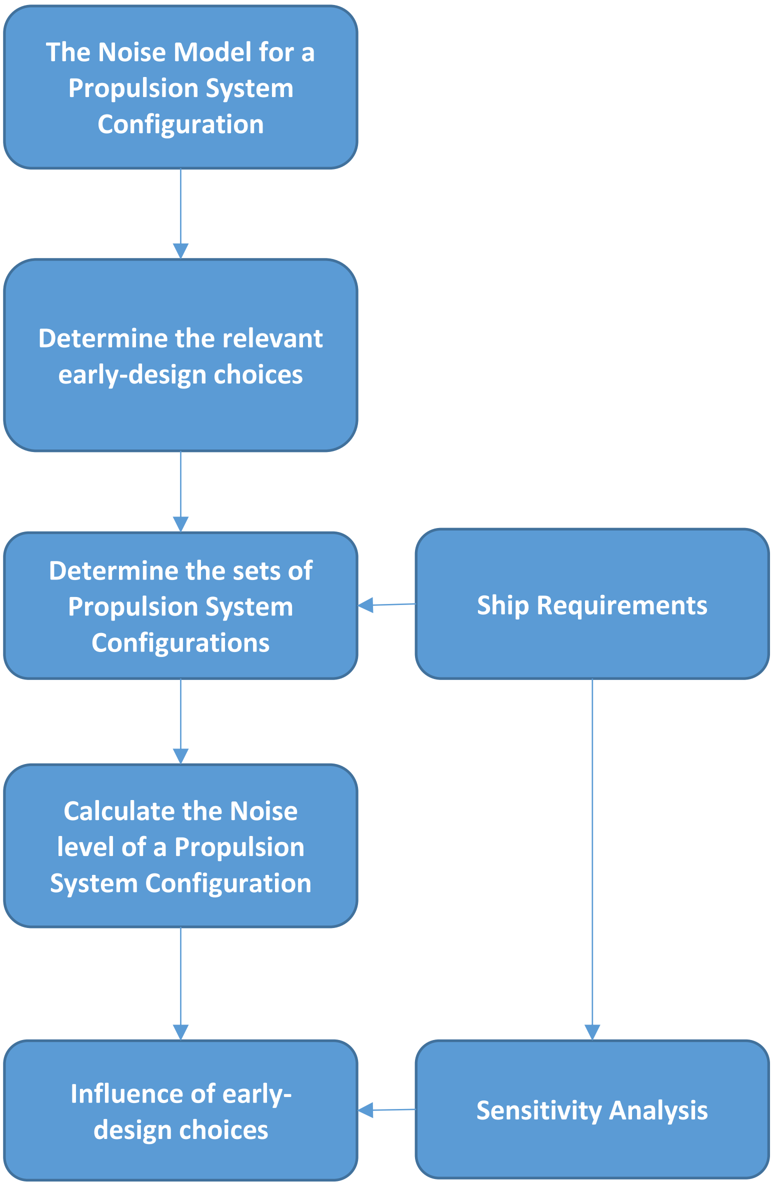


Figure 3.1 Research Approach Overview

3.2 SNAME Model

3.2.1 Airborne and Structure-Borne Noise Sources Level

Diesel Engine

The diesel engine is one of the significant noise sources on-board of a ship. The casing will transmit the vibration energy from inside it to the surrounding air. There is much literature exists on how to calculate the sound power level of the diesel engine casing [24] [33] [31]. The equation from Heckl is as follows [31],

For a diesel engine with the nominal speed over 700 rpm :

$$L_{w,de\_casing} = 57 + 10 * \log\left[\frac{N_{nom} * P_{nom} * \left(1 + \frac{P_{nom}}{m}\right)}{\frac{f}{1000} + \frac{1000}{f}}\right] + 20 * \log\frac{N_{act}}{N_{nom}} \tag{3.1}$$

Where N_{nom} is the nominal speed of the engine in rpm, P_{nom} is the nominal power in kW, m is the mass of the engine in kg, f is the octave band center frequency and N_{act} is the actual speed of the engine.

For a diesel engine with the nominal speed below 700 rpm :

$$L_{w,de\_casing} = 3 + 4.5 * \log(P_{nom}) + A * [5.5 * \log(P_{nom}) + 10 * \log(N_{nom}) - 43] + B \tag{3.2}$$

A and B are coefficients which are different for every center frequency [31],

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
A	1.6	1.5	1.4	1.3	1.2	0.8	0.8	0.8	0.8
B	93	96	98	100	102	102	96	91	86

Table 3.1 A and B Coefficients

Compared to another empirical formula [24] [33] ; the Heckl one is more accurate especially when determining the noise level of diesel engine that operates at lower than the nominal rpm [31]. The intake noise of the engine can also be a primary source if it is un-ducted. In this analysis, it is assumed that the intake is always ducted.

The vibration level equation of a diesel engine is also taken from Heckl. It derives the structure-borne noise value of the hard-mounted diesel engine. The formula from Heckl can be used as the vibration level on the foundation. One should calculate L_{aB1} & L_{aB2} first and combine those values to determine the final L_{ab}. The equation of L_{aB1} is as follows,

$$L_{aB1} = 55 + 5.5 * \log(P_{nom}) + 10 * \log(N_{nom}) + 30 * \log(\frac{N_{act}}{N_{nom}}) + \Delta \tag{3.3}$$

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Δ	-17	-19	-20	-21	-24	-28	-34	-43	-50

Table 3.2 Octave Band Adjustments Values for L_{aB1}

The L_{aB2} can be expressed as,

$$L_{aB2} = A + B * \log(P_{nom}/1000) \tag{3.4}$$

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
100-250 rpm									
A	74	73	73	63	63	60	58	56	54
B	4	3	2	10	7	6	1	1	1
250-600 rpm									
A	86	82	80	78	80	75	69	64	59
B	1	3	3	8	6	5	1	1	1
600-1000 rpm									
A	91	86	85	83	85	83	79	78	77
B	6	6	8.5	5	7	4.5	4	3	2

Table 3.3 A and B Coefficients for L_{aB2}

Finally, the L_{aB} is equal to,

$$L_{aB} = 0.5 * (L_{aB1} + L_{aB2}) + \Delta \tag{3.5}$$

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Δ	-17	-19	-20	-21	-24	-28	-34	-43	-50

Table 3.4 Octave Band Adjustments Values for L_{aB}

Reduction Gear

The sound power level from a gearbox depends on the manufacturing tolerance of the gears. The higher the machining precision, the lower the noise will be. The relation between them can be written as [31],

$$L_{w,gearbox} = 68 + 10 * \log(P_{in}) - 10 * \log\left(\frac{f_m}{f} + \frac{f^2}{f_m^2}\right) + \Delta \tag{3.6}$$

Where P_{in} is the input power to the gearbox in kW, f is the octave band center frequency f_m is the meshing frequency and Δ is tolerance adjustment according to the gear class. The latter one presented by the table below [31].

Class	B3	C1	C2	C3	D1	D2	D3
Δ_{tol}	0	2.5	5	7.5	10	12.5	15

Table 3.5 Gear Manufacturing Tolerance Adjustment Values

The acceleration level of the gearbox on the foundation also depends on the manufacturing tolerance. The acceleration level can be expressed as follows,

$$L_{aB,gearbox} = 67 + 10 * \log(P_{nom}) - 20 * \log(1 + R_h) + \Delta_{tol} + \Delta_2 + \Delta_3 \tag{3.7}$$

Where R_h is the radius of the gear housing in metres and Δ_3 is an additional 5 dB if the gear mesh frequency coincides with one of the frequency from the octave band.

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Δ_2	-25	-16	-18	-9	-1	3	8	7	0

Table 3.6 Octave Band Adjustments Values for $L_{aB,gearbox}$

All the gearboxes used in this thesis are assumed to have the same class which is B3 class.

Generator

It is quite hard to measure the sound power level from the generator. This problem occurs mainly because its associated prime mover is masking the noise from the generator. SNAME has a very rough approximation to calculate the noise generated by the generator [24].

$$L_{w,generator} = 34 + 10 * \log(P_{nom}) + 7 * \log(N_{nom}) + \Delta \tag{3.8}$$

Where P_{nom} is the rated power and N_{nom} is the rated speed. Afterwards, the value from the formula above should be added to the adjustment value to get the octave band airborne noise source levels [24].

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Δ	8	11	12	13	13	10	8	5	0

Table 3.7 Adjustment Values for Generator

The SNAME model assumed that at low frequencies the noise generated by the generator is dominated by its rotation frequencies, pole passage frequencies, and low-order harmonics. [24].

The electrical generator also uses the nominal power and speed of the equipment to calculate the structure-borne noise baseline level. It is a very conservative value too since the contributions of the prime mover contaminate the measurement data. The equation is as follows,

$$L_{aB,generator} = 42 + 10 * \log(P_{nom}) + 7 * \log(N_{nom}) \tag{3.9}$$

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Δ	-17	-19	-20	-21	-24	-28	-34	-43	-50

Table 3.8 Octave Band Adjustments Values for $L_{aB,generator}$

Electrical Motor

The electric motor can produce a high level of noise in the machinery space, especially for totally enclosed fan-cooled motors (TEFC). The high airborne noise from electric motor happens due to the aerodynamic noise from the cooling fan. The drip-proof motor will have a lower noise compare to the TEFC since it uses a heat exchanger instead of a fan to prevent the overheating of the machine. The sound power level of TEFC motor can be expressed as [24],

$$L_{w,electric\_motor\_TEFC} = 5 + 13 * \log(P_{nom}) + 15 * log\ (N_{nom}) \tag{3.10}$$

Where P_{nom} is the rated power in hp and N_{nom} is the nominal speed of the electric motor. The drip proof motor source level should be reduced by 10 dB since it generally has lower noise compare to TEFC motor.

The octave band levels are obtained by adding the adjustments values to the source levels. The table below provides the values for AC and DC motor [24].

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
AC Motor	5	6	10	14	15	15	14	8	1
DC Motor	0	0	5	10	15	15	14	8	1

Table 3.9 Adjustments Values for AC and DC Electric Motor

The vibration level of the electrical motor from the SNAME is based on a small amount of database. In that sense, it is hard to derive a semi-empirical formula to predict the frequency-dependent structure-borne noise source level. The model only gives an “envelope value” in which most of the electric motors will lie. As a consequence, if the electric motors have different installed power, then the vibration level is going to be the same which is not plausible. An alternative solution to this problem is to use the generator structure-borne noise equation to calculate the electrical motor vibration level. This simplification can be done since both of them have similar characteristics. All the electrical motors in the analysis are assumed to be AC motor and have a drip-proof casing.

3.2.2 Airborne Noise Transmission Path in a Room

Room Constant

Room constant is the amount of sound absorption by boundary surfaces in a compartment. The absorption coefficient depends on the material. In SNAME model, Sabine absorption coefficients are used to calculate the room constant. If a compartment consists of different materials, then the room constant is a sum of several terms. The expression for this is as follows,

$$R = S_1 * \alpha_1 + S_2 * \alpha_2 + S_3 * \alpha_3 + \cdots + S_n * \alpha_n \tag{3.11}$$

Where R is the room constant, S_1 is the first boundary surface, α_1 is the Sabine absorption coefficient for the first boundary surface and n is the number of individual boundary surfaces in a compartment. If the compartment has the same surfaces the equation above can be simplified into,

$$R = S_{tot} * \alpha \tag{3.12}$$

Where S_{tot} is the total surface area of the compartment and α is the Sabine absorption coefficient. When the information of boundary surface material is not available, it can be either hard surface or soft surface. Approximate values of the Sabine absorption coefficients are presented in the table below.

f(Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Hard Surfaces (α_h)	0.1	0.1	0.09	0.05	0.02	0.01	0.01	0.01	0.01
Soft Surfaces (α_s)	0.1	0.2	0.25	0.4	0.6	0.7	0.7	0.6	0.5

Table 3.10 Absorption Coefficients for Soft and Hard Surfaces

The complete list of Sabine coefficients can be found in the literature (ref).

The SNAME model also takes into account the absorption by the non-boundary surfaces such as machinery, furniture and among others. In order to do that, one must determine the non-boundary surface area first. It is relatively a simple task because the model uses correction factors to calculate it, which can be written as follows,

$$S_{nb\_hard} = S_{tot} * \beta_{hard} \tag{3.13}$$

$$S_{nb\_soft} = S_{tot} * \beta_{soft} \tag{3.14}$$

Where S_{nb_hard} , S_{nb_soft} is the area of hard and soft non boundary surfaces and β_{hard} , β_{soft} is the hard and the soft correction factors respectively. The value for the latter one depends on the types of compartments being investigated.

Compartment	β_{hard}	β_{soft}
Lounge, Wardroom, Officers' Mess	0	0.1
Crew's Mess, Offices	0.2	0
Berthing	0	0.2
Main Engine Room	0.5	0.2
Auxiliary Machinery Room	0.4	0.2
Secondary Auxiliary Machinery Room	0.3	0.1

Table 3.11 Non-Boundary Surfaces Correction Area

Afterwards, the non-boundary surfaces area need to multiply by the respective hard and soft absorption coefficients (α_h and α_s) to get the contributions of non-boundary surfaces to the overall room constant.

$$R_{nb\_hard} = S_{nb\_hard} * \alpha_h \tag{3.15}$$

$$R_{nb\_soft} = S_{nb\_soft} * \alpha_s \tag{3.16}$$

Therefore, the total room constant can be expressed as,

$$R_{tot} = R + R_{nb\_hard} + R_{nb\_soft} \tag{3.17}$$

Where R_{tot} is the total room constant of a compartment.

Reverberant Field

The walls and other objects can reflect the sound wave in a room if the walls have no or very low absorption coefficients [14]. The consequence of this phenomenon is that the sound pressure level received by the receiver far away from the source is equal to the sound pressure level close to the source. In practice, the room without absorption coefficient at all, typically called reverberation chamber, is used to measure the sound power level of a noisy machine.

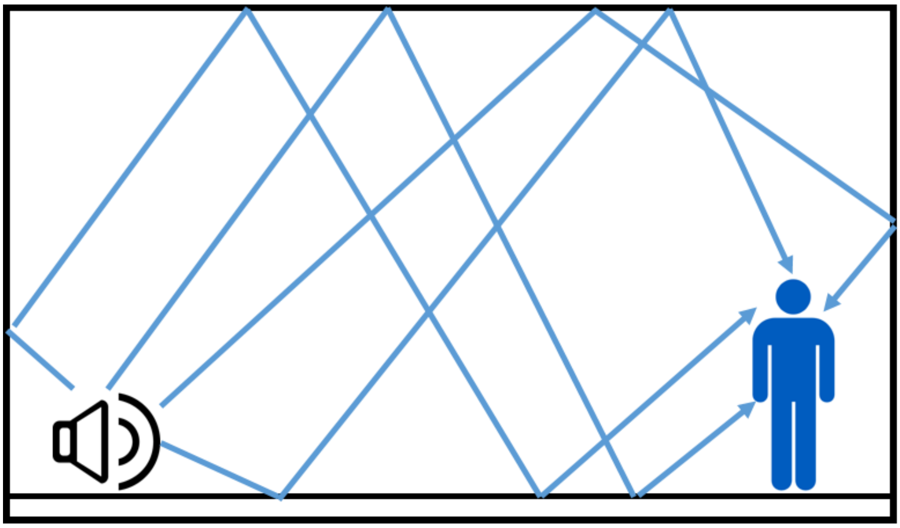


Figure 3.2 Sound Wave in Reverberant Field

In ship structures, most of the compartments are built from steel. Based on Sabine absorption coefficients, the steel has minimal absorption value. It means that the reverberant field contributions dominate the total sound pressure level in most of the shipboard spaces. Nevertheless, there is an exception when the compartment has an excellent acoustical treatment, or the receiver is located very close to the noise source. The following equation is used to calculate the reverberant sound pressure level in SNAME model,

$$L_{p,rev} = L_w - 10 * \log(R_{tot}) + 16 \tag{3.18}$$

Where R_{tot} is the total room constant value which can be computed by Equation 3.17. If there are multiple noise sources in the room, L_w is the total sound power lever from the log summation of every noise source.

Direct Field

When the sound sources radiate noise directly to the receiver without encountering any acoustical treatment, it is known as a direct field sound pressure level. The direct sound is a function of distance and source directivity factor. If the receiver is far away from the sources, then the contributions from the direct sound will become less critical.

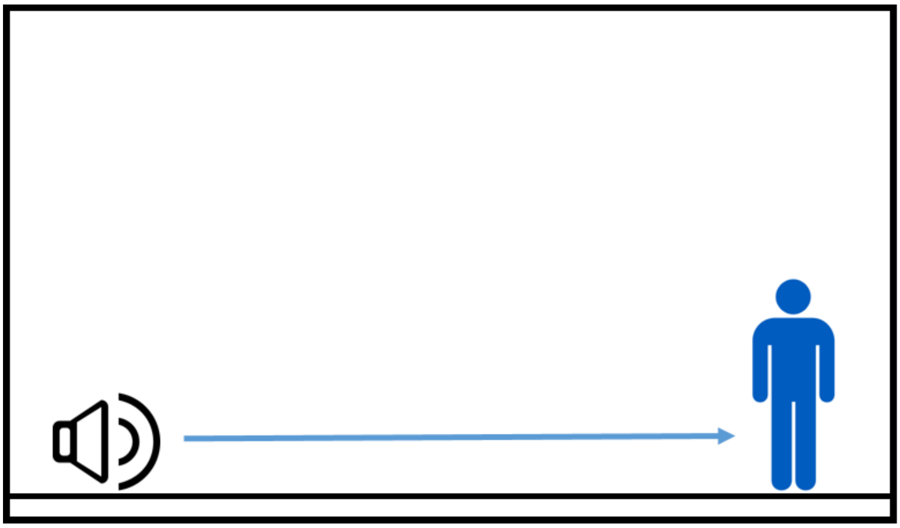


Figure 3.3 Sound Wave in Direct Field

The direct sound requires the distance between the noise source and the receiver, and the distance between the receiver to the compartment boundaries to calculate its level. The following equation used to estimate the direct field sound pressure level.

$$L_{p,dir} = L_w - 20 * \log(r) + 10 * \log(Q) - 1 \tag{3.19}$$

Where r is the distance between the center point of the noise source to the receiver location in feet and Q is the source directivity factor.

In a large acoustic space where a significant volume occupied by the furniture and fittings, SNAME found that the direct field sound pressure level with the distance over 10 feet from noise sources tend to decrease at a higher rate. In that sense, for calculating the direct field sound pressure level at a range larger than 10 ft, one should use the following expression,

$$L_{p,dir} = L_w - 30 * \log(r) + 10 * \log(Q) + 9 \tag{3.20}$$

If there is more than one noise source inside the room, the total direct field sound pressure level can be calculated by combining the individual sound pressure level contribution. Note that every source can have a different distance to the receiver.

3.2.3 Total Sound Pressure Levels

The total sound pressure level experienced by the receiver at a specific location in a compartment containing noise sources is equal to the summation of the octave band sound pressure level from the reverberant field and direct field. Firstly, one should calculate the reverberant field of the noise sources. Secondly, the direct field sound pressure level for every noise source needs to be computed. Lastly, do the logarithmic summation of the reverberant and the direct sound pressure level. If the direct field contribution of a source is 10 dB lower than the reverberant one, it is possible to neglect the direct field contribution from the summation. The overall process illustrated in more detailed in the figure below.

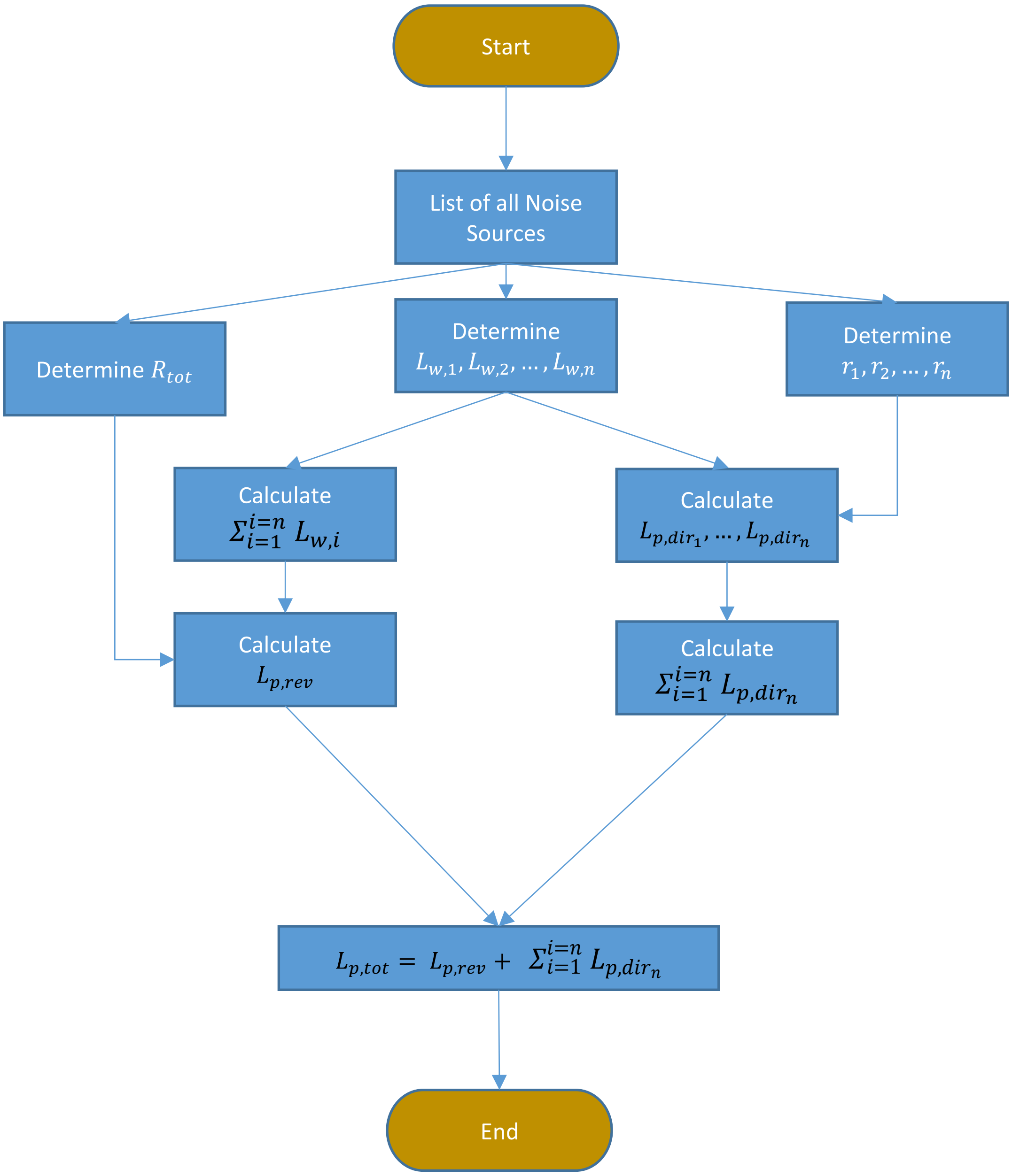


Figure 3.4 Overall process to determine total sound pressure level at the receiver location

3.2.4 Structure-borne Noise Transmission Path

Mounting Attachments

In order to get the structure-borne noise level on the foundation, one needs to subtract the free acceleration level of the source with the transfer functions of the machinery mounting. The transfer functions are obtained by reducing the measured acceleration on the engine to the measured vibration level below the mounts (on the top of the machinery foundation).

The SNAME model categorises the transfer function of each mounting types by the weight of the equipment and the foundation type. Class 1 includes sources weighing less than 1000 lb, Class 2 is for sources weighing between 1000 lb and 10000 lb, and Class 3 includes sources that weigh over 10000 lb. For the foundation, there are two types of foundations, type A and B. The first one is a foundation that is relatively light, with a beam less than 6 inches in web depth while the latter one is a rigid and massive structure. The table below shows the weight classes and foundation types of the noise sources.

Noise Sources	Weight Class	Foundation Type
Diesel Engine	3	B
Diesel Generator	3	B
Main Reduction Gear	3	B
Motor	1,2 or 3	A/B

Table 3.12 Weight Classes and Foundation Types for the Noise Sources

There are four types of mountings that are usually used in the ship structure, i.e., (1) Hard mounting, (2) High-frequency isolation mounts, (3) Low-frequency isolation mounts and (4) Two-stages mounting.

Hard mounting is when the machine is directly connected to its foundation. The foundation will limit the vibration of the engine. Although this is the most economical method to mount an engine, the transfer function values are quite low. High-frequency isolation mounts use the distributed isolation material pads to reduce the vibration level transmission. The pads usually have a thickness less than 1 inch. Therefore, it is not necessary to provide extra height to the engine room when using this kind of mounting. The low-frequency isolation mount is one of the most effective ways to reduce the transmission of vibration level from the vibrating machinery to the foundation. The natural frequency of the mounting should be lower than the first natural frequency of the engine. For shipboard installations, the typical value for the mounting natural frequency is less than 15 Hz. The two-stages isolation system is similar to low-frequency mounting, but there is an additional structure between the foundation and the engine. This structure is usually called “intermediate raft”. In this arrangement, the engine is resiliently mounted to the raft, in which the raft is also resiliently mounted to the foundation. A very high transfer function values can be found in the two-stage mounting system. A typical two-stage mounting in the shipboard installations will have the natural frequencies less than 30 Hz. All the transfer function for every type of mounting can be found in the appendix C.1.

Machinery Foundations

There are two types of machinery foundations categorised by SNAME, the plate-beam foundations and the pipe machinery foundations. The first one is used for the foundation of medium to heavy weight machinery, and the latter one is used for the light and medium machinery.

The transmission losses values are taken from the difference between the vibration levels on the foundation top to the deck plate around the foundation. The experimental data done by SNAME showed that the vibration level of the deck plate around the foundation is higher than the level on the foundation top. In that sense, the transmission losses values will be negative for both foundation types. This amplification is because the plates are excited at their resonant frequencies. For the higher frequencies, the transmission losses will be higher since the resonant effect is decreased. If the engine is hard-mounted, there is no need to include the transmission losses induced by the machinery foundations.

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Plate-beam Foundation	-10	-13	-13	-10	-7	-5	-2	0	0
Pipe foundation	-8	-11	-11	-8	-5	-5	-5	-1	-1

Table 3.13 Transmission losses by machinery foundation

Ship Structures

The SNAME model categorises three losses by ship structures when the structure-borne noise travels along the structures to the receiver location. The first one is the effective source area. It is used to determine the vibration level at the bulkheads or the boundaries surfaces of the machinery compartment. The transmission losses within the source compartment can be expressed as follows,

$$\Delta L_a = 10 * \log \frac{(r_1 + r_2) * (1 - 0.35 * \varepsilon_r)}{(a + 2 * r_{fr}) * (1 - 0.35 * \varepsilon_0)} \qquad (3.2120)$$

Where r_1 and r_2 are the distances from the end of foundation to the center of a certain boundary surface and r_{fr} is the minimum frame spacing.

$$\varepsilon_0 = \frac{(a - b) * (a + b + 4 * r_{fr})}{a + 2 * r_{fr}} \qquad (3.22)$$

$$\varepsilon_r = \frac{(a - b) * (a + b + 4 * r_{fr})}{(r_1 + r_2)^2} \qquad (3.23)$$

Where a is the length of foundation and b is the width of foundation.

The second type is the transmission loss beyond the effective source area. The vibration level dissipates when travelling through the ship structure along the path from the source to the receiver. This transmission loss is a summation of the damping loss by hull structure and the intersection loss. Moreover, it also depends on the size of the structure, the distance from the source to the location of interest and the intersection between the source and structure interest. The equation to predict the damping loss is as follows,

$$\Delta_{Ld} = 0.126 * \sqrt{f} * \sum_{N} \frac{l_i * \eta_i}{\sqrt{t_i}} \tag{3.24}$$

Where f is the frequency of interest in Hz, N is the number of “structures” between the source compartment and the receiver location, l_i is the length of the i^{th} structures in feet, η_i is the loss factor of the i^{th} structures, t_i is the thickness of the plate and i is the order of the structure. The elements that can be considered as “structures” are the deck, bulkhead, shell side and the shell side between the adjacent intersections with other deck, bulkheads, or shell sides. One should take the least number of structure possible for the structure-borne noise propagation path (the shortest path). The values of hull loss factor can be found in appendix C.2

The last one is the transmission loss due to the intersection of ship structures. It creates discontinuities in the structure-borne noise transmission path. The more intersections along the path to the receiver location, the lower the vibration level in that location. The intersection loss is independent of the frequency which means it has the same value over the octave band frequency. The following equation can be used to calculate the total transmission loss by the intersection of ship structures,

$$TL_{inter} = \sum_{N} \frac{TL_n}{\sqrt{n}} \tag{3.25}$$

Where N is the number of connections between the source compartment and the location of interest, TL_n is the transmission loss for the n^{th} intersection and n is the connection number, for instance, $n = 1$ for the first intersection, $n = 2$ for the second intersection and $n = N$ for the last intersection. If the distance between the source and the receiver is really far that makes N exceeds five, the structure-borne noise becomes insignificant.

$$TL_n = 10 * \log \frac{1}{T_{ij}} \tag{3.26}$$

Where T_{ij} is equal to,

$$T_{ij} = 2 * \frac{\dfrac{\rho_j * t_j^{2.5}}{\rho_i * t_i^{2.5}}}{\left[1 + \sum_{k=2}^{k=M} \dfrac{\rho_k * t_k^{2.5}}{\rho_i * t_i^{2.5}}\right]} \tag{3.27}$$

ρ is the density of the plates, t is the metal thickness of the plates, i is the subscript of the plate with oncoming energy, j is the subscript of the plate that receiving energy. M value is depend on the type of the connection, for ‘L’ connections $M = 2$, ‘T’ connections M =3 and the cross connections $M = 4$.

3.2.5 Total Acceleration Levels

The calculation of vibration levels received at the location of interest is more complicated than the transmission of airborne noise. Four transmission losses need to be considered. In general, the steps to predict the vibration level at the receiver location can be illustrated by the figure below.

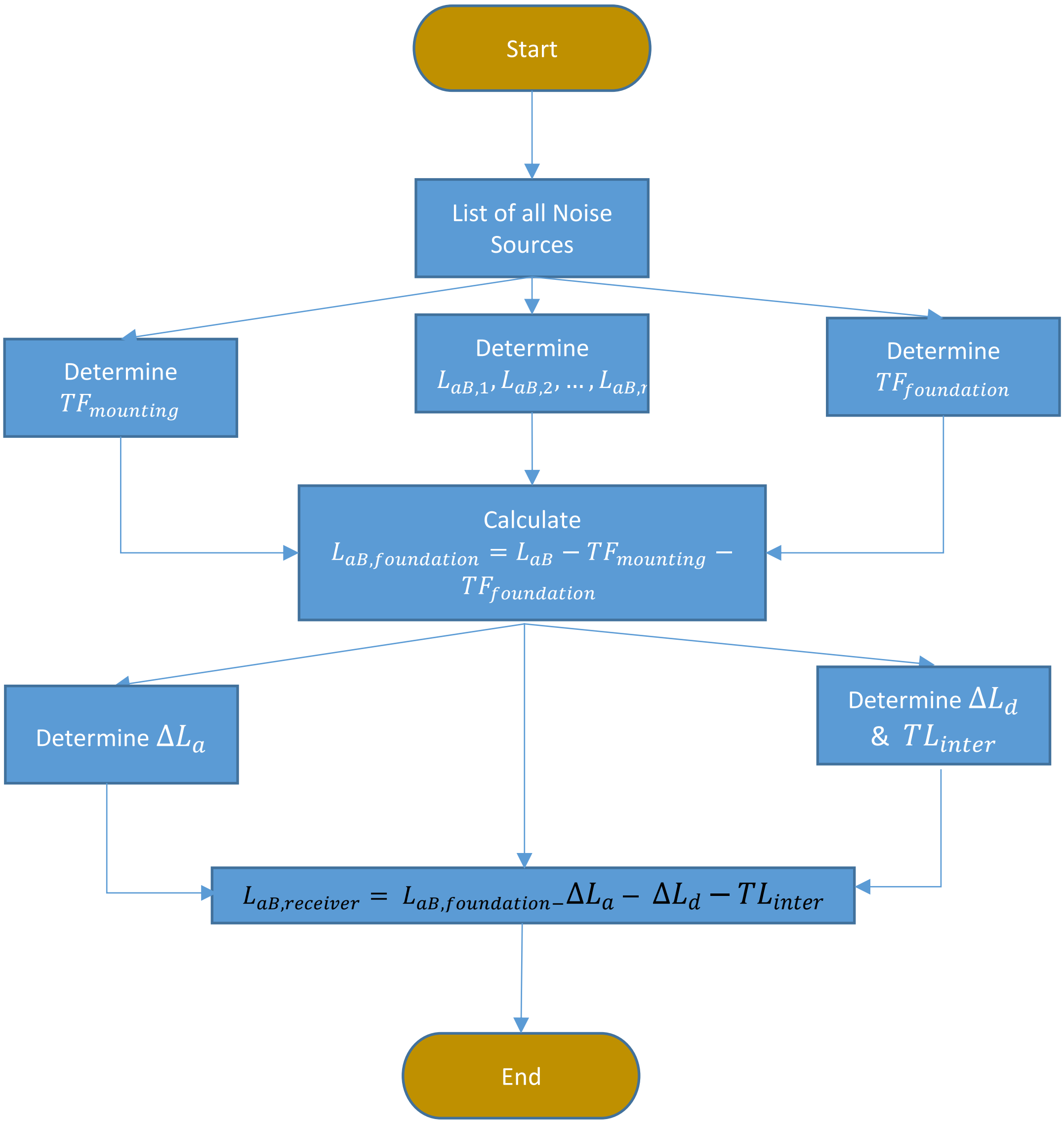


Figure 3.5 Overall process to determine the acceleration levels at the receiver location

3.3 Compartment Noise Measure

For every calculation, the compartment noise source levels will be computed. The results of these compartment noise source levels will be in the 1/1 octave band frequency. Therefore, it has n rows and nine columns in which per column represent the sound level per frequency. The values per row can be transformed to an equivalent value to make it comparable with other variables. The equivalent level here is an average value of sound over a range of frequency. The following formula shows how to get the equivalent sound pressure value,

$$L_{p,eq} = 10 * \log\left(\sum_{i=1}^{i=n} 10^{\frac{L_{p,i}}{10}}\right) \tag{3.28}$$

Where $L_{p,eq}$ is the equivalent sound pressure level and i represents the number of columns. The drawback of using this value is losing the detailed information in the frequency band. Nevertheless, it is not a very big concern since the compartment sound power level will have the same trend.

It is also necessary to convert the 1/1 octave band compartment sound pressure level to an equivalent A-weighted noise level. This conversion can be done by subtracting the nine-octave band values to the A-scale correction factors and combining those values into an equivalent one. This measure commonly used as the criteria for noise limit regulations [34]. The table below presents the A-scale correction factors.

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
A-weighting	-39.4	-26.2	-16.1	-8.6	-3.2	0	1.1	1	-1.1

Table 3.14 A-weighting adjustment values

Therefore, the A-weighted sound pressure level can be expressed as follows,

$$L_A = L_p + AV_a \tag{3.29}$$

Where L_A is the adjusted sound pressure level and AV_a is the A-weighting adjustment values. Afterwards, the equivalent value can be calculated by,

$$L_{A,eq} = 10 * \log\left(\sum_{i=1}^{i=n} 10^{\frac{L_{A,i}}{10}}\right) \tag{3.30}$$

3.4 Varying Input Parameters

There are two types of inputs to the model, the ship requirements, and the design choices of the propulsion systems. Each of them consists of unique parameters that will affect the outcome of the model. Some of these parameters have constraints simply due to the limitation of the existing technology and design plausibleness. The table below describes all of those in more details.

Types of Input	Parameters	Constraints
Ship Requirements	Total Installed Power	-
	Propulsion Power	-
	Auxiliary Power	-
Design Choices	Transmission Types	[DM,DE,DH, Battery]
	Number of Shafts	$N_{shaft} \leq 2$
	Number of Diesel Engines	$0 \leq N_{de} \leq 4$
	Number of Gearboxes	$0 \leq N_{gb} \leq 2$
	Number of Diesel Gensets	$0 \leq N_{dg} \leq 6$
	Number of Electric Motors	$0 \leq N_{em} \leq 2$
	Number of Compartments	$0 \leq N_{room} \leq 2$
Loading Points	Loading Points	$10\% \leq \gamma \leq 100\%$

Table 3.15 Types of Inputs and Parameters

The first parameter defines the installed power of the vessel. It is used as the rule of thumb for marine engineers to choose what type of engines and arrangements they would like to use. Ships with different purposes most likely will have different energy split between the auxiliary and the propulsion requirements, for instance, tankers and cruise ships. In tankers, the electrical load is evidently low compared to the cruise ships because it does not have a lot of passengers and no entertainment centre required in such vessel. In that sense, it would be obvious that on cruise ships one can find more diesel gensets or at least gensets with higher power than in tankers. It is vital to decide on the installed power of the vessel before going into details about the possible power configurations. The propulsion power and auxiliary power parameters can be determined arbitrarily. There is no limitation regarding those parameters. If a cargo ship is analysed, Deltamarin [35] provides a guideline on how to decide the propulsion and the auxiliary demand of the vessel.

The second type is the design choices one can have for the propulsion system configurations. The idea behind this input is that by changing the configuration, one can achieve lower sound levels generate by the propulsion systems. The effect of each parameter is going to be investigated by varying its entire range of values while keeping the others constant. The constraint for the number of diesel generator sets is chosen arbitrarily. It needs to be limited since theoretically it is possible to have an infinite amount of diesel generator sets in a ship. The last parameter is the loading points of the ship. This parameter used to determine the power demand in part load conditions. Although the loading point is not a design choice, the part load condition will affect the noise level of a propulsion system by changing the operation point of the equipment. It is important to note that this parameter will not modify the propulsion system arrangement.

3.4.1 .Ship Requirements

The necessary ship data required here is the total installed power of a vessel. It is the summation of the propulsion load and auxiliary load. This type of basic data used as the basis of the plant concept.

$$P_{tot} = P_{prop} + P_{aux} \tag{3.31}$$

If it is not possible to determine the auxiliary load yet since it is still in early design phase, one can use the method from Deltamarin guide [35] for a general cargo vessel. The nominal auxiliary load is as follows,

$$P_{aux,0} = 120 + 0.8 * P_B^{0.65} \tag{3.32}$$

Where P_B is the brake power of the engines. The auxiliary load when maneuvering can be expressed as,

$$P_{aux,man} = 1.3 * P_{aux,0} \tag{3.33}$$

The required power for main engine auxiliaries,

$$P_{aux,ME} = 0.07 * P_B \tag{3.34}$$

Note that these empirical formulas are specifically for a general cargo vessel, if the ship type being investigated has entirely different characteristics, then these values will be invalid.

3.4.2 Transmission Types

There are three main transmission types to deliver the power from the prime movers to the propeller, i.e., mechanical transmission, electrical transmission, and hybrid transmission. In order to know what kind of machinery that a propulsion system configuration requires, the engineer should decide on the transmission type first. In the diesel mechanical plant and the diesel hybrid plant, the prime mover will deliver the power to the propeller via mechanical-shaft while in the diesel-electric plant a cable connection will replace the shaft.

Mechanical Transmission

The diesel-mechanical plant consists of diesel engine(s) and reduction gear(s) to deliver power to the propeller. The diesel-generator set will provide the auxiliary load. It is not possible to do the shared load since there is no connection between the diesel engine and the diesel-generator set. The figure below illustrates the diesel-mechanical plant.

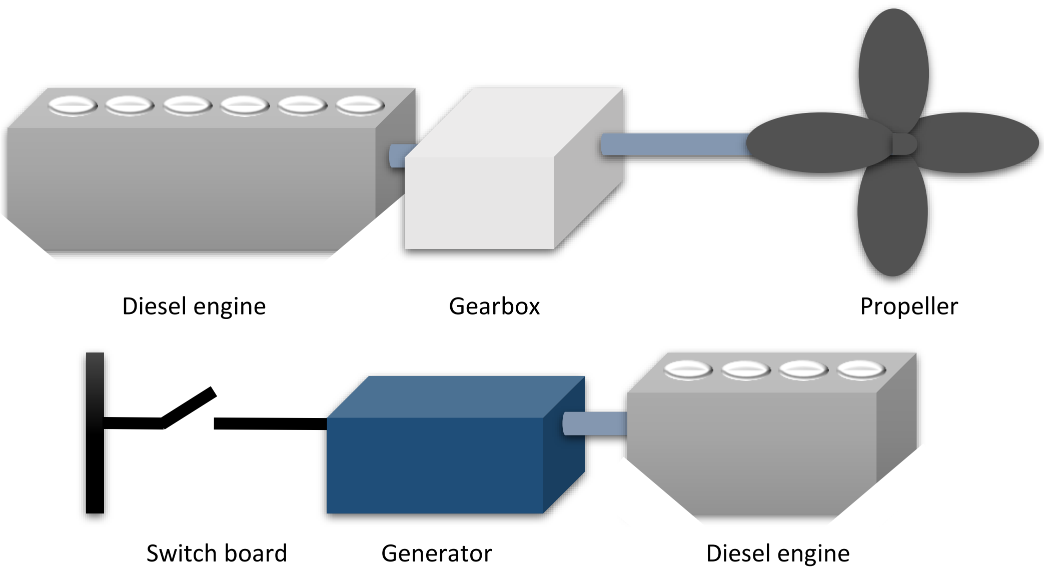


Figure 3.6 Basic Diesel-Mechanical Plant Configuration

Electrical Transmission

In the diesel-electrical plant, the connection between the propeller and the prime mover will use a cable and an electric motor instead of using a shaft and a gearbox. The main diesel-generator set produces electricity for the electric motor. The auxiliary diesel-generator set will provide the electrical load.

There are two types of diesel-electrical plants, i.e., (1) Conventional Electric Propulsion, and (2) Integrated Electric Propulsion (IEP). It is not possible to do the shared load if the plant uses the first option. In conventional electric propulsion, the power from the main diesel-generator set is “devoted” only to the propulsion system meaning it is not possible to use it for providing power to the non-propulsion system. On the other hand, the IEP system gives a possibility to distribute the power for both the ship propulsion and the electrical load ship.

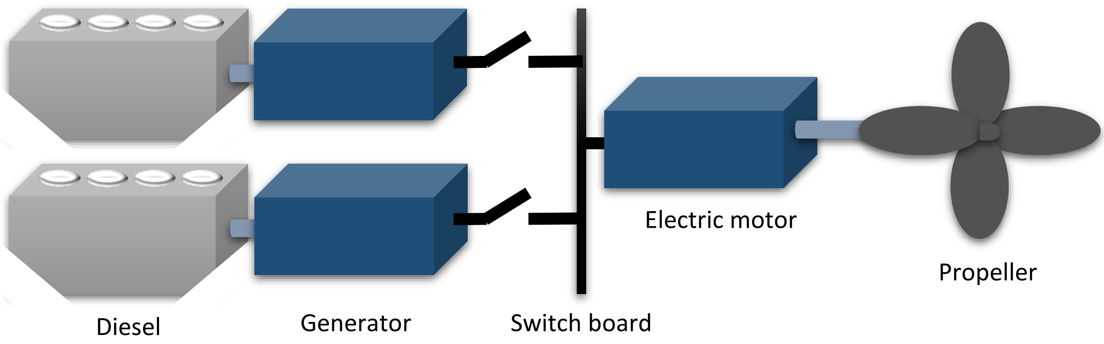


Figure 3.7 Basic Diesel-Electrical Plant Configuration

Hybrid Transmission

The diesel hybrid plant is a combination of the diesel-mechanical and the diesel-electrical plant. The configuration consists of diesel engine(s) and gearbox(s) to provide power to the propeller. In addition, the gearbox is also connected to the electric motor which has two different operation modes, i.e., (1) generating electricity and (2) providing propulsion power for booster mode. The first one has known as Power Take-Off (PTO) modes and the latter known as Power Take Input (PTI) modes. The diesel-generator set will provide the auxiliary load during PTO modes. In PTI modes, diesel-generator sets will drive the electric motor. When the propulsion power is supplied only by the diesel generator set, it is known as the Power Take Home (PTH) modes.

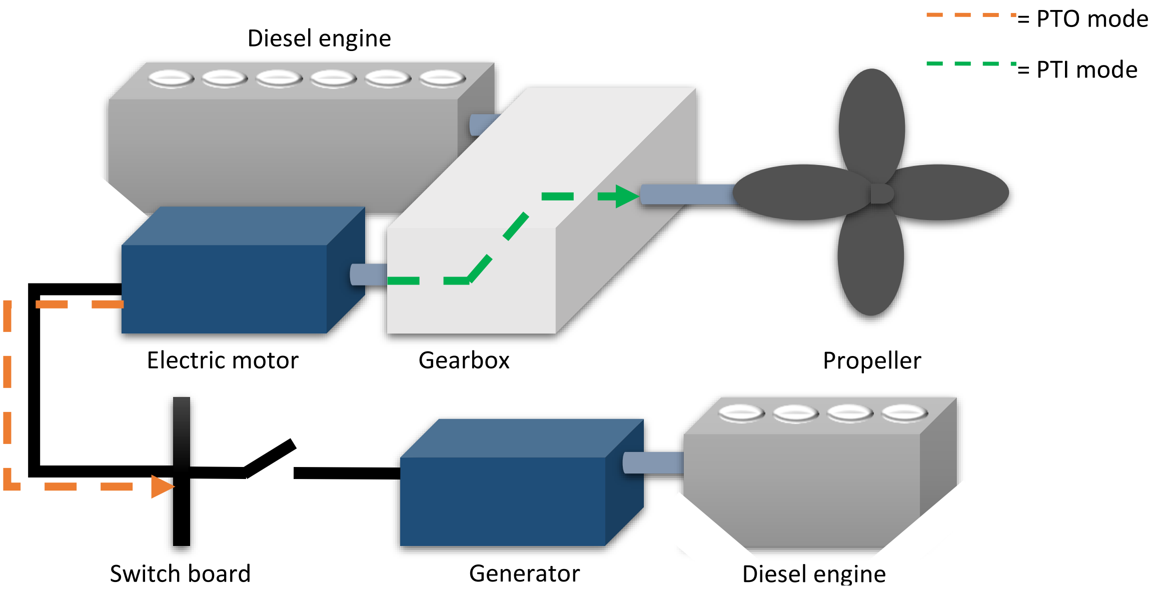


Figure 3.8 Basic Diesel-Hybrid Plant Configuration

The table below shows the summary of the propulsion system with its main equipment. The capabilities of each arrangement to operate in different modes are also provided here.

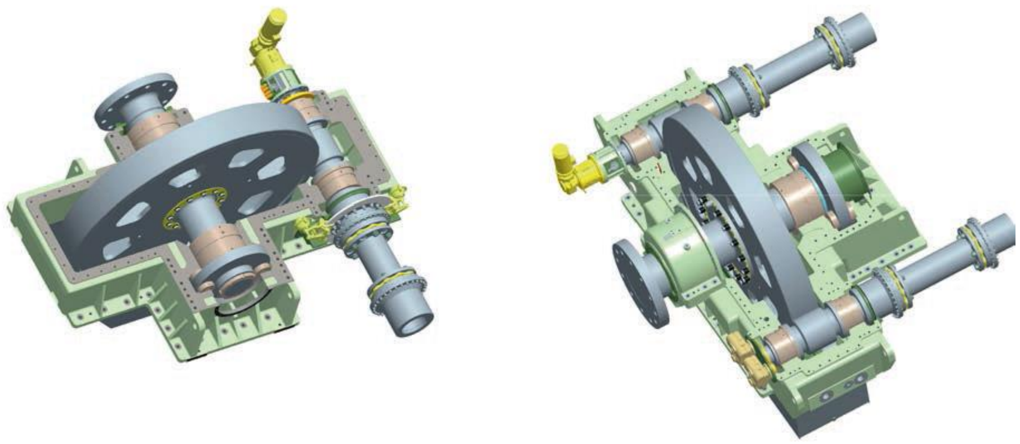
	Machinery				Operation		
	Diesel Engine	Reduction Gear	Electrical Motor	Diesel - Genset	PTO	PTI	PTH
DM	✓	✓	×	✓	×	×	×
DE	✓	×	✓	✓	×	×	✓
DH	✓	✓	✓	✓	✓	✓	✓

Table 3.16 Propulsion System Configurations main equipment and operation modes

3.4.3 Number of Shafts and Engines

The number of engines is the general term for the number of diesel engines, number of diesel generator sets, number of electric motors and number of gearboxes while the number of shafts defines the number of propeller shafts. The equipment required for the propulsion system is determined by choosing the transmission type. There are many possible propulsion system arrangements, for instance, a single shaft with a single engine, a single shaft with multiple engines, multiple shafts and so on. It is not practical to analyse all possible combinations. In that sense, the limitation must be made so that the number of combinations can be limited to a clear and finite set.

In a diesel-mechanical plant it is possible to have multiple engines in one shaft thanks to the Double Input Single Output (DISO) Gearbox. For the single shaft with a single engine, one can use the Single Input Single Output (SISO) Gearbox. An important note when using the DISO gearbox is that the engines should operate at the same speed when operating together.



(a) (b)

Figure 3.9 (a) Single Input Single Output (SISO) Gearbox (b) Double Input Single Output (DISO) Gearbox [36]

The maximum number of diesel engines with a single shaft is limited to two engines. It is not very common to have more than two diesel engines in one shaft due to the large dimensions of the gearbox. Therefore, such configuration is not considered in this analysis. Father-son configuration can be done, but they should operate at the same speed.

The diesel-electrical plant can have multiple diesel generator sets for one propeller. Theoretically, there is no limitation on the number of the diesel generator sets, thanks to the electrical motor and cable which replace the gearbox-shaft connection. However, from the Table 3.15 the maximum number of diesel gensets is set to a finite number. This constraint does not create a big consequence to the analysis since the important thing is to see if one can lower the sound produced by increasing the number of diesel gensets.

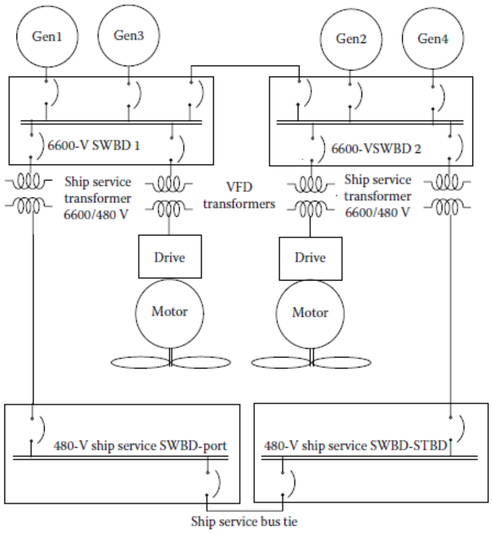


Figure 3.10 Example of IEP System Architecture. SWBD = Switch Board, STBD = Star Board [37]

The last one, the diesel-hybrid plant, is capable of having multiple diesel engines and electrical motors per shaft theoretically speaking. However, in this thesis, only one diesel engine and a single electric motor are allowable for one gearbox in a hybrid system. Again, it is because of the scarcity of a gearbox with more than two inputs. Therefore, the maximum number of diesel engines and the number of electric motors for a single shaft is one due to the gearbox limitation. It is not possible to use the SISO gearbox in this configuration.

From the above explanation, it is evident that these parameters, i.e.,(1) number of shafts (2) the nominal power, and (3) the number of the primary and auxiliary engines are related to each other. The summary of the relationship between them and the limitation of each transmission type can be seen in the table below.

Equipment		Single engine-Single shaft and Twin Shaft Plant		
		DM	DE	DH
Diesel Engine	$P_{b,de}$	$\frac{P_{tot} - P_{aux}}{N_{shaft}}$	–	$0 \le P_{b,de} \le P_{tot}$
	N_{de}	N_{shaft}	–	N_{shaft}
Diesel Genset	$P_{b,dg}$	$\frac{P_{tot} - P_{prop}}{N_{dg}}$	$\frac{P_{tot}}{N_{shaft}}$	$0 \le P_{b,dg} \le \frac{P_{tot}}{N_{dg}}$
	N_{dg}	$1 \le N_{dg} \le 6$	N_{shaft}	$0 \le N_{dg} \le 6$
Gearbox	$P_{b,gb}$	$\frac{P_{tot} - P_{aux}}{N_{shaft}}$	–	$\frac{P_{tot} - P_{aux}}{N_{shaft}}$
	$N_{gb,SISO}$	N_{shaft}	–	–
	$N_{gb,DISO}$	–	–	N_{shaft}
Electrical Motor	$P_{b,em}$	–	$\frac{P_{prop}}{N_{shaft}}$	$0 \le P_{b,em} \le \left\lvert \frac{P_{prop} - P_{b,de}}{N_{gb,DISO}} \right\rvert$
	N_{em}	–	N_{shaft}	$N_{gb,DISO}$

Table 3.17 Relation between number of shafts, nominal power and number of engines for the Single engine-Single Shaft Plant and the Twin Shaft Plant

Parameters		Multiple engines-Single shafts and Multiple engines-Multiple Shafts		
		DM	**DE**	**DH**
Diesel Engine	$P_{b,de}$	$\frac{P_{tot} - P_{aux}}{N_{de}}$	–	–
	N_{de}	$0 < N_{de} \leq 4$	–	–
Diesel Genset	$P_{b,dg}$	$\frac{P_{tot} - P_{prop}}{N_{dg}}$	$\frac{P_{tot}}{N_{dg}}$	–
	N_{dg}	$1 \leq N_{dg} \leq 6$	$1 \leq N_{dg} \leq 6$	–
Gearbox	$P_{b,gb}$	$\frac{P_{tot} - P_{aux}}{N_{shaft}}$	–	–
	$N_{gb,SISO}$	–	–	–
	$N_{gb,DISO}$	N_{shaft}	–	–
Electrical Motor	$P_{b,em}$	–	$\frac{P_{prop}}{N_{shaft}}$	–
	N_{em}	–	N_{shaft}	–

Table 3.18 Relation between number of shafts, nominal power and number of engines for the Multiple engines-Single shaft Plant and the Multiple engines-Multiple Shafts Plant

3.4.4 Loading Points

From the subchapter 3.4.1, one can know that the total installed power(P_{tot}) is equal to the summation of propeller load (P_{prop}) and auxiliary load (P_{aux}). All these values are maximal values. However, a ship not only operates at the nominal point but also at part load conditions which mean less than the P_{tot}. This can be expressed as follows,

$$P_{tot,part} = P_{prop} * \gamma_1 + P_{aux} * \gamma_2 \tag{3.35}$$

Where $P_{tot,part}$ is the part load total demand and γ is the loading factor. The latter one is a value of $0 \leq \gamma \leq 1$. In that sense, $P_{tot,part}$ can be represented as a vector,

$$P_{tot,part} = \begin{bmatrix} P_{tot,part_{\gamma=0}} \\ \cdot \\ \cdot \\ \cdot \\ P_{tot,part_{\gamma=1}} \end{bmatrix} \tag{3.36}$$

If both of the loading factors (γ_1 and γ_2) is equal to one then $P_{tot,part_{\gamma}} = P_{tot}$. The value of these factors can be any number between zero and one. However, the summation from the P_{prop} and P_{aux} should not exceed the $P_{tot,part}$.

The part load total demand ($P_{tot,part}$) can be fulfilled by the devoted engines or a combination between primary and auxiliary engine power, depends on the transmission types. In this way, noise can be decreased by finding the best power combination between the main engine and the auxiliary engine. Therefore, $P_{tot,part}$ can be written as,

$$P_{tot,part_\gamma} = P_{main_\gamma} + P_{aux,eng_\gamma} \tag{3.37}$$

Since the possible combination can be more than one, then it is convenient to make equation 3.37 into a vector form,,

$$P_{tot,part_\gamma} = \begin{bmatrix} P_{main_{combination=i}} \\ \cdot \\ \cdot \\ \cdot \\ P_{main_{combination=u}} \end{bmatrix} + \begin{bmatrix} P_{aux_{combination=i}} \\ \cdot \\ \cdot \\ \cdot \\ P_{aux_{combination=u}} \end{bmatrix} \tag{3.38}$$

Where P_{main_γ} and P_{aux,eng_γ} are the power contribution from the main engine and the auxiliary engine respectively. The vector form is required if there will be more than one combination between the main engine and auxiliary engine to achieve the $P_{tot,part_\gamma}$. In a diesel-hybrid plant, the electrical motor power is either being generated by the auxiliary engine or the main engine. Therefore, there is no need to put the power of electrical motor in the equation 3.38.

3.4.5 Number of Compartments

The compartment here is defined as the room where all the equipment is located depending on the arrangement chosen by the designer. As can be seen from Table 3.15 the number of rooms is only limited to two compartments. Theoretically, it is possible to have more than two compartments but the idea behind this parameter is to see if increasing the number of rooms will affect the propulsion system noise level. Therefore, setting the maximum number of rooms into two is reasonable.

In this thesis, all the arrangements always have two types of rooms, i.e., (1) Room for the propeller driver (2) Room for the diesel genset. For a diesel-mechanical plant, the propeller driver consists of the diesel engine and the gearbox. The hybrid system is like the diesel-mechanical plant but with an added equipment, which is an electrical motor. On the other hand, the diesel-electrical plant has an electric motor instead of a diesel engine inside the propeller driver compartment. In the single shaft arrangement, it is not possible to have two propeller driver compartments.

	Equipment		
	Diesel-Mechanical	**Diesel-Electrical**	**Diesel-Hybrid**
Propeller Driver's Room	Diesel Engines Gearboxes	Electric Motors	Diesel Engines Gearboxes Electric Motors
Diesel Genset's Room	Diesel Gensets	Diesel Gensets	Diesel Gensets

Table 3.19 Two types of rooms with the equipment for every transmission types

When there are two compartments for a type of configuration, the number of engines and the room dimensions are identical to each other. For example, if one chooses to have six diesel gensets in a diesel electrical configuration with two compartments, then each compartment will have three gensets inside it. The equipment is always arranged symmetrically with an orientation to port and starboard sides.

In that sense, the number of engines per compartment can be written as follows,

$$N_{equipment@compartment} = \frac{N_{equipment}}{N_{compartment}} \tag{3.39}$$

Where $N_{equipment}$ is the number of a certain engine, $N_{compartment}$ is the number of compartments.

The dimensions of the room are also important to predict because it will affect the airborne noise level in the receiver location. The basis of this prediction is the equipment sizing which will be explained in subchapter 3.5. When calculating the dimensions of the room, one should consider the minimum distance from the equipment to the wall for the maintenance purpose. The International Maritime Organization (IMO) guidelines, MSC/Circ.834, does not describe exactly how much space is needed but rather considers what kind of space should be provided. Nevertheless, the project guide documentations from the engine manufacturers such as MAN and Wartsila usually give a minimum distance between the equipment and the boundary surface of the room. This information is then used as the input to the model.

Considering each transmission type has different equipment, the expressions to calculate the room dimensions are also unique for every arrangement. The expressions below show the general formulas required to predict the room dimensions.

For Diesel-Mechanical Driver's Room,

$$L_{room} = L_{de} + L_{gb} + \Omega_{min} \tag{3.40}$$

$$W_{room} = \Psi_{min} + B_{min} + max\,\{N_{de} * W_{de}\,;\; N_{gb} * W_{gb}\} \tag{3.41}$$

$$H_{room} = \chi_{min} + \;max\,\{H_{de}\,; H_{gb}\} \tag{3.42}$$

For Diesel-Electrical Driver's Room,

$$L_{room} = L_{em} + \;\Omega_{min} \tag{3.43}$$

$$W_{room} = \;\Psi_{min} + B_{min} + W_{em} \tag{3.44}$$

$$H_{room} = \;\chi_{min} + H_{em} \tag{3.45}$$

For Diesel-Hybrid Driver's Room,

$$L_{room} = L_{gb} + max\,\{L_{em}\,; L_{de}\} \tag{3.46}$$

$$W_{room} = \Psi_{min} + B_{min} + max\,\{N_{de} * W_{de} + N_{em} * W_{em}\,;\; N_{gb} * W_{gb}\} \tag{3.47}$$

$$H_{room} = \chi_{min} + max\,\{H_{de}\,; H_{gb}\,; H_{em}\} \tag{3.48}$$

For Diesel-Genset Room,

$$L_{room} = L_{dg} + \Omega_{min} \tag{3.49}$$

$$W_{room} = \Psi_{min} + B_{min} + W_{dg} \tag{3.50}$$

$$H_{room} = \chi_{min} + H_{dg} \tag{3.51}$$

Where B_{min} is the minimum distance between two equipment in parallel. The Ω_{min}, Ψ_{min} and χ_{min} is the minimum distance requirement between the length, width and height of the equipment to the boundary surface respectively. This will make sure the maintenance space in every compartment is fulfilled.

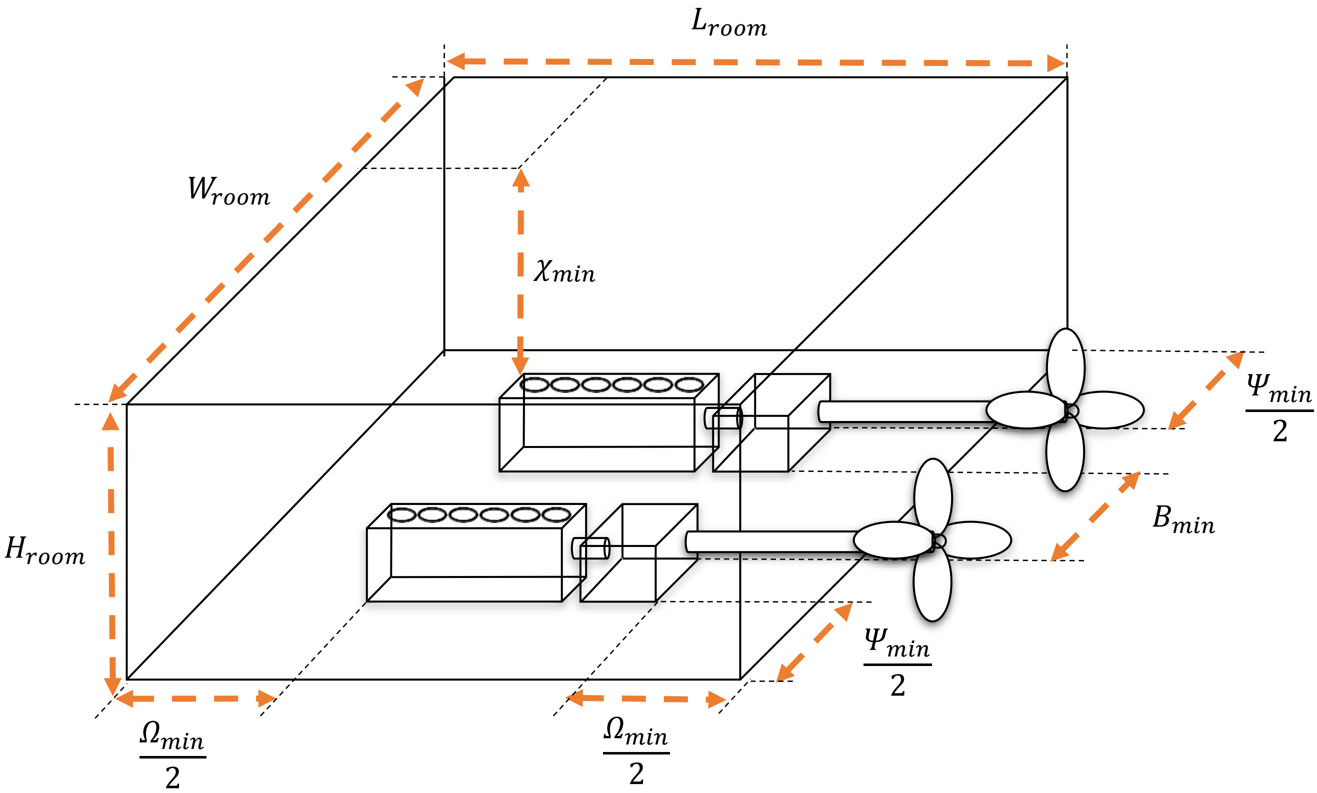


Figure 3.11 Engine Room dimensions details

The airborne noise interaction between two types of rooms is considered negligible. The transmission loss from the bulkhead significantly reduces the sound pressure level from the source room. Consequently, it becomes really small compared to the noise in the receiver room. However, the structure-borne noise interaction is not negligible. The contribution from the other compartment needs to be considered to get the final acceleration level at the location of interest.

3.5 Engine Sizing Model

The dimensions of the engines need to be modelled because it will affect the dimensions of the compartment. This sub-chapter discusses the theories behind the sizing model. The sizing model uses the regression analysis or a combination of the first principle and regression analysis to predict the equipment dimensions. The first principle is used to size the core of the machine. It consists of primary and secondary elements. Together, they are used to determine the dimensions of the main parts of an engine, which can be related to the actual dimensions by regression analysis [32]. There are four sizing models to be created, i.e., the diesel engine sizing model, the diesel generator set model, the electric motor model and the reduction gear model.

A study by *De Vos et al.* [32] is used to determine the diesel engine secondary elements and the overall engine dimensions. The primary element model of the diesel engine is developed in this thesis instead of using the formula from *De Vos et al.* [32]. The modification is made because the equation to determine the primary element model from *De Vos et al* [32] is sensitive to the mean piston speed. The diesel generator set also has similar modifications to the diesel engine. The electric motor sizing model uses the method by *De Vos et al* [32] for sizing the core of the machine and the overall dimensions. The gearbox sizing model is developed in this project by implementing only the regression analysis. It can be done by finding a relation between important features of the gearbox.

An extensive database for each equipment is also created to do the regression analysis. Firstly, the diesel engine database contains the L and V four strokes engines along with the two strokes engines. Secondly, the diesel generator database is also similar to the diesel engine, but it does not have the two strokes engine variety. Thirdly, the electric motor database consists only of induction motor type. Lastly, the reduction gear database contains the SISO and the DISO gearbox but only single-stage reduction gear is considered. In chapter 5, the verification analysis will be done by comparing the predicted dimensions against the real dimensions of the equipment.

3.5.1 Diesel Engine Sizing Model

The primary element of the diesel engine is the cylinder. Based on this, one should find the Bore Diameter (D_b) and the Stroke Length (L_s) first [32]. In this model, the regression analysis is implemented to calculate the D_b. A good relation is found between the power per cylinder with the diameter of the engine from the database. This is not surprising because the expression to calculate power per cylinder is also a function of the D_b [38],

$$\frac{P_b}{z} = P_i = \frac{\pi}{8} * \frac{(p_{me} * c_m * D_b^2)}{k} \tag{3.52}$$

It is evident that with larger D_b one can get more power from a cylinder, assuming the L_s is constant and the shape ratio (λ_s) is greater than equal to one. It is important to note that it is not possible to have λ_s below one otherwise the combustion will not occur [39]. The relation between them for the medium-speed diesel engine four strokes can be seen from the graph below. For other diesel engine types see Appendix B.

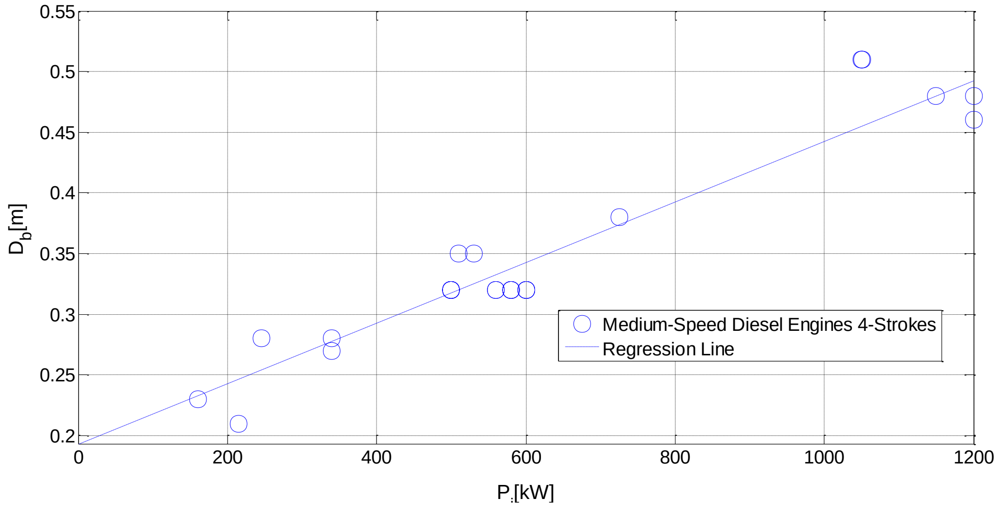


Figure 3.12 Linear regression analysis of power per cylinder and bore diameter

The R squared from the regression line is equal to 0.92 which means that the relationship is valid and shows a very good fit between each other. The simple linear regression equation then can be written as follows,

$$D_b = \varepsilon_{D_b,1} + \varepsilon_{D_b,2} * P_i \tag{3.53}$$

Where $\varepsilon_{D_b,1}$ and $\varepsilon_{D_b,2}$ are the regression coefficients which equal to 0.19 and 2.49 * 10^-4 respectively. For the prediction of $L_{s,}$ one should find the engine nominal speed and the mean piston speed first. The formula is as follows [39],

$$L_s = \frac{c_m}{2 * \frac{N_{nom}}{60}} \tag{3.54}$$

Where N_{nom} is the nominal speed in rpm and c_m is the mean piston speed in m/s. The c_m taken from the typical value of a certain diesel engine type. In this way, the predicted mean effective pressure engine will always be in the technology parameter range. Furthermore, the N_{nom} of the engine will be determined by regression analysis too. From the database, it can be seen that the D_b has a good linear relation to the N_{nom} of the engine. The graph below shows the fit between D_b and N_{nom} from 64 four strokes medium-speed diesel engines data.

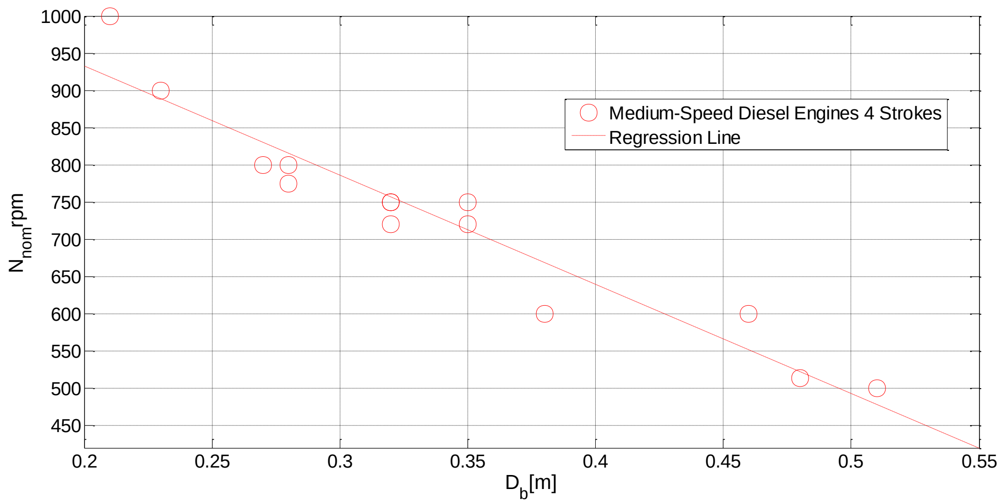


Figure 3.13 Linear regression analysis between bore diameter and engine nominal speed

The R squared value is 0.93, and the regression equation is a function of D_b.

$$N_{nom} = \varepsilon_{rpm,1} + \varepsilon_{rpm,2} * D_b \tag{3.55}$$

The $\varepsilon_{rpm,1}$ and $\varepsilon_{rpm,2}$ are the regression coefficients which equal to 1226,1 and -1.47 * 10^-3.

Another alternative to calculating the primary elements is to use the first principle equation. This method can be done if the database is unavailable. The nominal engine speed needs to be calculated first using the power per cylinder and the typical value of the "three main players", i.e., mean effective pressure, mean piston speed and shape ratio [39].

$$N_{nom}^2 = \frac{\pi * p_{me} * c_m^3 * z}{32 * k * \lambda_s^2 * P_b} \tag{3.56}$$

The "three players" typical values for the four strokes medium speed diesel engine are equal to 24 bar, 10 m/s and 1.3 respectively. For different types of diesel engines, one can find the values in [39] . However, in this equation, the great sensitivity of the mean piston speed followed by the shape ratio may lead to misleading engine speed prediction.

After the sizing of the primary elements is completed, the second element can be determined. The secondary element is the crankshaft. The length of it is rather simple to calculate because it can be determined by using the number of cylinders that are connected to the crankshaft and their diameters [32]. For an L-engine, it is estimated as $i * D_b$ and if it is a V-engine the length becomes $\frac{i}{2} * D_b$. This will directly become the core length of the diesel engine.

Determining the core dimensions of the engine depends on two factors, i.e., the cylinder arrangements and the construction types. The first one can be L, V, or a boxer engine while the construction types may differ between the trunk piston type or the crosshead type construction. For the types of cylinder arrangements, if it has the same power output, the V engine obviously will be shorter than the L engine, but the V engine will lead to a wider dimension. These two typical difference will reflect in the calculation of the length and the width of core dimensions. In a crosshead engine, the height of the engine will be most likely higher than the trunk piston type because the crosshead construction is mounted separately from the piston to counteract the side forces, which leads to the extra stroke length between the crankshaft outer diameter and the bottom of the cylinder. On the other hand, the trunk piston type used the piston itself to counteract the side forces. Therefore, no extra length required. However, this means that the trunk piston construction has a larger piston. The following expressions show how to predict the core dimensions of the diesel engines [32],

$$L_{core,de} = i * D_b \qquad for\ L\ engines \tag{3.57}$$

$$L_{core,de} = \frac{i * D_b}{2} \qquad for\ V\ engines \tag{3.58}$$

$$W_{core,de} = 2 * \max\left\{\left(\frac{L_s}{2} + (1 + c_t) * L_s\right) * sin\left(\frac{\alpha}{2}\right) + \frac{D_b}{2} * cos\left(\frac{\alpha}{2}\right); \frac{L_s}{2}\right\} \tag{3.59}$$

$$H_{core,de} = \frac{L_s}{2} + \max\left\{\left(\frac{L_s}{2} + (1 + c_t) * L_s\right) * cos\left(\frac{\alpha}{2}\right) + \frac{D_b}{2} * sin\left(\frac{\alpha}{2}\right); \frac{L_s}{2}\right\} \tag{3.60}$$

Where c_t is the construction type, if it is a cross head type then $c_t = 1$ while for the trunk piston type $c_t = 0$. The α is the angle between cylinders, for L-engine $\alpha = 0$ while in the V-engine of course it is always $\alpha > 0$.

Another regression analysis has to be done to determine the overall length of the engine from the core size of the engine. The regression method can be done by plotting the core dimensions to the actual dimensions of the engine. There are 64 data used to derive the constant from simple linear regression. Again, these graphs below only show the four strokes medium speed diesel engine analysis.

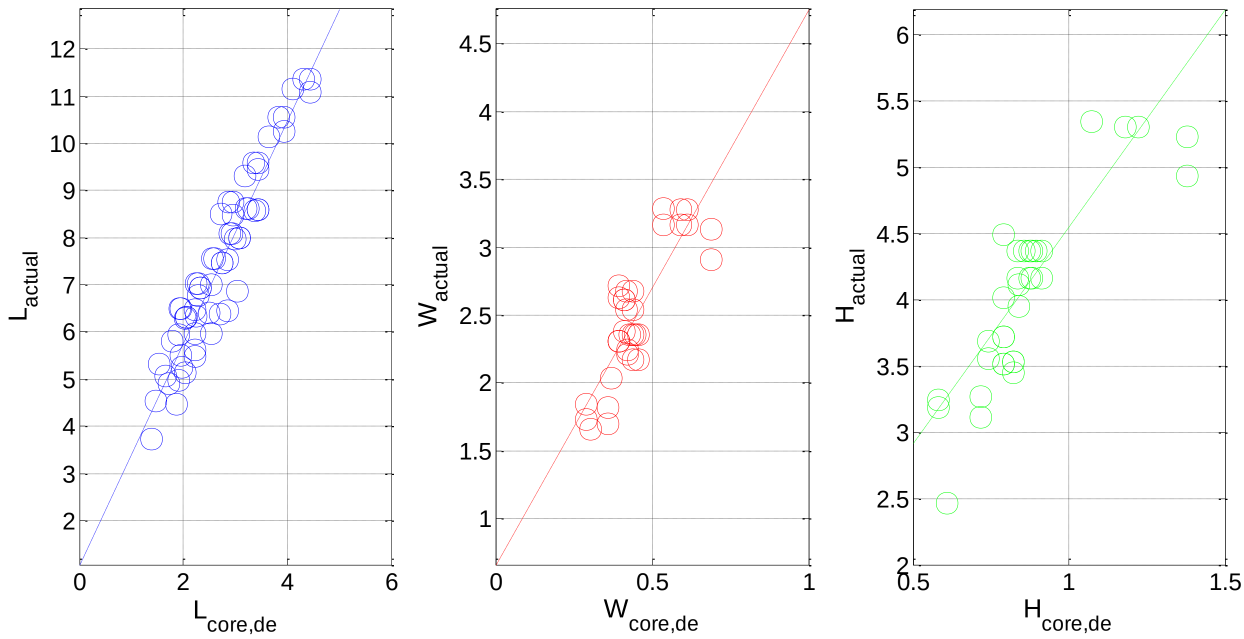


Figure 3.14 Linear regression analysis between diesel engines actual and core dimensions

Therefore, the predicted overall dimensions can be written as follows,

$$L_{overall,de} = \beta_{de,1} * L_{core,de} \tag{3.61}$$

$$W_{overall,de} = \beta_{de,2} * W_{core,de} \tag{3.62}$$

$$H_{overall,de} = \beta_{de,3} * H_{core,de} \tag{3.63}$$

Where $\beta_{de,1}$, $\beta_{de,2}$ and $\beta_{de,3}$ are the regression coefficients for length, width, and height respectively. It is important to know that the difference between the actual dimension and the predicted dimension is due to the “disturbances” in the actual engine. The disturbances here are the turbocharger, cooling water, lubrication oil supply and other equipment attached to the engine which varying a lot between engines.

3.5.2 Gearbox Sizing Model

There are two types of gearboxes considered in this thesis, the SISO, and DISO Gearbox. For both types, the regression analysis is used to predict the size of the equipment. Nevertheless, the relations implemented in the analysis are different between the SISO and DISO Gearbox. This distinction is because they have different characteristics even though they look similar. It needs to be mentioned that the gearbox data are limited due to the scarcity of it, especially the DISO gearbox. Therefore, the database is not as big and detailed as diesel engines.

For the SISO gearbox, the dimensions are dictated by the ratio and moment output of the gearbox. Furthermore, the pinion dimension and the wheel dimension are directly proportional to the moment and the ratio of it. The pinion and wheel are the primary element and secondary element for sizing the gearbox respectively [32]. Also, it is assumed that the gearbox has 100% efficiency. Therefore the output power from the gearbox is equal to the brake power of the engine. In that sense, it can be expressed by,

$$M_{gb,output} = \frac{P_b}{2 * \pi * N_{propeller}} \tag{3.64}$$

Where $N_{propeller}$ is the nominal speed of the propeller. The regression analysis between the moment output and the dimensions of the SISO gearbox separated into two different types. The first type is a gearbox with the ratio below five and the other types is a gearbox with the ratio above five. This needs to be done because the fit between dimensions and moment output shows a lack of correlation when everything puts together into the same line. The graphs below show the relations between the moment output and dimensions of the gearboxes.

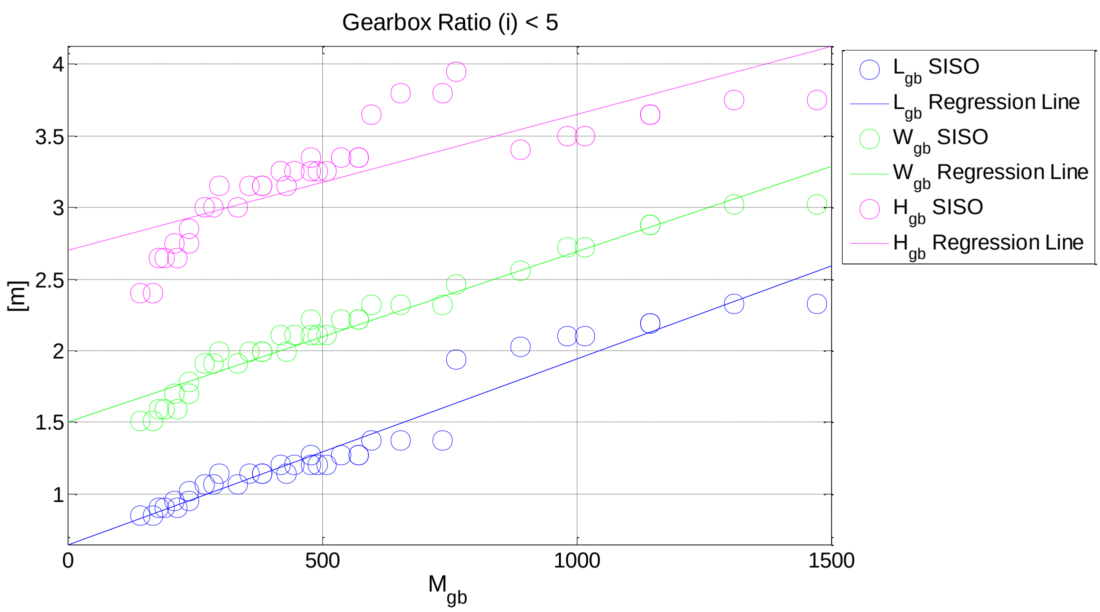


Figure 3.15 Relation between moment output and dimensions of the gearboxes for i < 5

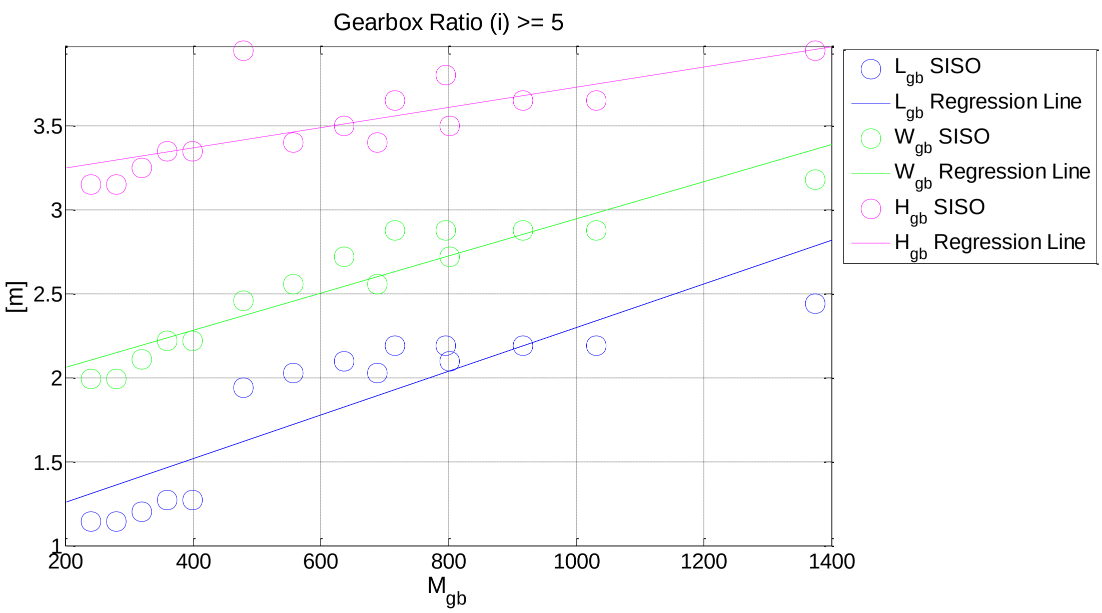


Figure 3.16 Relation between moment output and dimensions of the gearboxes for i ≥ 5

For the length and the width of the equipment, the relations seem acceptable with the R squared value is 0.94 and 0.95 for the ratio below five, and 0.76 and 0.88 for the ratio greater than equal to five. Nevertheless, the R squared value for the height is quite low, 0.67 for the ratio below five ,and 0.55 for the ratio greater than equal to five. These values are acceptable because the height of the gearbox will not affect the room height calculation significantly. It rather depends on the diesel engine height. The equations from the analyses are as follows,

For $i < 5$,

$$L_{overall,SISO} = \beta_{gb,1} + \beta_{gb,2} * M_{gb,output} \tag{3.65}$$

$$W_{overall,SISO} = \beta_{gb,3} + \beta_{gb,4} * M_{gb,output} \quad (3.66)$$

$$H_{overall,SISO} = \beta_{gb,5} + \beta_{gb,6} * M_{gb,output} \quad (3.67)$$

For $i \geq 5$,

$$L_{overall,SISO} = \beta_{gb,7} + \beta_{gb,8} * M_{gb,output} \quad (3.68)$$

$$W_{overall,SISO} = \beta_{gb,9} + \beta_{gb,10} * M_{gb,output} \quad (3.69)$$

$$H_{overall,SISO} = \beta_{gb,11} + \beta_{gb,12} * M_{gb,output} \quad (3.70)$$

The DISO gearbox model uses the two relations to define its dimensions. The first one is the relation between the moment input and the length of the gearbox. This relation is true because the length of the gearbox is determined by the thickness of the wheel diameter which dictated by the moment input to the gearbox. The second relation is the width of the diesel engine with the gearbox length and the gearbox height. It is expected that the relation will show a good fit because with double input feature the gearbox manufacturers should adapt to the dimensions of the engine, especially the width of it. Moreover, they also have to take into account the minimum distance between engines when designing the DISO gearbox.

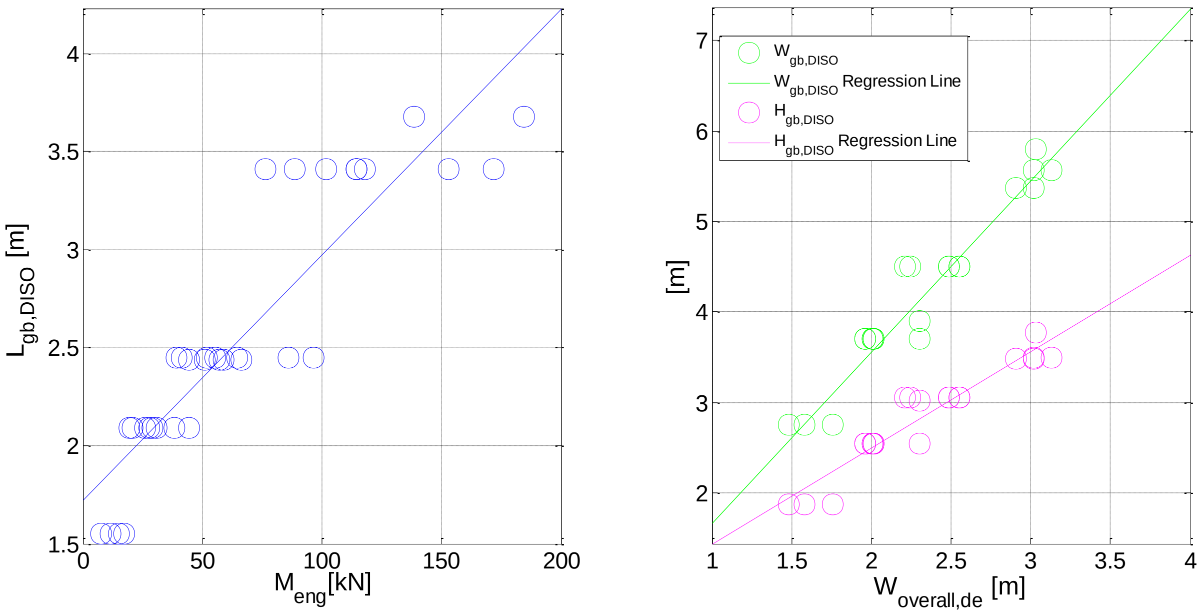


Figure 3.17 Relation between the moment input and the gearboxes length (left); Relation between the diesel engines width and the gearboxes dimensions (right)

The equations to calculate the DISO Gearbox dimensions are as follows,

$$L_{overall,DISO} = \beta_{gb,13} + \beta_{gb,14} * M_{gb,input} \quad (3.71)$$

$$W_{overall,DISO} = \beta_{gb,15} + \beta_{gb,16} * W_{overall,de} \quad (3.72)$$

$$H_{overall,DISO} = \beta_{gb,17} + \beta_{gb,18} * H_{overall,de} \tag{3.73}$$

The relation in the left figure shows a decent fit with R squared value of 0.81, the green line and the magenta line in the right figure show a better fit with a similar R squared value of 0.93. Therefore, it is possible to use these relations to predict the size of the equipment although there is no first principle involved here.

3.5.3 Electric Motor Sizing Model

There are many different motor types have been used or at least proposed for ship propulsion, i.e., direct current (DC) motors, induction motors, ordinary wound field synchronous motors, synchronous motors with a permanent magnet, synchronous motors with superconducting field windings, acyclic motor and doubly fed induction motors [40]. The induction motor is one of the most matured technology for use in large ship and widely used in heavy industry because of the low capital cost and low maintenance. Another advantage is that with an induction motor, the Integrated Electric Propulsion (IEP) technology might be accelerated [37]. Therefore, in this thesis, only the induction motor is being considered as the propulsion motor.

Based on [32], the primary element of the electrical motor is the rotor part. The dimensions of it can be computed by the first principle relationships using the circumferential speed, the rotor shape factor, the brake power, and the mean shear stress of the motor. It is possible to have a short rotor with a large diameter or vice versa to deliver the same power output. This flexibility means that the electric motor has less limitation in its shape factor compared to the diesel engine. The formulas used to calculate the primary elements are as follows [32],

$$D_R = \sqrt{\frac{P_{b,em}}{\pi * \tau_{em} * v_t * \lambda_R}} \tag{3.74}$$

$$L_R = \sqrt{\frac{P_{em} * \lambda_R}{\pi * \tau_{em} * v_t}} \tag{3.75}$$

Where τ_{em} is the mean shear stress in kN/m2, v_t is the circumferential speed in m/s, P_{em} is the electrical motor power output and λ_R is the shape factor that can be written as,

$$\lambda_R = \frac{L_R}{D_R} \tag{3.76}$$

The secondary element of the electric motor is the stator. The parameters used to size the stator is the ratio(s) between the rotor diameter and the stator diameter. Unfortunately, it is not a functional parameter that can be determined by the designer of ship systems. A typical value for s used in this analysis is in the range between 0.45-0.55. This range is taken from the literature [32]. Since the stator dimensions cover all the primary element, therefore the dimensions of it are equal to the core dimensions of an electric motor which can be written as follows,

$$L_{core,em} = L_{stator} = L_R \quad (3.77)$$

$$W_{core,em} = D_{stator} = \frac{D_R}{s} \quad (3.78)$$

$$H_{core,em} = D_{stator} = \frac{D_R}{s} \quad (3.79)$$

Afterwards, the core dimensions of the equipment need to be compared to the overall dimensions. In order to that, one should use the regression analysis to find the regression coefficients between those. As stated earlier, only the induction motor will be considered in this thesis. Therefore, the electric motor database only consists of induction motor but with various types, such as high voltage and low voltage motor. The database built by using the catalogue from ABB induction motor [41]. The graphs below show the analysis of 46 different ABB-HXR low voltage induction motors,

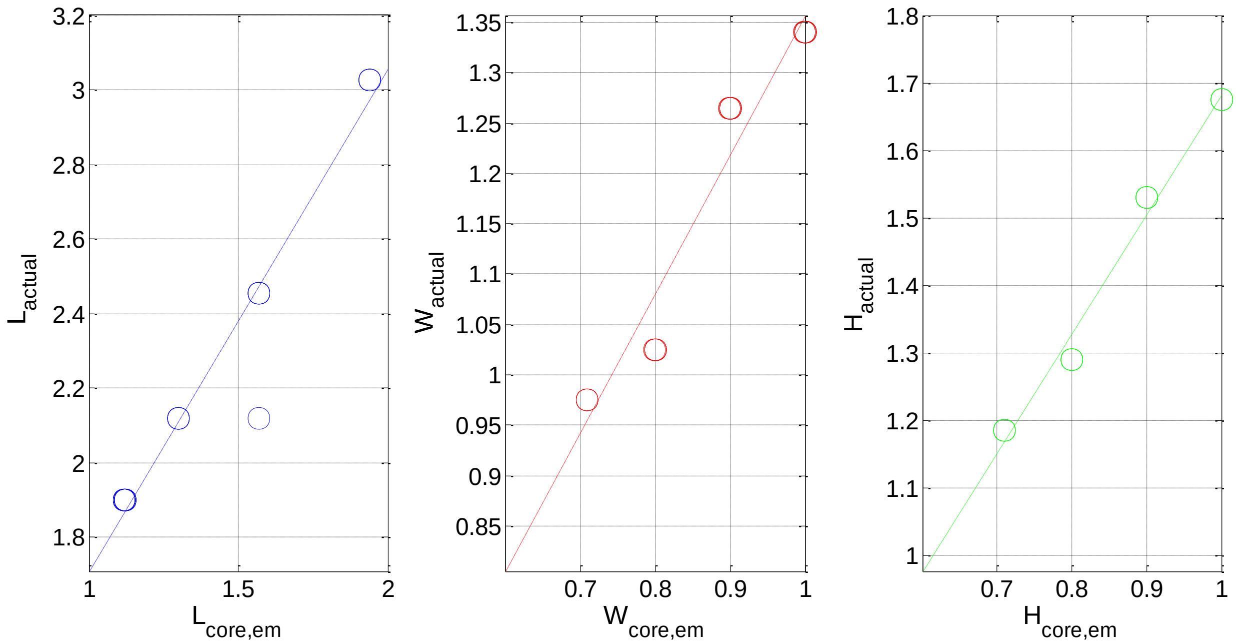


Figure 3.18 Linear regression analysis of electric motors actual and core dimensions

Therefore, the predicted overall dimensions can be written as follows,

$$L_{overall,em} = \beta_{1,em} + \beta_{2,em} * L_{core,em} \quad (3.80)$$

$$W_{overall,em} = \beta_{3,em} + \beta_{4,em} * W_{core,em} \quad (3.81)$$

$$H_{overall,em} = \beta_{5,em} + \beta_{6,em} * H_{core,em} \quad (3.82)$$

Based on the graph it is evident that the core dimensions correlate well with the overall dimensions. The R squared value for every line is above 0.9, which justifies that the relation is valid to use. It is important to note that the regression coefficients from the graph above only valid for the ABB-HXR low-voltage induction motor.

3.5.4 Diesel-Generator Set Sizing Model

In principle, the diesel generator set is a diesel engine that drives a generator to produce the electricity. Normally, the diesel engine is larger than the size of the generator. Therefore, the core dimensions of the diesel engine are also the core dimensions of the diesel generator set. It is assumed that the diesel engine used as the prime mover for the generator has the same characteristic as the diesel engine used as the propulsion driver. In this way, there is no need to create a new model for the diesel generator set sizing; one can use the diesel engine sizing model from subchapter 3.5.1 and defines a new relation between the diesel engine core dimensions to the overall dimensions of the generator set. The figures below show the relation between the core and the actual dimensions of the four strokes medium speed in-line diesel generator sets.

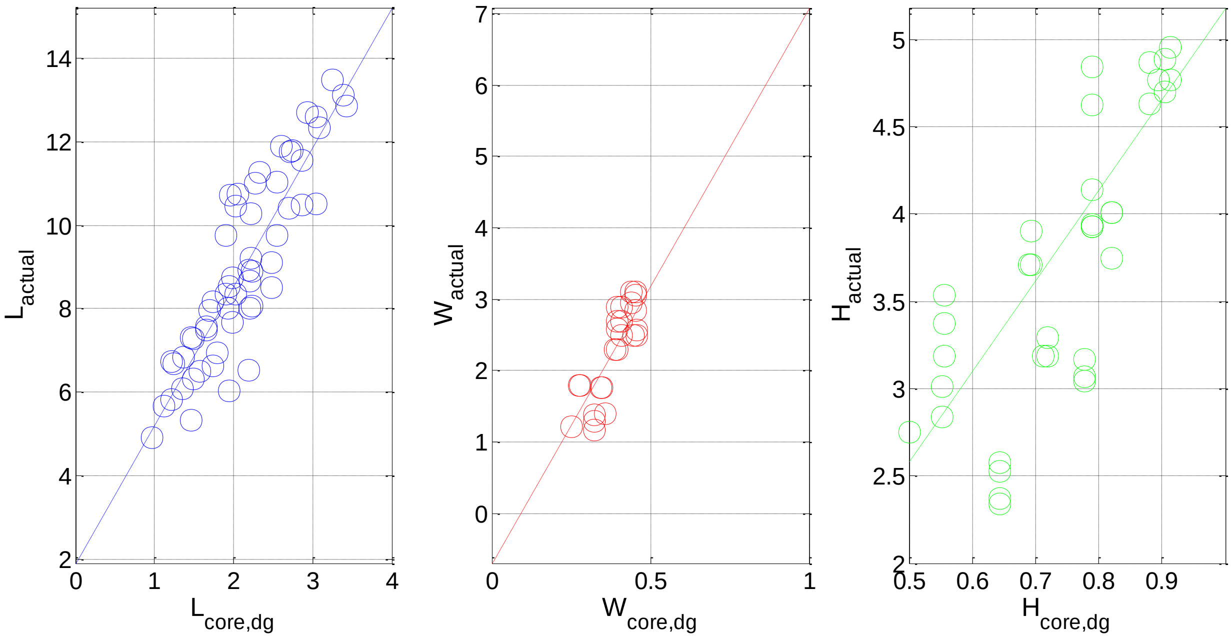


Figure 3.19 Linear regression analysis between diesel generator sets actual and core dimensions

Therefore, the predicted dimensions can be written as follows,

$$L_{overall,dg} = \beta_{dg,1} + \beta_{dg,2} * L_{core,de} \tag{3.83}$$

$$W_{overall,dg} = \beta_{dg,1} + \beta_{dg,2} * W_{core,de} \tag{3.84}$$

$$H_{overall,dg} = \beta_{dg,3} + \beta_{dg,4} * H_{core,de} \tag{3.85}$$

In the height relation, the scatters show less fit compared to the length and width of the diesel generator set. This deviation is because there are many disturbances factor in the genset height. For instance, the actual height between MAN 8L28/32H and MAN 9L28/32H is different although they have similar primary element dimensions, cylinder arrangement, and construction type.

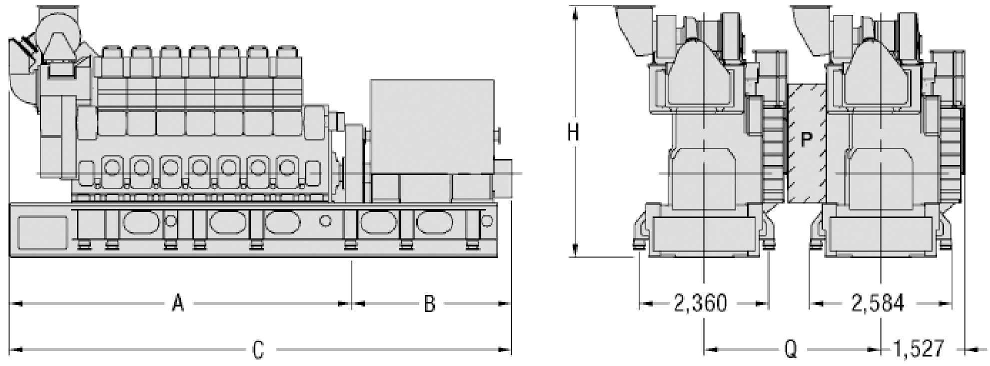


Figure 3.20 Typical MAN Diesel generator technical drawing [42]

From the technical drawing, it can be seen that the base frame of the diesel genset is being taken into account for the overall height. That can be the main reason why they have different height. However, there is no detail about the base frame dimensions in the project guide documentation.

3.6 Evaluation Methodology

This subchapter will describe in detail how the evaluation method works to assess the effect of every parameter on the noise level. The overview of the assessment methodology is shown in the figure below. It can evaluate the effect of each design choice to the noise level of the analysed power configuration. In this way, the consequences of choosing a particular design option towards the generated airborne and structure-borne noise can be identified. The importance of doing this in the early design phase is quite high since it can be a preventive action rather than corrective measures against the noise problem. The designer can also use the results of this evaluation as a consideration when choosing the propulsion system of a ship. In this way, the designer can have less uncertainty in the early design phase. The final output of the evaluation methodology is the value of noise levels of a propulsion system as a function of a certain parameter. As stated in the previous chapter, there are only four types of transmissions that are being investigated in this thesis, the Diesel Mechanical, the Diesel Electrical, the Diesel Hybrid and the Battery. However, the methodology can be used for another type of transmission with some adjustments.

A normal procedure to evaluate the propulsion system noise level is to vary one aspect and keep the other constant. However, it is possible that if one makes a change in one parameter, that change will affect the other parameters. These relations will give various effects on the acoustic performance of a propulsion configuration.

The transmission type is the core of the design choices. It has a direct relation to the sound level and indirect relations to the other parameters. Once it changes, almost every parameter needs to be modified. The number of engines has an indirect relation to other parameters as well when it is modified. It can affect the number of the shafts (except the number of diesel generator sets). It also works the other way around, because when the number of shafts is changed, then the number of engines also needs to be changed. When varying the number of rooms, it is also related to the number of engines because it is not possible to change the number of rooms if an arrangement only has a single engine. The operation point is also related to the number of engines because more equipment means a larger set of possible load combinations to reach a total part load demand. The operation point is varying over the loading factors, the number of engines and the transmission types. Sub-chapter 3.8 will explain the characteristics of the loading combination per transmission type in more detail.

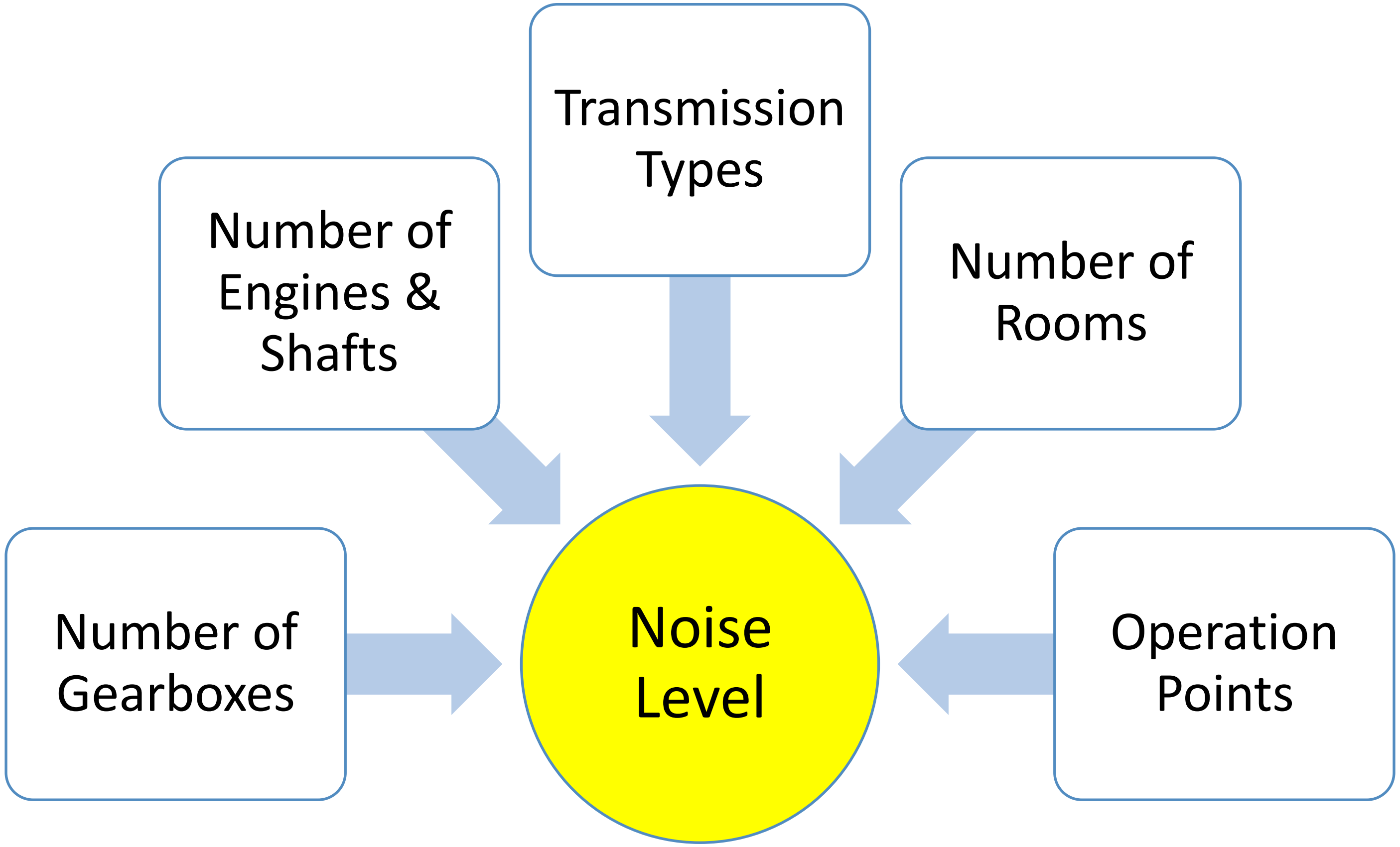


Figure 3.21 The evaluation methodology

3.7 Engine Envelope and Load Line

The engine envelope and the load line needs to be determined first before defining the load combinations. Engine envelope is a graph representing the possible operating range of an engine, indicated by engine power vs engine speed. The engine manufacturers usually provide this information in the engine catalogue, which will be used in this thesis. Predicting the envelope is not within the scope of this project.

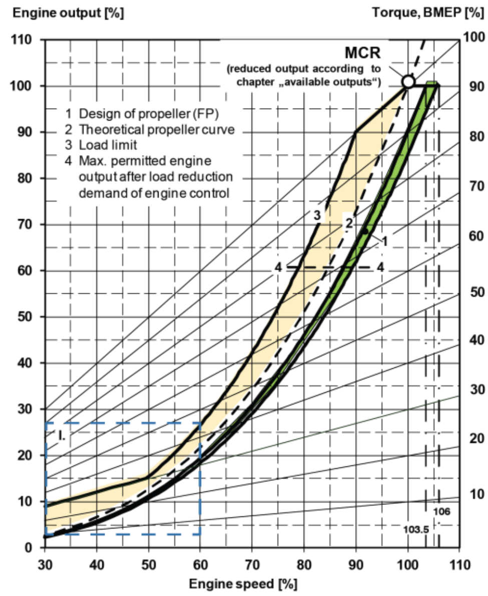


Figure 3.22 Example of Engine Envelope from the Engine Manufacturer Documentation [42]

The load line of the engine is taken from the optimal operation points on a fixed pitch propeller (FPP). This data also provided by the manufacturers [42]. If the loading point is not defined in the documentation, one can use the extrapolation from the trend line. It is important to define the load line because it is used to determine the engine power and the speed in part load conditions. In the load line calculation, for every point below 25% engine output, the values are extrapolated.

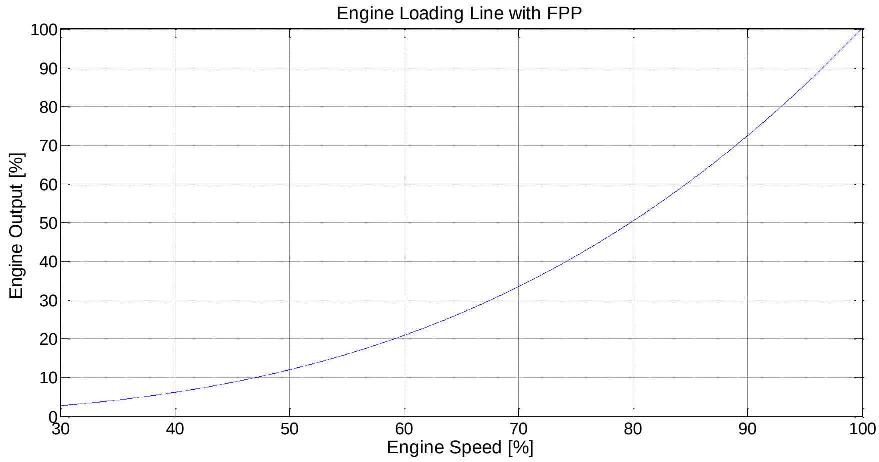


Figure 3.23 Typical MAN Engine Load Line with FPP

3.8 Operation Conditions

The analysis of the loading point changing effect on the propulsion system noise level will be done in this thesis. When the loading point is lower than 100% it means that the engine works at part load condition. If the propulsion driver configuration only has one engine, the possible operation condition is only one combination (except for diesel-hybrid). However, this project also investigates the propulsion system with multiple engines, thus will create more than one possible operation conditions in part load condition. Therefore, it is required to create a method to determine the possible load combinations to fulfil the part load demand. The method will be different for every transmission type due to unique operation modes per transmission type.

The inputs to this method are the propulsion part load demand and the auxiliary part load demand. Both of these parameters are affected by the gamma. The model will determine all the possible load combinations, which can be used to provide the power for the part load conditions. These combinations then will be used as the input to the load line to determine the actual speed of the engine. Afterwards, the noise model calculates the noise level of these load combinations. The load combination with the lowest noise level is chosen as the operation point in the part load condition. When the part load total demand is higher than the brake power of one engine, usually the lowest noise level is produced by operating one engine in a high loading point while the other engine is in a very low loading point. This operation is not convenient for the overall efficiency of the propulsion plant. Therefore, in order to maintain the optimal value of fuel consumption, the method will choose the equal loading point as the best operating point. The equal loading point means that the load will be shared equally to some or all of the available engines. Although it does not give the lowest noise level, the difference between the noise level in equal split condition and the lowest noise level is small.

3.8.1 Diesel-Mechanical Loading Combinations

Firstly, the diesel mechanical is the most rigid one regarding operation modes. There is no connection between the $P_{b,de}$ and $P_{b,dg}$ meaning that each of these values will be devoted only to its own system. For instance, the $P_{b,de}$ can only generate power for the P_{prop} and $P_{b,dg}$ provides only for the P_{aux}.

Defining the possible load combinations for a certain part load total demand are relatively simple. First, a matrix consisting of the engines power step needs to be created. The number of rows is equal to the amount of series of number starting from zero to the brake power of the engine while the column is equal to the number of engines. It is better to set the interval not too low to avoid creating too many combinations. The following matrix show possible combinations of n number of diesel engines with 50 as the chosen interval,

$$P_{sets} = \begin{bmatrix} 0_1 & 0_2 & \cdots & 0_n \\ 50_1 & 0_2 & \dots & 0_n \\ \vdots & \vdots & \ddots & \vdots \\ P_{b,1} & P_{b,2} & \cdots & P_{b,n} \end{bmatrix} \tag{3.86}$$

Secondly, choose every row that has the summation equal to the part load total demand since the matrix in equation 3.86 consists all possible loading point. This step needs to be done to create the matrix that consists the combination of a certain demand. The evaluation function is as follows,

$$P_{comb} = \begin{cases} P_{b,1}, P_{b,2}, \dots, P_{b,n} \quad if\ (\sum P_{sets}(n_{row}) = P_{prop,part}) \\ 0 \quad if\ (\sum P_{sets}(n_{row}) \neq P_{prop,part}) \end{cases} \tag{3.87}$$

Where n_{row} is the number of rows, $P_{b,n}$ is the actual power of the n engine and $P_{prop,part}$ is the part load propulsion demand This function should be implemented in every rows inside the P_{sets} matrix. However, not all of these combinations are possible to operate with the diesel-mechanical plant. Therefore, one needs to set a constraint which depends on the arrangement of this plant. For multiple engines with a single shaft the constraints are,

1. ***If the part load total demand is greater than the nominal power of one engine, then the only possible combination is when all the engines in one shaft operate in symmetrical load.***
2. ***If the part load total demand is lower equal than nominal power of one engine, then it is possible to use only that particular engine to provide the power while the other engine is in standby mode.***

If the number of diesel engines is equal to the number of shafts, then there is no constraint for the P_{comb}. One can directly use it as an input to the sound calculation. The last step is to define the gearbox input power in part load condition. It is quite straightforward because the power input to the gearbox is always equal to the total engine power output of a certain gearbox.

$$P_{b,gb} = \sum P_{b,de} \quad @gearbox \tag{3.88}$$

3.8.2 Diesel-Electrical Plant Loading Combinations

Unlike the diesel-mechanical, the diesel-electrical plant is not a devoted system thanks to the Integrated Electric Propulsion (IEP) configuration. Therefore, the power generated from $P_{b,dg}$ can be used for both P_{prop} and P_{aux}. The possible combinations of a part load condition become more complex. The first step is always the same, one should define the P_{sets} and P_{comb} matrix. The method is similar to diesel-mechanical plant (see equation 3.87) but with little adjustments in P_{comb}. This should be done for the diesel generator set and the electric motor power.

$$P_{comb,dg} = \begin{cases} P_{b,1}, P_{b,2}, \dots, P_{b,n} \quad if\ (\sum P_{sets}(n_{row}) = P_{tot,part}) \\ 0 \quad if\ (\sum P_{sets}(n_{row}) \neq P_{tot,part}) \end{cases} \tag{3.89}$$

$$P_{comb,em} = \begin{cases} P_{b,1}, P_{b,2}, \dots, P_{b,n} \quad if\ (\sum P_{sets}(n_{row}) = P_{prop,part}) \\ 0 \quad if\ (\sum P_{sets}(n_{row}) \neq P_{prop,part}) \end{cases} \tag{3.90}$$

Where n_{row} is the number of rows, $P_{b,n}$ is the actual power of the n engine, $P_{prop,part}$ is the part load propulsion demand and $P_{tot,part}$ is the part load total demand.

All these combinations are always possible to implement for a certain part load condition due to the flexibility of diesel-electrical plant. There is no constraint requires to be defined.

The $P_{comb,dg}$ and $P_{comb,em}$ can be used directly as an input to the noise analysis in part load condition. However, the calculation time is significantly increasing due to the number of possible combinations.

3.8.3 Diesel-Hybrid Plant Loading Combinations

The last one is the diesel hybrid plant, which is the most complicated one. According to subchapter 3.4.2, the diesel hybrid can do all the possible types of operations, i.e., PTO, PTI, and PTH. This versatility will lead to a small change in how to create the P_{sets} matrix. In a hybrid system, the diesel generator set, and the diesel engine can work together or separately. Therefore the P_{sets} is equal to,

$$P_{sets} = \begin{bmatrix} 0_{de1} & \cdots & 0_n & 0_{dg1} & \cdots & 0_n \\ 50_{de1} & \vdots & 50_{den} & 50_{dg1} & \vdots & 50_n \\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\ P_{b,de1} & \cdots & P_{b,den} & P_{b,dg1} & \cdots & P_{b,dgn} \end{bmatrix} \tag{3.91}$$

Evaluating only possible combinations for a certain part load using the function as follows,

$$P_{comb} = \begin{Bmatrix} P_{b,de1}, P_{b,dg1}, \dots, P_{b,dgn}, P_{b,den}\ if\ \left(\sum P_{sets}(n_{row}) = P_{tot,part}\right) \\ 0\ \ if\ \left(\sum P_{sets}(n_{row}) \neq P_{tot,part}\right) \end{Bmatrix} \tag{3.92}$$

Where n_{row} is the number of rows, $P_{b,den}$ is the actual power of the n diesel engine, $P_{b,dgn}$ is the actual power of the n diesel genset and $P_{tot,part}$ is the part load total demand.

The electric motor actual power in a part load condition depends on the hybrid operation types. In that sense, there are three equations to calculate the output from the electric machine.

$$P_{b,em,PTO} = P_{b,de,part} - P_{prop,part} \tag{3.93}$$

$$P_{b,em,PTI} = P_{b,dg,part} - P_{aux,part} \tag{3.94}$$

$$P_{b,em,PTH} = P_{prop,part} \tag{3.95}$$

There is no constraint regarding the electric motor because it can always adapt to the engine speed by installing a frequency converter. The gearbox power input is also varying over the operation condition. For instance, in PTO mode the input to the gearbox will be greater than the $P_{prop,part}$ while in PTI mode its equal to the $P_{prop,part}$.

$$P_{b,gb,PTO} = P_{b,de,part} \tag{3.96}$$

$$P_{b,gb,PTI} = P_{prop,part} \tag{3.97}$$

$$P_{b,gb,PTH} = P_{b,dg,part} \tag{3.98}$$

In order to know which operation is used in a load combination, one should check these three conditions,

$$Operation\ Condition = \begin{Bmatrix} PTO \quad if \quad P_{prop,part} \leq P_{b,de,part} \\ PTI \quad if \quad P_{aux,part} \leq P_{b,dg,part} \\ PTH \quad if \quad P_{b,de,part} = 0 \end{Bmatrix} \tag{3.99}$$

The calculation time will take longer time compared to other transmission types. This is due to more possible combinations in the diesel-hybrid plant.

3.8.4 Minimum Noise Source Levels in Part Load Condition

The algorithm will calculate the compartment noise levels with all the possible load combination for every load factor. The number of computations will be limited to the sets of possible load combination. The load vector dimensions change for every load factor due to different part load total demand. The variation in load vector also results in a change of row dimension for noise levels matrix. Afterwards, the $L_{A,eq}$ can be calculated, creating a vector consists of $L_{A,eq}$ for every loading combination. Finally, the lowest $L_{A,eq}$ value is taken along with the loading combination of each equipment. However, when the part load total demand is higher than the brake power of the engine, the equal loading point will be used. This is because one needs to consider the diesel engine efficiency. Although the equal loading point operation does not give the lowest $L_{A,eq}$ but the difference between the $L_{A,eq}$ at equal split operation and the minimum $L_{A,eq}$ is really small.

4

Case Study

4.1 Introduction

The results of the case study are presented here. The effects of each parameter on the noise level generated from the propulsion system are analysed thoroughly to understand better the consequence of early design choices. The diesel-mechanical plant results will be presented first, followed by the diesel-electrical plant and the diesel-hybrid plant. A new parameter will be introduced only for the diesel-hybrid due to the flexibility of loading split between its equipment.

A case study must be done to investigate the effect of each parameter on the sound level produced by the propulsion systems. It is not feasible to analyse the influence of both the ship requirements and the design choices at the same time. Therefore, the effect of the ship requirements will be investigated in the next chapter to see if the trends found in this chapter applies to various ship specifications.

The ship requirements are not necessarily from an actual ship. The aim of this case stud is analysing the behaviour of each parameter for given power demand. The focus is not on how accurate these assumptions are, but on the plausibility of the methodology. Therefore, an imaginary vessel is used with the name of "Vessel A." It is a medium size vessel with the medium-speed engine. The propeller speed of the ship is 200 rpm. The function of the ship is not defined since it is not related to the analysis. The details of the ship data are as follows,

"Vessel A" Requirements			
No	**Variables**	**Value**	
1	P_total	5000	kW
2	P_prop	4000	kW
3	P_aux	1000	kW
4	Loading Point	100	%
5	P_demand	5000	kW
6	P_prop,demand	4000	kW
7	P_aux,demand	1000	kW

Table 4.1 Vessel "A" power demand details

The operational profile of the vessel is not required to be defined, but if one would like to extend the analysis into a real operational profile, then the model and methodology are capable of doing that. However, such analysis is out of the scope of this project.

4.2 Diesel Mechanical (DM) Plant Results

The analysis has to be done in accordance with the evaluation methodology. The first parameter to investigate is the number of engines. However, changing the number of diesel engines sometimes have an impact on the the changes of the nominal engine speed. Therefore, before going directly to the number of engines, it is best to do the engine speed analysis to have a better understanding of the nominal speed effect on the propulsion system noise.

4.2.1 Varying Engine Speed

Propulsion System Arrangement (1-DM)		
No	**Parameters**	**Description**
1	**Transmission Type**	Diesel-Mechanical
2	**Number of Shafts**	1
3	**Number of Diesel Engines**	1
4	**Number of Gearboxes**	1-SISO
5	**Number of Electric Motors**	0
6	**Number of Rooms**	1

Prime Movers 1			
No	**Variables**	**Value**	
1	**Pb**	4000	kW
2	**N_nom**	[300:1000]	rpm
3	**z**	8	cylinders
4	**alpha**	0	degree
5	**cm**	10.25	m/s

Reduction Gearbox SISO			
No	**Variables**	**Value**	
1	**Pgb**	4000.00	kW
2	**N_prop**	200.00	rpm
3	**I**	3.80	
4	**Mgb**	190.99	kN

Table 4.2 Input Data for the Engine Speed Changing

The table above shows all the inputs to the model for the varying engine speed simulation. The basic configuration of diesel-mechanical is a plant with a single shaft and single engine. The range of speed chosen based on the minimum and maximum limit on the medium speed diesel engine [39]. There are two equations to predict the sound power level of a diesel engine, one for the engine with the nominal speed above 700 rpm (see equation 3.1) and the other for the nominal speed lower equal than 700 rpm (see equation 3.2). There are no sound level results above 950 rpm because with such engine speed the shape ratio of the engine will become less than one.

The sound level results show a great difference between the two formulas. It is somehow not a plausible result, especially the contrast of sound level at 700 rpm and 725 rpm. The noise level of engine with 700 rpm result is 5 dBA higher compared to the engine with 725 rpm. This significant difference leads to the further investigation regarding the formula to calculate the diesel engine sound power level below the nominal speed of 700 rpm. There is no details explanation of what types of engines and experiments used to create the empirical formula [31]. However, based on the equation, the high sound level is due to the contribution from the constant B. This constant is added to the equation if the diesel engine has a nominal speed below 700 rpm. The high value of the constant might be because of the coefficients were derived from the sound measurement of engines with power per cylinder above 1500 kW/cyl, which of course have higher sound power level compared to the engines used in this analysis . When investigating the medium speed diesel engine, the equation 3.2 maybe not valid for this type of engine. The equation 3.1 could be more applicable, even though the nominal speed is below 700 rpm. Another solution is probably to update the constant B from the literature since the formula was derived almost 20 years ago. The technology for medium-speed diesel has been advanced so far since that time.

The same thing also happened in the structure-borne noise level results. There is a significant decline, approximately 5 dB, for an engine with the nominal speed below 600 rpm. However, the structure-borne noise level difference between the nominal speed of 625 rpm and 600 rpm is too high and not plausible. There is no detailed explanation from the SNAME on why there is a significant reduction in the empirical constant for an engine with the nominal speed below 600 rpm. One possibility is that the SNAME measure the vibration level of the engines by clustering them into a certain speed range and average the results for each range.

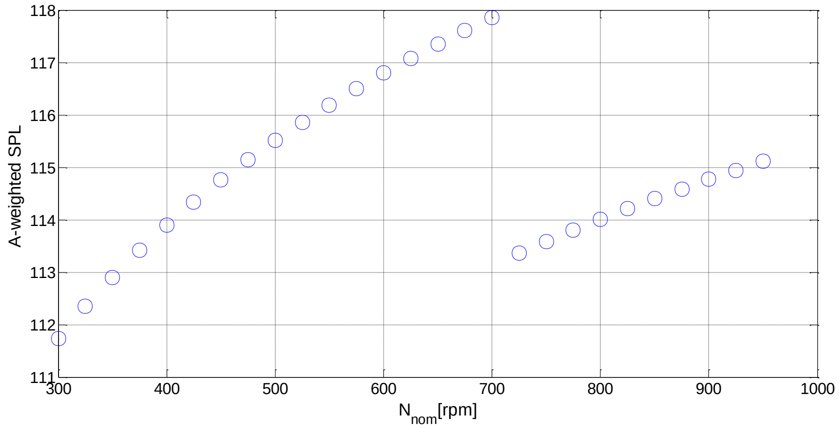


Figure 4.1 Diesel Engine nominal speed changing Airborne Noise results

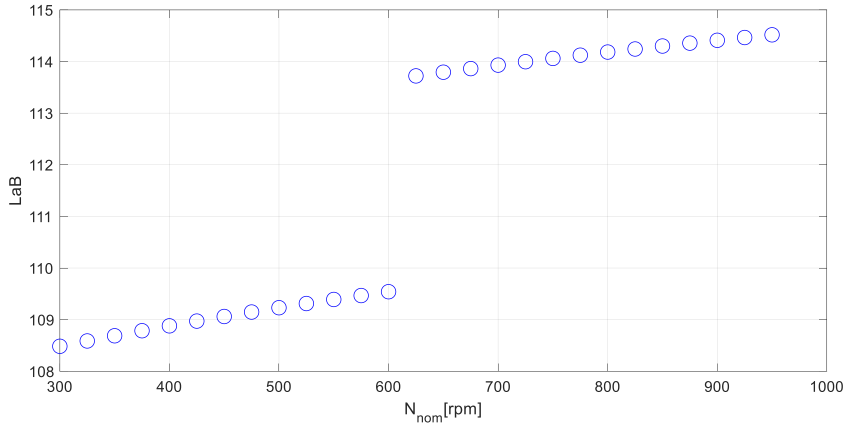


Figure 4.2 Diesel Engine nominal speed changing Structure-borne Noise results

4.2.2 Varying the Number of Engines and Shafts

		2-DM	3-DM	4-DM
No	**Parameters**	**Description**	**Description**	**Description**
1	**Transmission Type**	DM	DM	DM
2	**Number of Shafts**	1	2	2
3	**Number of Diesel Engines**	2	2	4
4	**Number of Gearboxes**	1-DISO	2-SISO	2-DISO
5	**Number of Electric Motors**	0	0	0
6	**Number of Rooms**	1	1	1

Table 4.3 Input data for the Number of engines and Shafts Changing

As explained in the previous chapter, the number of diesel engines and shafts are related to each other. The number of shafts also determine the number of gearboxes and vice versa. In that sense, there is no need to analyse the number of gearboxes as a parameter in the case study. The number of diesel engines per shaft affecting the gearbox types used. If there are multiple engines in one shaft, one should use the DISO-gearbox type. The compartment dimensions for each configuration also have different values due to the variation of the equipment dimensions.

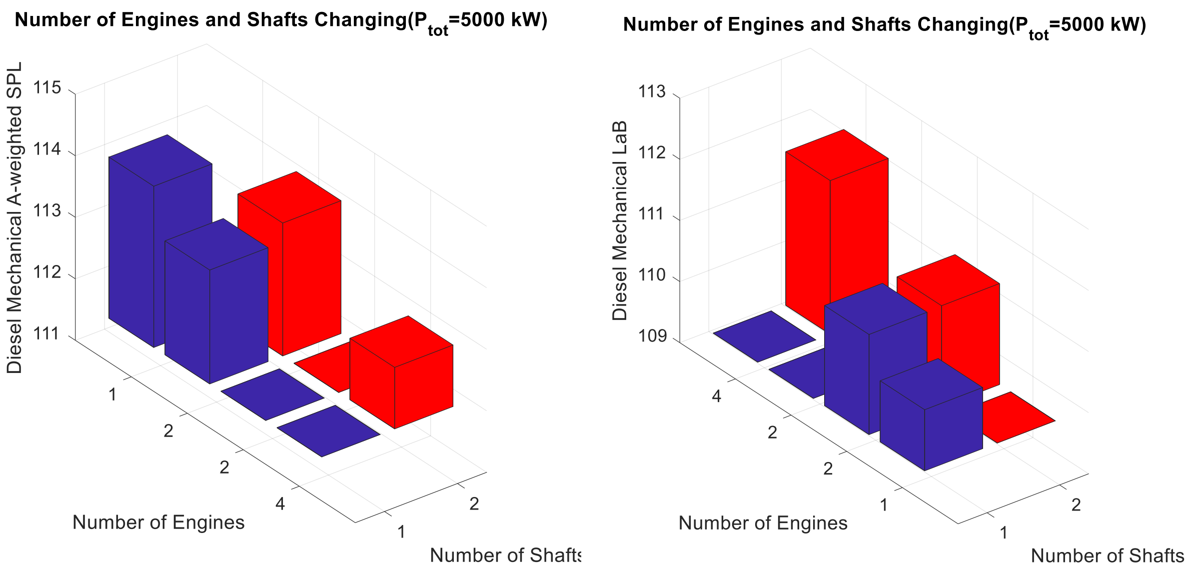


Figure 4.3 Number of engines and Shafts Changing Airborne and Structure-borne Noise Results

The figure above indicates that the increase in the number of diesel engines results in the reduction of the airborne noise level by 0.65 dBA. Even though the engine speed is increasing in the multiple engines arrangement (2-DM), the power per engine is lower, and the room volume is larger which makes the absorption by the surfaces much higher. The twin shaft plant (3-DM) shows higher noise level compared to the multiple engines arrangement with a single shaft. This result is not surprising because there are two gearboxes installed in the twin shaft arrangement. The multiple engines with multiple shafts plant (4-DM) configuration produces the lowest noise. It generates airborne noise of 112.2 dBA.

On the other hand, the structure-borne noise results showed the total opposite of the airborne noise. Increasing the number of diesel engines will give an increase to the structure-borne noise level. The lowest structure-borne noise is achieved by using the single-engine arrangement. The two engines arrangements (2-DM and 4-DM) give almost the same vibration level in the receiver location. The multiple engines with multiple shafts give the highest level, which equals to 112 dB.

4.2.3 Varying the Number of Rooms

		2-DM	3-DM	4-DM
No	**Parameters**	**Description**	**Description**	**Description**
1	**Transmission Type**	Diesel-Mechanical	Diesel-Mechanical	Diesel-Mechanical
2	**Number of Shafts**	1	2	2
3	**Number of Diesel Engines**	2	2	4
4	**Number of Gearboxes**	1-DISO	2-SISO	2-DISO
5	**Number of Electric Motors**	0	0	0
6	**Number of Rooms**	[1:2]	[1:2]	[1:2]

Table 4.4 Input data for the number of rooms changing

The previous analysis always uses only one compartment, either for the propulsion driver room or for the diesel generator room. The effect of increasing the number of rooms is investigated here. However, in single engine single shaft (1-DM) plant, it is not possible to have more than one room.

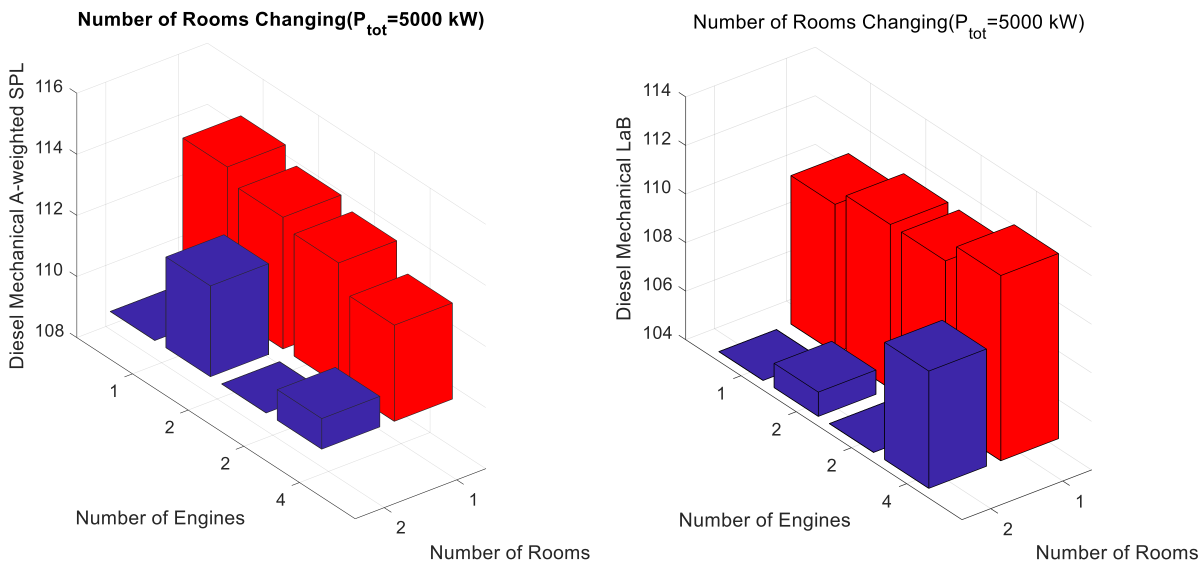


Figure 4.4 Number of rooms changing Airborne Noise and Structure-borne Noise results

The arrangements with two rooms give good acoustic performances compared to a single room. In an arrangement with two engines and two shafts, the decrease of the SPL is 1.4 dBA while in the four engines configuration the reduction is higher, almost 2 dBA. A significant reduction is due to the reduced number of diesel engines per compartment. The transmission loss by the bulkhead also plays a significant role because it makes the airborne noise contribution from the adjacent room is negligible.

The structure-borne noise also benefits from adding the number of rooms. A higher improvement can be seen from the figure. The twin shaft arrangement with two rooms has a reduction of 5.8 dB and the four engines arrangement with two rooms improvement is 2.8 dB. These reductions are quite high thanks to the intersection loss of the bulkhead. Although the bulkhead separates the room, the contribution of vibration level from the adjacent room still needs to be considered. The drawback of adding compartment is that one should provide wider overall dimensions to the engine room.

4.2.4 Varying Loading Point

		2-DM	3-DM	4-DM
No	**Parameters**	**Description**	**Description**	**Description**
1	**Transmission Type**	DM	DM	DM
2	**Number of Shafts**	1	2	2
3	**Number of Diesel Engines**	2	2	4
4	**Number of Gearboxes**	1-DISO	2-SISO	2-DISO
5	**Number of Electric Motors**	0	0	0
6	**Number of Rooms**	1	1	1

Ship Requirements			
No	**Variables**	**Value**	
1	**P_total**	5000	kW
2	**P_prop**	4000	kW
3	**P_aux**	1000	kW
4	**Loading Point**	[10:10:100]	%
5	**P_demand**	[0:5000]	kW
6	**P_prop,demand**	[0:4000]	kW
7	**P_aux,demand**	[0:1000]	kW

Table 4.5 Input data for loading point changing

The part load conditions chosen are between 10% and 100% load. Although it is not common to have a 10% loading condition, it is assumed that the vessel “A” has this kind of operation. Moreover, the vessel “A” is assumed to have the ability to sail with the trailing shaft in the multiple gearboxes configurations. All four diesel-mechanical arrangements will be evaluated with the varying loading points. In this analysis, the number of rooms is equal to one. This assumption is also applied to the diesel-electrical and diesel-hybrid plant.

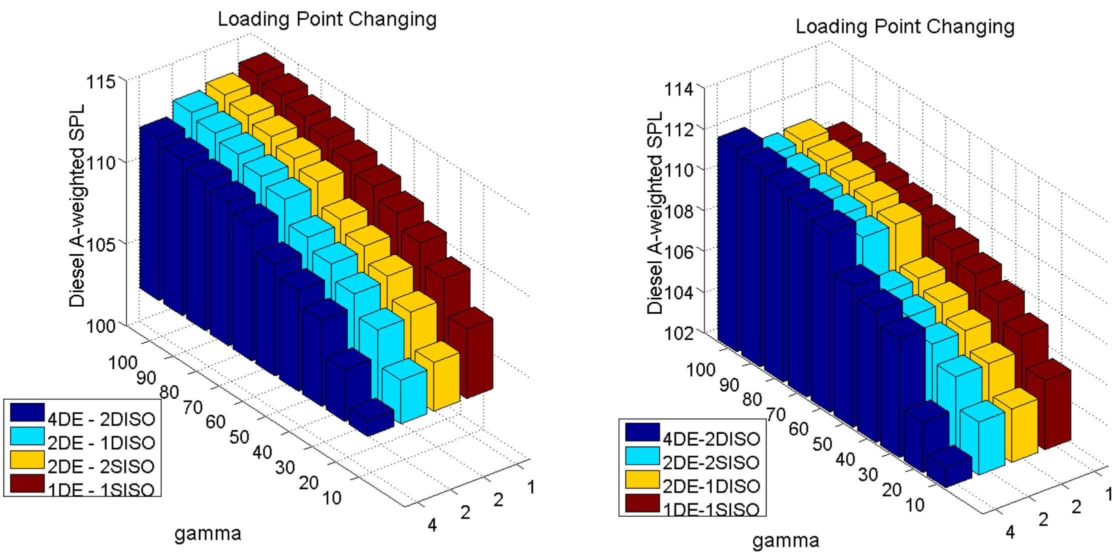


Figure 4.5 Loading point changing Airborne and Structure-borne Noise

The lowest airborne noise level for all part-load conditions is achieved by implementing the four engines with two DISO gearboxes. The highest airborne noise is produced by one engine with one SISO gearbox. The multiple engine configurations have lower noise because it is possible to turn off other engines when the operation only requires a small amount of power. On the contrary, the single-engine configuration is not capable of doing so. The results also indicate that two engines configurations with one DISO gearbox produce less noise rather than use two SISO gearboxes.

For the structure-borne noise, the single-engine arrangement produces the lowest value for loading point above 50%. This result is expected because all the engines need to operate if the loading point is above 50%. When the loading point is ranging from 10% to 20%, the four engines configuration has the lowest vibration due to only one small engine operates.

4.3 Diesel Electrical (DE) Plant Results

In the diesel-electrical plan, the diesel generator sets, and the electric motors are considered as the major sources of noise. They are located separately in this plant because there are many other supporting equipment between them, i.e., switchboard, control room, transformer (optional), inverter and PWM converter. Therefore, there is no airborne and structure-borne noise interaction between each other. The analysis will be done separately between them. The maximum number of diesel generator sets in the diesel-electrical plant is higher than the diesel-mechanical plant.

4.3.1 Varying Engine Speed

Propulsion System Arrangement		
No	**Parameters**	**Value**
1	Transmission Type	Diesel-Electrical
2	Number of Shafts	1
3	Number of Diesel Generator Sets	1
4	Number of Gearboxes	0
5	Number of Electric Motors	1
6	Number of Rooms	1

Diesel Genset 1			
No	**Variables**	**Value**	
1	Pb	5000	kW
2	N_nom	[300:1000]	rpm
3	z	8	cylinders
4	alpha	0	degree
5	cm	10.25	m/s

Electric Motor 1			
No	**Variables**	**Value**	
1	Pem	4000.00	kW
2	N_em	[200:700]	rpm
3	Mem	190.99	kN
4	Type	AC	
5	Casing	TEFC	
6	s	0.5	
7	vt	23.28	m/s
8	tau_e	16.14	kN/m2

Table 4.6 Input data for rpm changing

The minimum speed of the electric motor is equal to the propeller speed. In the diesel-electrical plant, it is assumed that the Variable Frequency Speed Driver is always installed to replace the function of the gearbox. The motor sound power level is increasing logarithmically towards higher nominal speed. It is expected because the expression to predict the motor sound power level depends on the nominal speed and the nominal power of the motor. The noise will be higher when the power remains constant, but the speed is increasing.

The diesel generator sets sound level mainly influenced by the sound power level from its prime mover. One can say that the generator sound being masked by the diesel engine noise. As explained earlier, the huge difference between the noise level above 700 rpm and below 700 rpm is due to the constant B in equation 3.2.

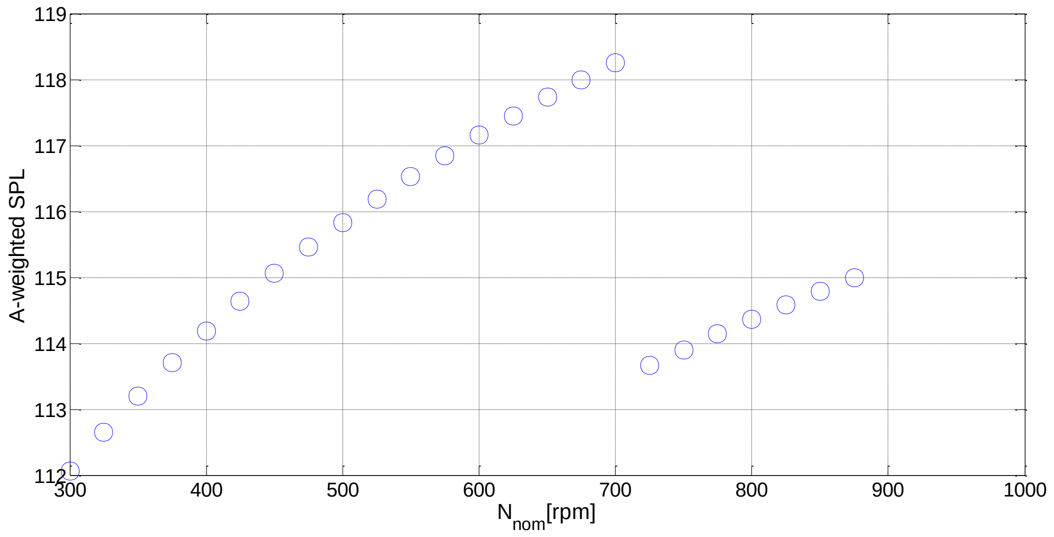


Figure 4.6 Diesel Genset nominal speed changing Airborne Noise results

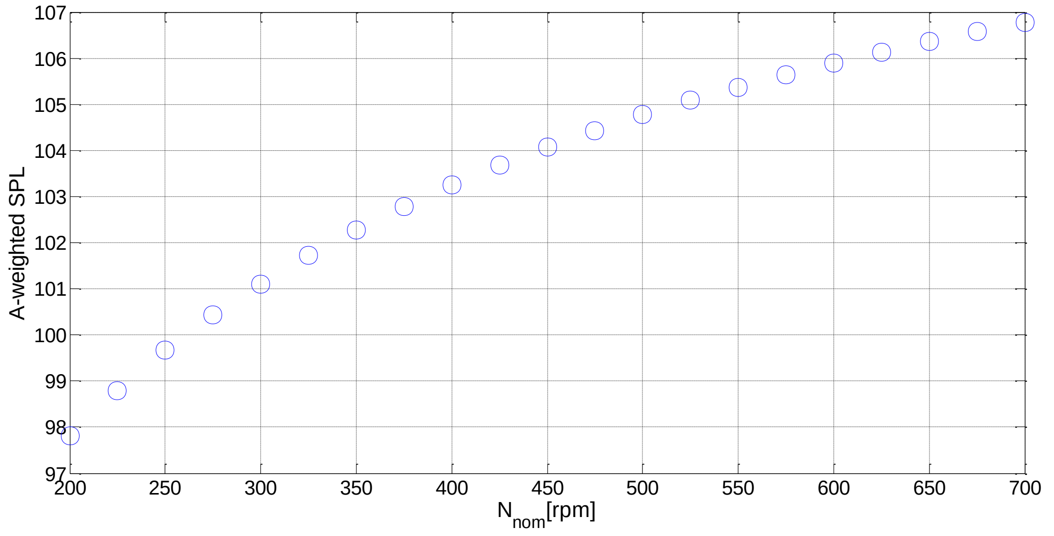


Figure 4.7 Electric motor nominal speed changing Airborne Noise results

The diesel-generator structure-borne noise results show similar characteristics to the diesel engine results. There is a big difference between the engine with speed below 600 rpm and above 600 rpm. The reason is similar to the explanation in the diesel-mechanical simulation. The electric motor structure-borne noise source level is directly proportional to its speed. The higher the nominal speed, the higher the vibration level will be. Unlike the diesel engine, the electric motor source level does not have speed clustering to calculate its structure-borne noise level. Therefore, the results are continuous, and there is no big jump between adjacent engine speed points.

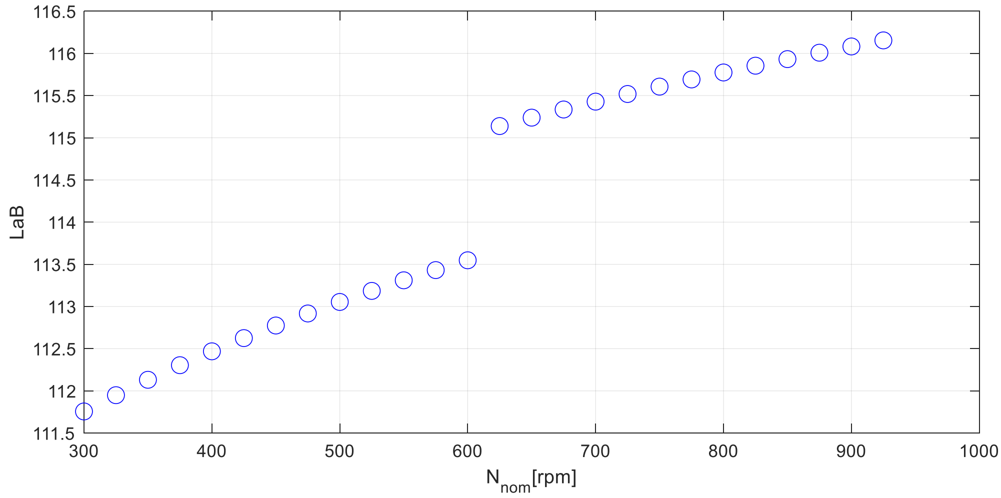


Figure 4.8 Diesel Genset nominal speed changing Structure-Borne Noise results

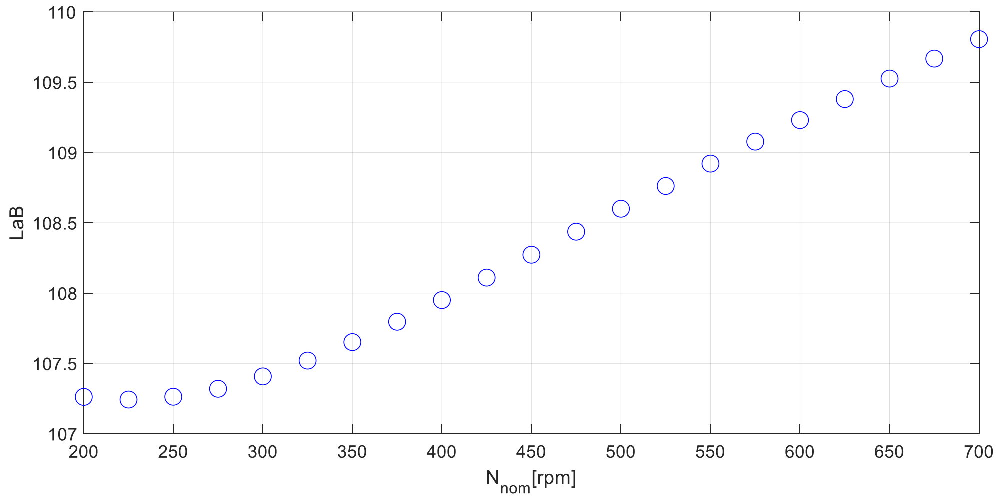


Figure 4.9 Electric Motor nominal speed changing Structure-Borne Noise results

4.3.2 Varying the Number of Engines and Shafts

		1-DE	2-DE	3-DE	4-DE	5-DE	6-DE
No	**Parameters**	**Desc.**	**Desc.**	**Desc.**	**Desc.**	**Desc.**	**Desc.**
1	**Transmission Type**	DE	DE	DE	DE	DE	DE
2	**Number of Shafts**	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
3	**Number of DG**	1	2	3	4	5	6
4	**Number of GB**	-	-	-	-	-	-
5	**Number of EM**	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
6	**Number of Rooms**	1	1	1	1	1	1

Table 4.7 Input for the number of engines and shafts changing simulations

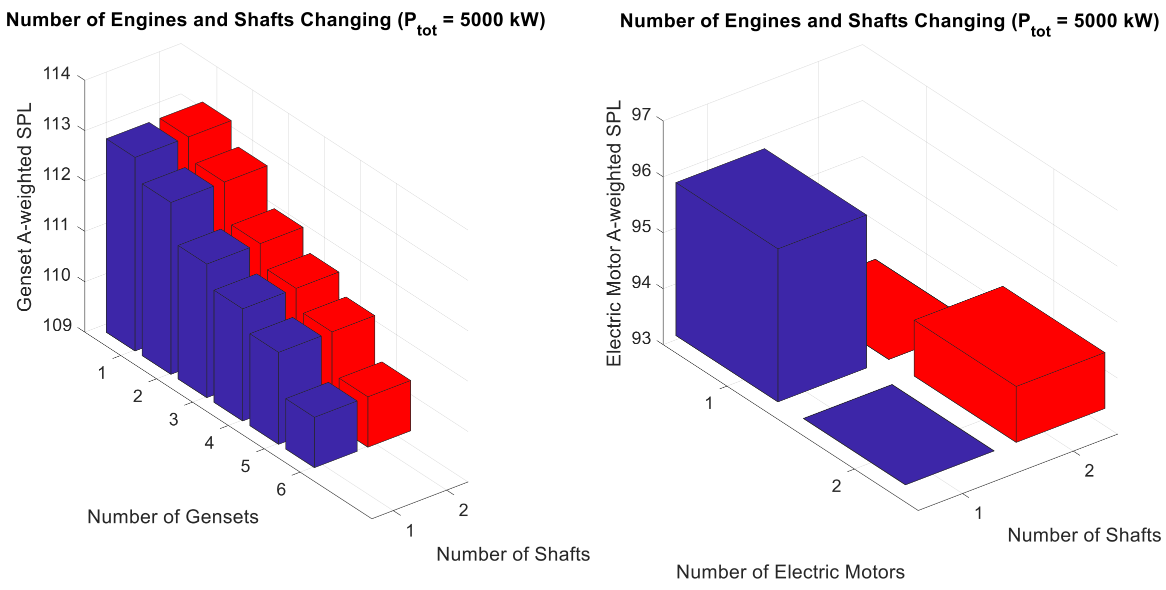


Figure 4.10 Diesel Gensets & Electric Motor number of engines and shafts changing Airborne Noise results

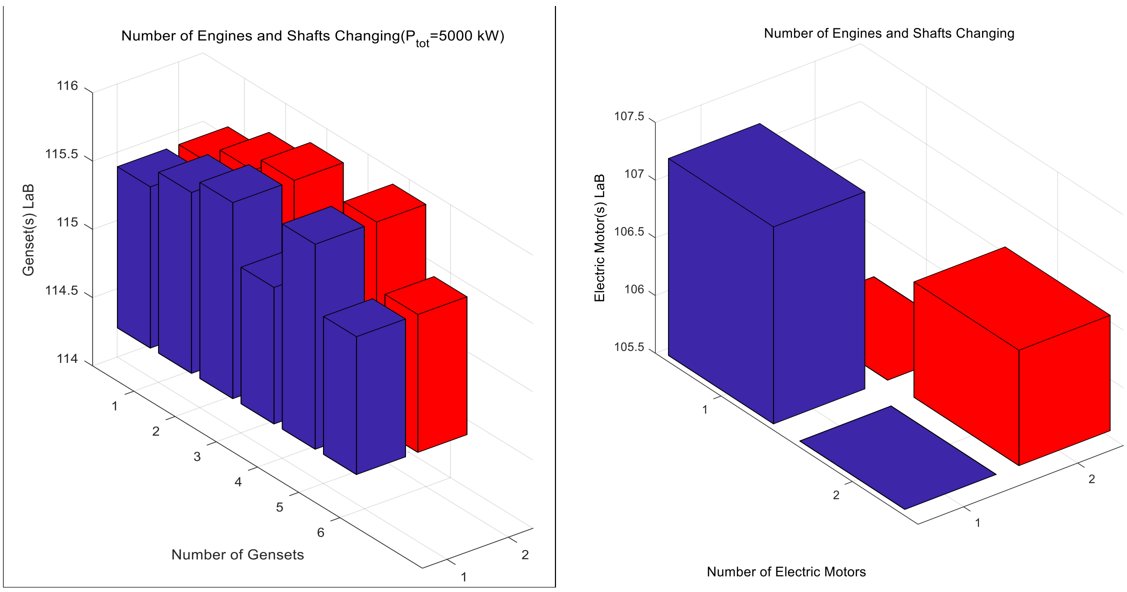


Figure 4.11 Diesel Gensets & Electric Motor number of engines and shafts changing Structure-borne Noise results

The number of shafts does not depend on the number of diesel generator sets in the diesel-electrical plant. It is related to the number of electric motors. In that sense, it is possible to use more than two diesel generator sets by using one shaft, but only one electrical motor allowed. Therefore, from the figures above, the number of shafts does not give any effect to the noise level in diesel genset compartment. It can be seen that increasing the number of diesel generator sets will lower the airborne noise level, which is similar to the diesel-mechanical plant. The lowest airborne noise level is produced by six diesel generator sets while the highest one is generated by the single engine arrangement.

The airborne noise levels are 112.8 dBA and 113.6 dBA for six engines configuration and one engine configuration respectively.

On the contrary, the structure-borne noise results seem to have a different effect. The decision in the early design phase by increasing the number of diesel generator sets will increase the vibration level. However, the structure-borne noise level of the four and six engines configurations are lower than the five and three engines configurations. These unusual results are due to the arrangement of the generator set. When the total number of diesel generator sets is an odd number, there will be an engine in the centre-line of the room. This location is directly facing the receiver location, which makes the distance between the engine and receiver location is closer. Therefore, the contribution of vibration level from the engine in the centre makes the total structure-borne noise level at the receiver location higher.

Interesting results can be found in the electric motor sound levels. It is obvious that increasing the number of it will proportionally increase the number of shafts. Since the maximum possible shaft is only two, the comparison is only between the single electric motor and the twin motors. From the results, it is evident that increasing the number of shafts will decrease the airborne and structure-borne noise level up to 2 dBA and 0.6 dB respectively. These reductions happen due to a larger room in two shafts arrangement and the lower noise level from each motor.

4.3.3 Varying the Number of Rooms

		1-DE	2-DE	3-DE	4-DE	5-DE	6-DE
No	Parameters	Desc.	Desc.	Desc.	Desc.	Desc.	Desc.
1	Transmission Type	DE	DE	DE	DE	DE	DE
2	Number of Shafts	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
3	Number of DG	1	2	3	4	5	6
4	Number of GB	-	-	-	-	-	-
5	Number of EM	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
6	Number of Rooms	1	2	2	2	2	2

Table 4.8 Input data for the number of rooms changing simulations

Similar to the diesel-mechanical plant, adding the number of compartments also produce a very good acoustical performance in the diesel-electrical plant. The airborne and structure-borne noise levels are reduced significantly. Nevertheless, this will create a consequence to the overall width of the diesel generator sets compartments and the electric motor compartments. The airborne noise levels reductions in the diesel generator set compartment are 1.5 dBA, 2 dBA and 1.9 dBA for two, four and six generator sets respectively while in the electric motor compartment the airborne noise is reduced by 1.7 dBA. Similar to diesel-mechanical, the effect of increasing the number of compartments will be more evident in the structure-borne noise levels. The reductions are 5.5 dB, 6.7 dB, 3 dB and 5.6 dB for two, four, six generator sets and two electric motors respectively.

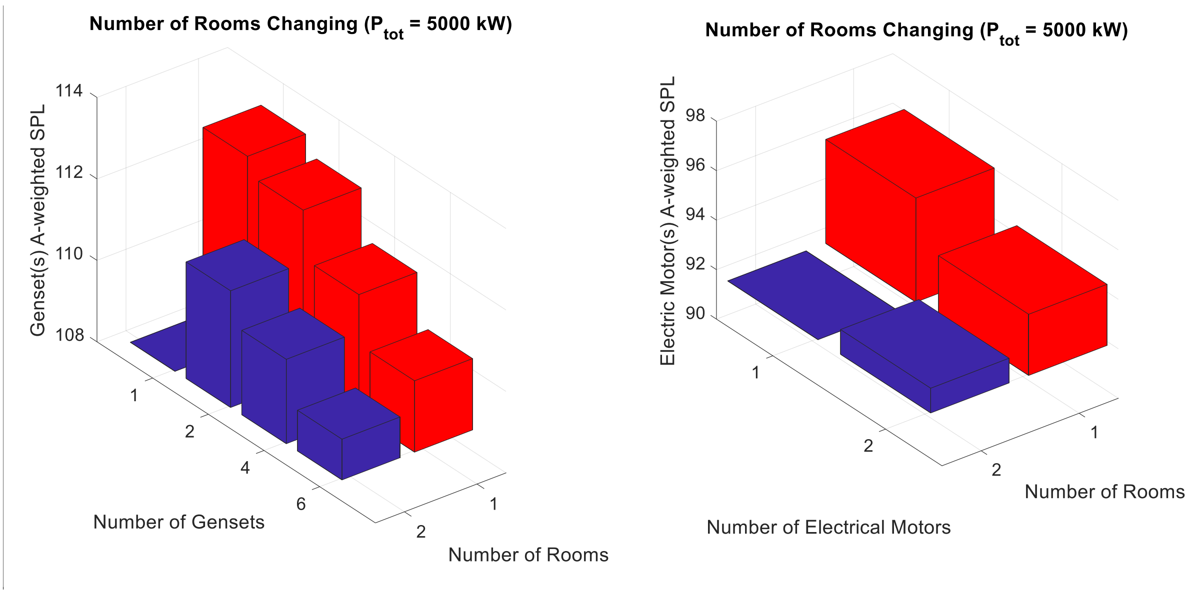


Figure 4.12 Diesel Genset & Electric Motor number of rooms changing Airborne Noise results

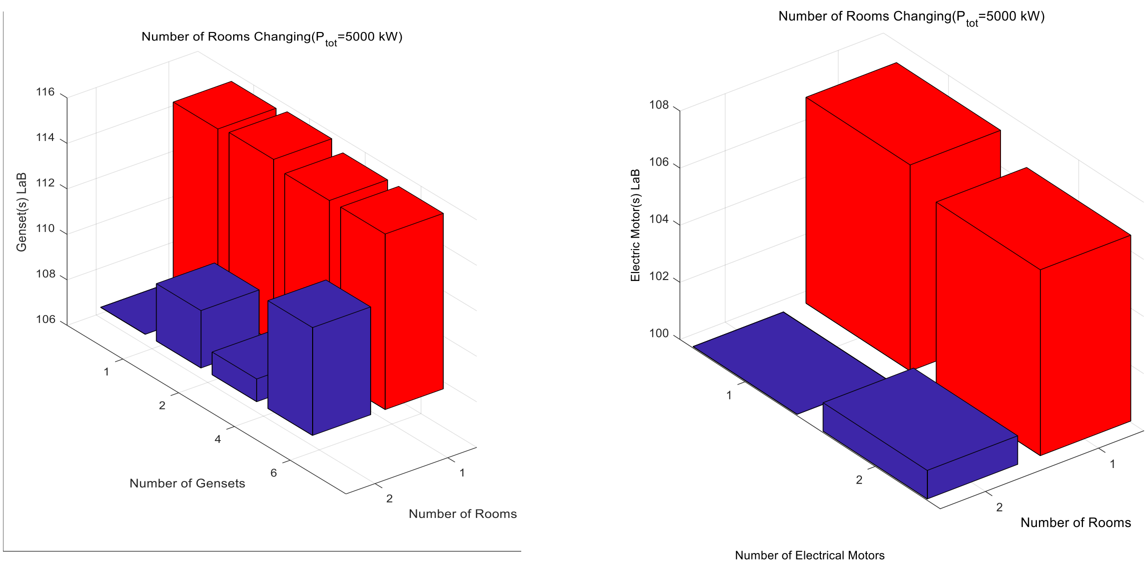


Figure 4.13 Diesel Genset & Electric Motor number of rooms changing Structure-borne Noise results

4.3.4 Varying Loading Point

		1-DE	2-DE	3-DE	4-DE	5-DE	6-DE
No	**Parameters**	**Desc.**	**Desc.**	**Desc.**	**Desc.**	**Desc.**	**Desc.**
1	**Transmission Type**	DE	DE	DE	DE	DE	DE
2	**Number of Shafts**	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
3	**Number of DG**	1	2	3	4	5	6
4	**Number of GB**	-	-	-	-	-	-
5	**Number of EM**	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
6	**Number of Rooms**	1	1	1	1	1	1

Ship Requirements			
No	**Variables**	**Value**	
1	P_total	5000	kW
2	P_prop	4000	kW
3	P_aux	1000	kW
4	Loading Point	[10:10:100]	%
5	P_demand	[0:5000]	kW
6	P_prop,demand	[0:4000]	kW
7	P_aux,demand	[0:1000]	kW

Table 4.9 Input data for loading point changing simulations

The diesel generator and the electric motor results show that with more engines one can have better acoustic performance in part load total conditions. More engines mean that more possibilities to operate with a smaller engine in the part load condition. This statement is proven in this analysis. All the minimum A-weighted sound pressure levels and acceleration levels are produced by the operation conditions with the least working equipment. For instance, when the loading point is equal to 40%, the part load total demand equals to 2000 kW. The minimum sound level of four diesel generator sets arrangement is produced by setting one diesel generator set to maximum power, 1250 kW, and the second diesel generator set to 750 kW. When the equal share load is being implemented, it generates a higher noise level. However, the difference is really small which makes operating in equal share load condition is a better option (fuel efficiency reason).

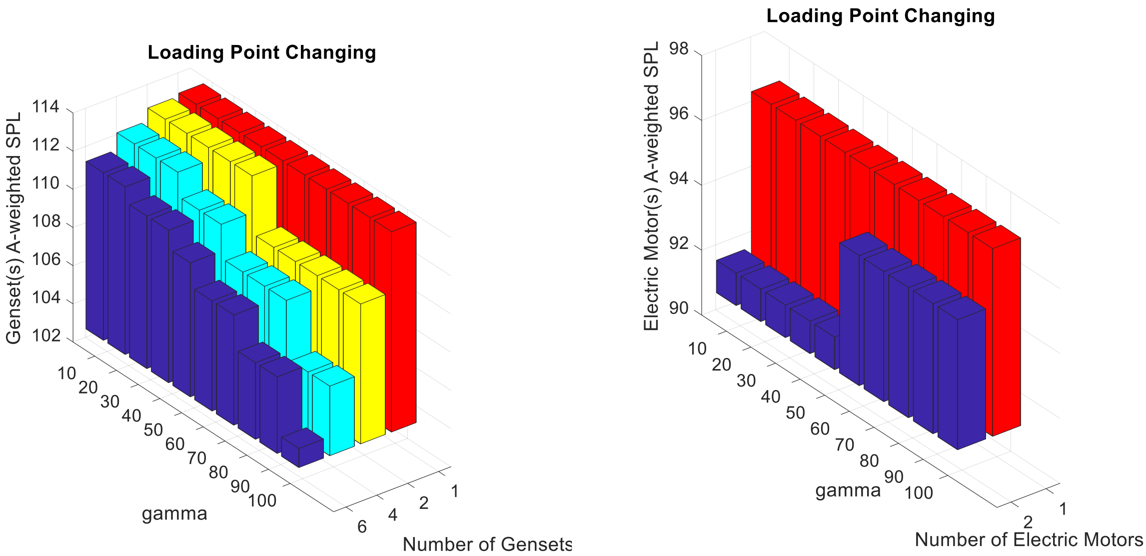


Figure 4.14 Diesel Genset & Electric Motor loading point changing Airborne Noise results

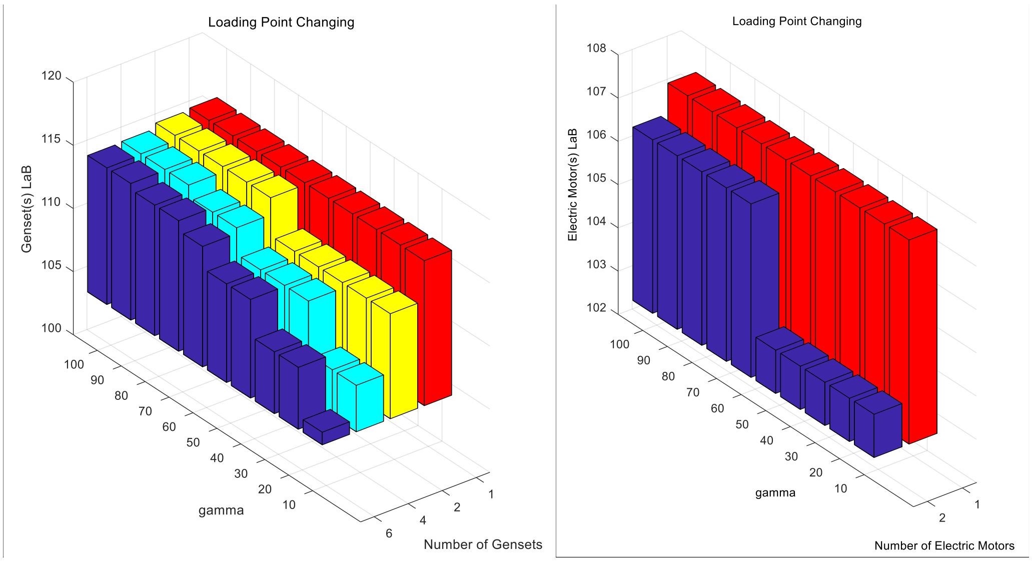


Figure 4.15 Diesel Genset & Electric Motor loading point changing Structure-borne Noise results

The single arrangement shows constant sound level over varying loading factor for both the diesel generator set and the electric motor. The first one is due to the characteristic of the generator set operation. It should operate at a constant speed regardless of the power required so that the current frequency stays constant. In that sense, with a constant speed in part load condition, the noise level will always stay the same (see equation 3.2). The latter one is due to the parameters that are used to calculate the noise source levels for the electric motor. It only takes into account the nominal speed and the nominal power (see equation 3.10). Therefore, if the electric motor is operating under part load conditions, then the sound pressure level will always be constant over the varying load demand.

4.4 Diesel-Hybrid Plant Results

As explained in the previous subchapter 3.4.2, the hybrid systems can do the PTI, the PTO and the PTH modes. It is assumed that the propulsion drivers and the generators are in a separate compartment. The analysis will be performed for both compartments. There are two new parameters, delta (Δ) and theta (θ), will be introduced in the diesel-hybrid plant. The first one defines how the total installed power is splited between the main propulsion drivers and the main electricity producers. The second one determines the split between the diesel engine and the electric motor. It is necessary to investigate how the noise is affected by these parameters. The designer can always have an option to have a configuration with larger diesel generator sets and smaller propulsion driver or smaller diesel engine with larger electric motor. Considering it is not possible to have three inputs to the gearbox, one can neglect the number of shafts parameter in the diesel-hybrid. This is because the number of shafts will always be the same as the number of diesel engines in the propulsion driver. Therefore, the number of shafts parameter effect will not be presented in this analysis.

4.4.1 Number of Engines and Delta Changing

		1-DH	2-DH
No	**Parameters**	**Desc.**	**Desc.**
1	**Transmission Type**	DH	DH
2	**Number of Shafts**	1	1
3	**Number of DE**	1	2
4	**Number of GB**	1-DISO	2-DISO
5	**Number of EM**	1	2
6	**Number of Rooms**	1	1
7	**Delta**	[0.5:0.1:1]	[0.5:0.1:1]

Table 4.10 Input data for delta changing

The nominal power of the diesel engines, electric motor, and diesel generator sets depends on the delta value. The range of the delta value is from 0.5 – 1. It is possible to have it less than 0.5, but this range is already enough to show the effect of delta. The nominal power of the equipment can be expressed as follows,

$$P_{b,de} = \Delta * P_{tot} \tag{4.1}$$

$$P_{b,dg} = (1 - \Delta) * P_{tot} \tag{4.2}$$

The electric motor nominal power in diesel hybrid with delta changing could be the minimum power or the maximum one. The minimum power is when the electric motor nominal power is enough to boost the diesel engine power to deliver the maximum propulsion demand. The maximum is simply when the electric motor is equal to the diesel generator sets total power. This analysis only takes into account the electric motor with the minimum power.

$$P_{b,em,min} = P_{prop} - P_{b,de} \tag{4.3}$$

$$P_{b,em,min} = P_{b,dg} \tag{4.4}$$

High delta value makes the noise level in the propulsion driver room higher. This statement is reasonable because by increasing the delta, the power of the diesel engine becomes larger which results in a greater sound level. As for the number of diesel engines and the number of electric motors, it gives less airborne noise with more engines and a higher level of structure-borne noise. The diesel genset room gives similar results to the diesel electrical plant results. It means that the diesel-electrical model can be used in the diesel-hybrid plant analysis. It is expected because the arrangement of diesel generators is the same, only the nominal power per engine is different.

An interesting result can be found when the delta is equal to 0.7 for the structure-borne noise. It can be seen that the vibration level at 0.7 is higher than the vibration level at 0.8. This difference happened because when the delta is equal to 0.8, the electric motor power is equal to zero, which means less equipment is installed in the propulsion driver room. The empirical formula derived to predict the motor structure-borne noise level is conservative. The term conservative means the source level tends to overestimate the prediction. Therefore, the reduction in noise is quite significant when there is no electric motor. However, this is not the case in the airborne noise. The diesel engine and the gearbox dictate the airborne noise level inside the engine room. These engines are masking the electric motor sound level.

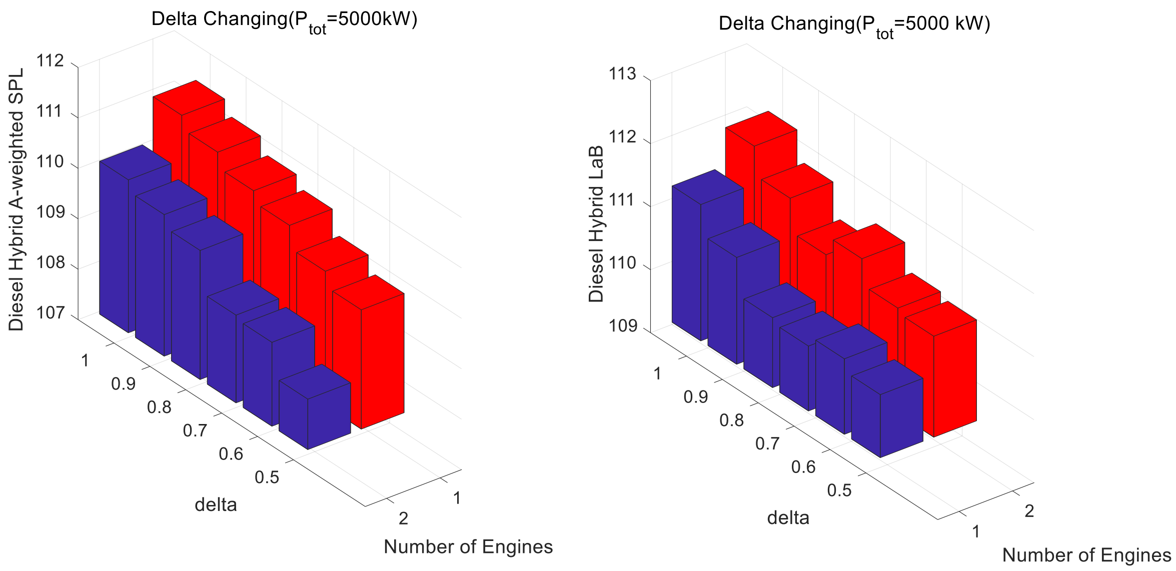


Figure 4.16 Diesel-Hybrid number of engines and delta changing Airborne Noise and Structure-borne Noise results

4.4.2 Varying the Number of Rooms

		1-DH	2-DH
No	**Parameters**	**Desc.**	**Desc.**
1	**Transmission Type**	DH	DH
2	**Number of Shafts**	1	1
3	**Number of DE**	1	2
4	**Number of GB**	1-DISO	2-DISO
5	**Number of EM**	1	2
6	**Number of Rooms**	1	2
7	**Delta**	[0.5:0.1:1]	[0.5:0.1:1]

Table 4.11 Input data for numbers of room changing

The results presented here are only the propulsion driver room changing. It is not necessary to analyse the diesel genset room again since it will have the same characteristic as diesel genset in the electrical plant. The results can be seen in the figure below. It is possible to reduce the noise inside the engine compartment by increasing the number of rooms. The noise levels are always lower for the arrangement with two compartments over varying delta.

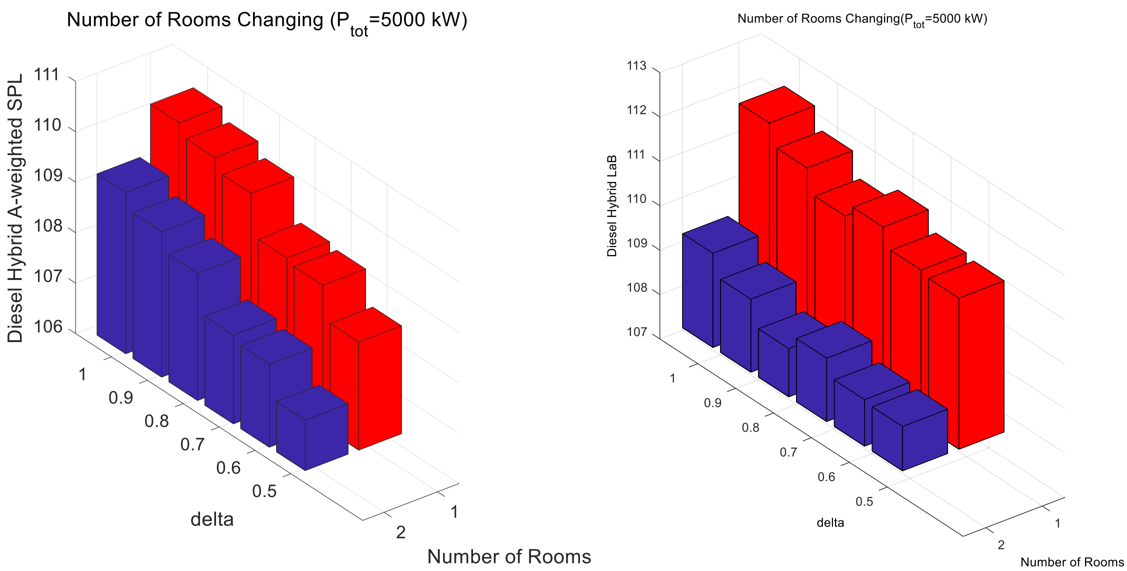


Figure 4.17 Diesel Hybrid number of rooms changing Airborne and Structure-borne Noise results

4.4.3 Varying Loading Point

		1-DH	2-DH
No	**Parameters**	**Desc.**	**Desc.**
1	**Transmission Type**	DH	DH
2	**Number of Shafts**	1	1
3	**Number of DE**	1	2
4	**Number of GB**	1-DISO	2-DISO
5	**Number of EM**	1	2
6	**Number of Rooms**	1	1
7	**Delta**	0.5	0.5

Ship Requirements			
No	**Variables**	**Value**	
1	**P_total**	5000	kW
2	**P_prop**	4000	kW
3	**P_aux**	1000	kW
4	**Loading Point**	[10:10:100]	%
5	**P_demand**	[0:5000]	kW
6	**P_prop,demand**	[0:4000]	kW
7	**P_aux,demand**	[0:1000]	kW

Table 4.12 Input data for loading point changing

When changing the loading point the delta will be kept constant instead of varying. The chosen delta is 0.5. It means that the total power will be provided half by the diesel engine and another half generated by the generator set. The results in propulsion driver compartment show a very low sound level in low part load condition, between 10% – 30%. It happens because only the electric motor works in this load range. It gets the power from the diesel generator sets. In this range, the diesel generator sets will provide the auxiliary and the propulsion load. The operation mode is the PTH mode. It is important to note that the diesel generator sets here have the same operation characteristics as in the electrical plant. They will create maximum sound power level regardless of the varying load. However, if it is possible to only operate with some engines instead of all engines, then the algorithm will choose to operate with less equipment. This type of operation is the case when the loading factor is between 10% - 20%. An interesting characteristic can be found when the loading points are greater than equal to 50%. At those loading points, all the machines need to operate. The algorithm will choose to operate the propulsion system with the equal load between the diesel engine(s) and the diesel generator set(s). This restriction will lead to an increase in the noise level of the propulsion system.

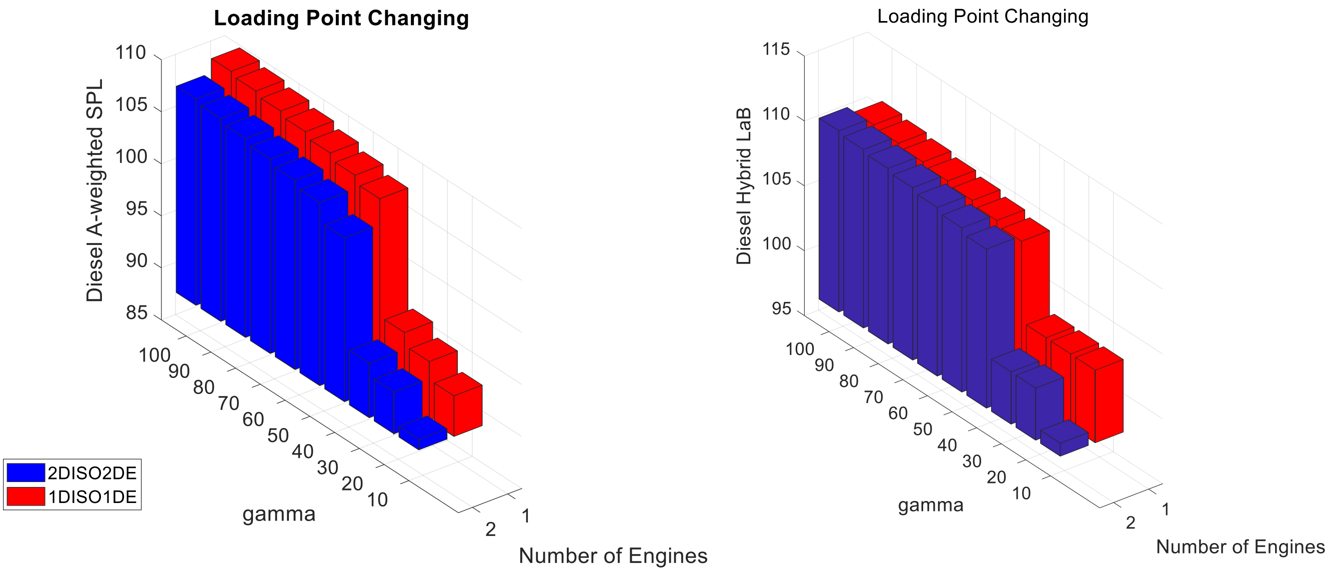


Figure 4.18 Diesel Hybrid loading point changing Airborne and Structure-borne Noise results

4.4.4 Varying the Split between the Nominal Power of Electric Motor and Diesel Engine Power

		1-DH	2-DH
No	**Parameters**	**Desc.**	**Desc.**
1	**Transmission Type**	DH	DH
2	**Number of Shafts**	1	1
3	**Number of DE**	1	2
4	**Number of GB**	1-DISO	2-DISO
5	**Number of EM**	1	2
6	**Number of Rooms**	1	1
7	**Delta**	0.5	0.5

Ship Requirements			
No	**Variables**	**Value**	
1	P_total	5000	kW
2	P_prop	4000	kW
3	P_aux	1000	kW
4	Loading Point	[10:10:100]	%
5	Pb_de	3500	kW
6	Theta	[0.125:0.125:0.875]	

Table 4.13 Input data for theta changing

Theta determines the split between the electric motor and the diesel engine nominal power. There is always an option to have a bigger electric motor and a smaller diesel engine or the other way around. If the theta is equal to zero, it means there is no diesel engine installed. The power of the diesel engine over varying theta can be expressed as follows,

$$P_{b,de\theta} = \theta * P_{b,de} \tag{4.5}$$

The electric motor power over varying theta is equal to,

$$P_{b,em\theta} = P_{prop} - P_{b,de\theta} \tag{4.6}$$

Therefore, the power of the diesel genset can be written as follows,

$$P_{b,dg\theta} = P_{tot} - P_{b,de\theta} \tag{4.7}$$

In this analysis, the theta range is 0.125 to 0.875 with an interval of 0.125. For instance, if the theta is equal to 0.875, the diesel engine power is equal to 3500 kW and the electric motor power equals to 500 kW. The simulation is done for the single and the multiple engines arrangements.

The graphs below demonstrate that the smaller the theta, the lower the noise level of the propulsion system. The noise level of the propulsion system is decreasing when the electric motor power is increasing. It is reasonable because the source level of the electric motor is lower than the diesel engine. In a low theta, the electric motor becomes the main source of noise thus masking the noise from the diesel engine. For the structure-borne noise level, the benefit obtained from decreasing the theta is more significant in the multiple engines arrangement.

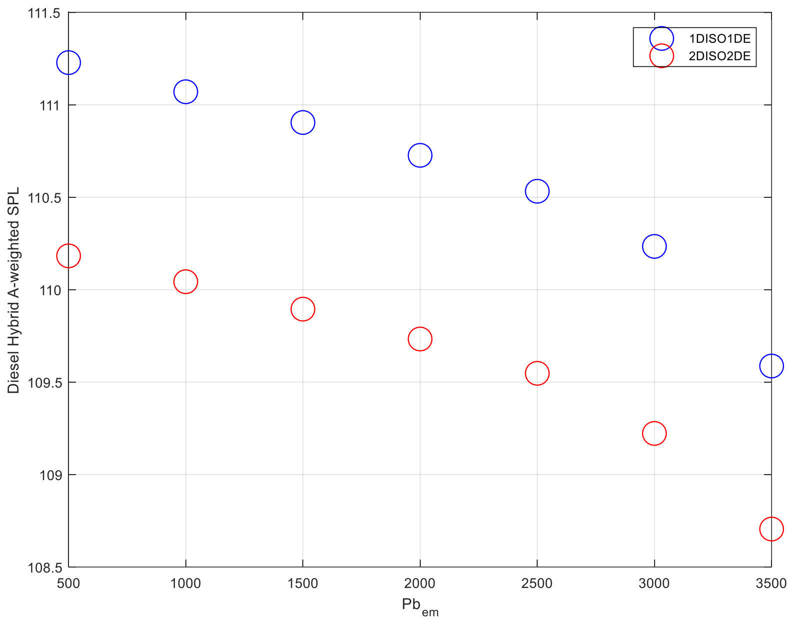


Figure 4.19 Diesel Hybrid theta changing Airborne Noise results

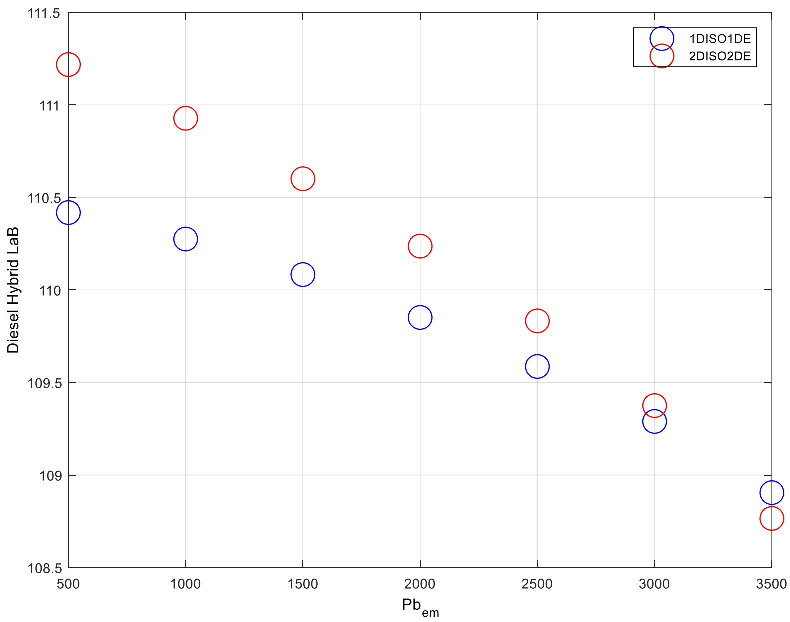


Figure 4.20 Diesel Hybrid theta changing Structure-borne Noise results

4.5 Comparison between Different Transmission Types

Diesel-Mechanical Arrangements :

		1-DM	2-DM	3-DM	4-DM
No	Parameters	Description	Description	Description	Description
1	Transmission Type	DM	DM	DM	DM
2	Number of Shafts	1	1	2	2
3	Number of Diesel Engines	1	2	2	4
4	Number of Gearboxes	1-SISO	1-DISO	2-SISO	2-DISO
5	Number of Electric Motors	0	0	0	0
6	Number of Rooms	1	1	1	1

Diesel-Electrical Arrangements :

		1-DE	2-DE	3-DE	4-DE	5-DE	6-DE
No	Parameters	Desc.	Desc.	Desc.	Desc.	Desc.	Desc.
1	Transmission Type	DE	DE	DE	DE	DE	DE
2	Number of Shafts	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
3	Number of DG	1	2	3	4	5	6
4	Number of GB	-	-	-	-	-	-
5	Number of EM	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]	[1:2]
6	Number of Rooms	1	1	1	1	1	1

Diesel-Hybrid Arrangements :

		1-DH	2-DH
No	Parameters	Desc.	Desc.
1	Transmission Type	DH	DH
2	Number of Shafts	1	2
3	Number of DE	1	2
4	Number of GB	1-DISO	2-DISO
5	Number of EM	1	2
6	Number of Rooms	1	1
7	Delta	0.5	0.5

Battery Arrangements :

		1-Bat	2-Bat
No	Parameters	Desc.	Desc.
1	Transmission Type	Battery	Battery
2	Number of Shafts	1	2
3	Number of DG	1	1
4	Number of GB	-	-

5	Number of EM	1	2
6	Number of Rooms	1	1

Ship Requirements :

Ship Requirements			
No	Variables	Value	
1	P_total	5000	kW
2	P_prop	4000	kW
3	P_aux	2000	kW
5	Loading Point	[10:10:100]	%

Table 4.14 Input data for comparison of the different transmission types

This sub-chapter will discuss the noise level comparison between different transmission types. It can be seen that the noise reduction is really small from the individual transmission type analyses. The range of noise reduction is more or less 0.5 – 2 dB both for the airborne and the structure-borne noise level. It is insignificant compared to the overall compartment noise. However, it is expected that the noise level between different transmission type will yield more significant difference. The reason is that every transmission type has its unique equipment and operation types. These will affect the overall noise level in a more significant way compared to the parameters analysed in the individual transmission type.

Four transmission types are going to be compared in this analysis, the diesel-mechanical plant, the diesel-electrical plant, the diesel-hybrid plant, and the battery plant. The vessel with the battery plant is assumed to be fully powered by the battery when sailing. Moreover, it is also assumed that there is no sound generated by the battery. In that sense, only the electric motor is the significant noise source level inside the engine room. The power source of the battery is supplied from the shore. It is assumed that all the arrangements of battery plant have one emergency diesel generator sets. The other propulsion system configurations and ship requirement details are presented in the tables above. The delta in diesel-hybrid is required to determine the split between the diesel engine power and diesel generator set nominal power in the hybrid plant. The comparison will be made in two categories, i.e., varying the number of engines and shafts and the loading point changing.

4.5.1 Varying the Number of Engines and Shafts

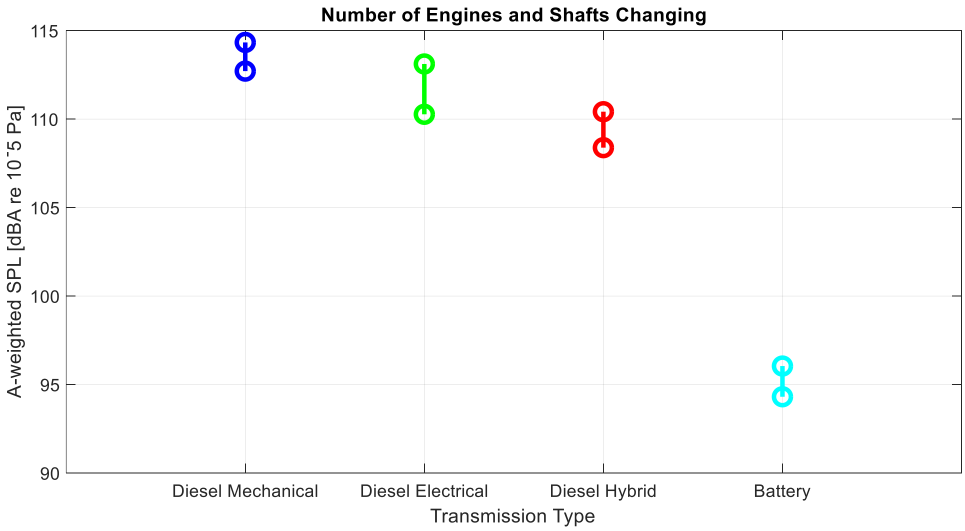


Figure 4.21 Comparison between different transmission types Airborne Noise Results

The figure above shows the results of the sound pressure level of the various possible configurations in a transmission type. The loading point is equal to 100%. The round marks illustrate the minimum and maximum sound pressure level of each transmission type. For instance, in diesel mechanical, the highest sound pressure level is generated by single engine configuration (1-DM), and four engines configuration generates the lowest one (4-DM). The two engines configurations (2-DM and 3-DM) lie between these values.

For diesel-electrical plants, there are six configurations. The lowest sound pressure is generated by the six engines arrangement (6-DE), and the highest one is produced by single engine configuration (1-DE). The interval between the highest and the lowest sound pressure level of the diesel-electrical plant is longer than the diesel-mechanical plant thanks to the maximum possible number of diesel generator sets. It is possible to have an even lower airborne noise level if there is no constraint for the number of diesel generator sets.

The diesel-hybrid plant has two configurations in this analysis. The lowest sound pressure level is achieved by applying the multiple engines arrangement (2-DH). Both maximum and minimum airborne noise level of the diesel hybrid is lower than the lowest airborne noise level of the diesel-mechanical plant. The ability of the diesel-hybrid to reduce the brake power of diesel engine by using the electric motor gives a positive impact on its noise level. The electric motor has a low airborne noise level compared to the diesel engine. The improvement for using the diesel-hybrid plant is 4 – 7 dBA compared to the highest sound pressure level. The single-engine configuration (1-DM) of diesel mechanical gives the highest airborne noise.

The battery plant also has two configurations in this analysis. The lowest sound pressure level is obtained by using multiple motors configuration (2-bat). It is obvious from the above figure that the battery gives the lowest airborne noise compared to the other transmission types. The improvement is 17 – 20 dBA lower than the highest sound pressure level in this analysis. This result is not surprising since in battery plant only electric motor is operating. Moreover, the electric motor is the equipment with one of the lowest airborne noise sources.

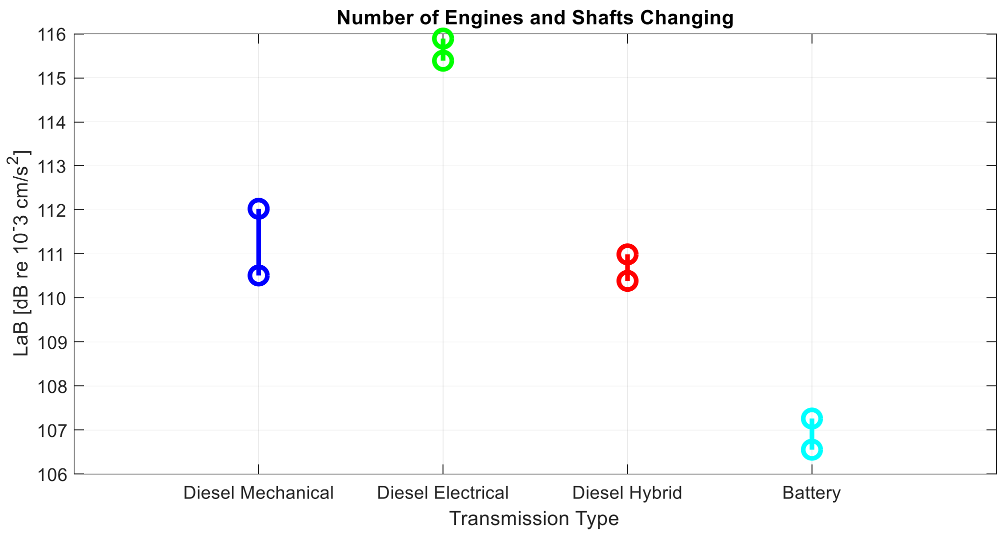


Figure 4.22 Comparison between different transmission types Structure-borne Noise results

The figure above shows the results of the acceleration level of the various possible configurations in a transmission type. The loading point is equal to 100%. The diesel mechanical lowest structure-borne noise level is produced by the single-engine configuration (1-DM), and the highest one is generated by the four engines configuration (4-DM). The results are the total opposite of the airborne noise level results. It happens because in the structure-borne noise level the nominal speed of the engine plays a bigger role compared to the engine room dimensions. The four engines configuration (4-DM) uses smaller engine compared to the single-engine configuration (1-DM) which results in the higher nominal speed of the diesel engine.

The diesel-electrical plant results show high structure-borne noise levels. There is a big gap between the diesel-mechanical and diesel-electrical plant maximum value, almost 5 dB. This difference is due to the overestimate of the generator structure-borne noise source level. It is quite hard to derive the accurate empirical formula for the structure-borne noise level of the generator. The reason is that the contribution of its prime mover contaminates the measurement of generator structure-borne noise level [24]. The highest acceleration level is generated by the five engines configuration (5-DE), and the lowest one is produced by the six engines configuration (6-DE). It happens because the five-diesel engine configuration requires one diesel generator set to be located at the centre-line

The diesel-hybrid plant highest acceleration level is generated by the multiple engines arrangement (2-DH). It is opposite too to the airborne noise results. The lowest structure-borne noise level of the diesel-hybrid plant is almost the same as the lowest structure-borne of the diesel-mechanical. This result might not be true because the electric motor source level is overestimated. The diesel-hybrid is expected to have much lower structure-borne noise level than these results in case of more accurate electric noise level prediction.

The battery plant also produces the lowest structure-borne noise level among others arrangement. The reduction is not as significant as the airborne noise results. It is understandable because the prediction of electric motor source level is conservative [24]. The highest acceleration level is generated by the single-engine configuration (1-Bat), and the lowest one is produced by the two engines configuration (2-Bat).

4.5.2 Varying Loading Point

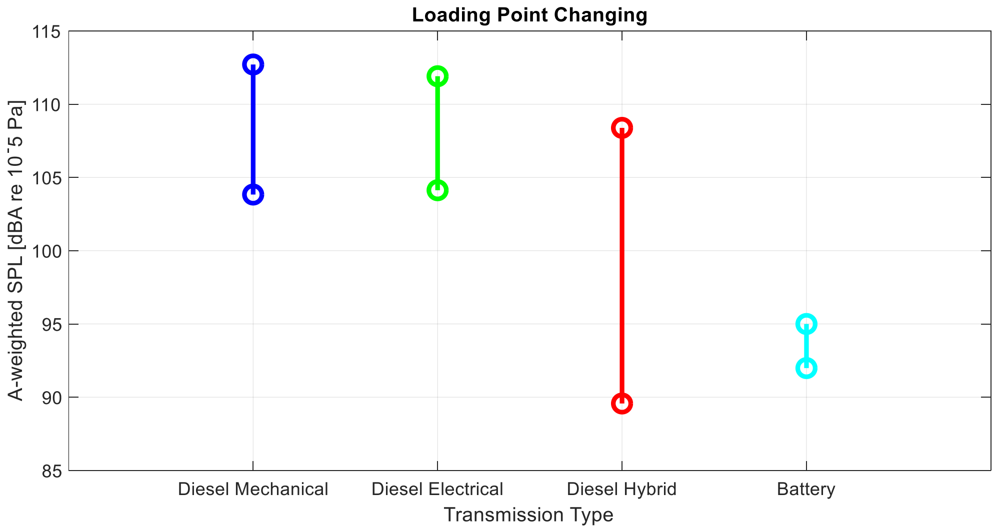


Figure 4.23 Comparison between different transmission types Airborne Noise results

The results presented here are the airborne noise levels over the changing loading point of various propulsion system configurations. The arrangements with the highest number of engines are chosen for every transmission type. For instance, the results of diesel mechanical are the noise levels from four engines configuration (4-DM). The reason one compares the highest number of engines possible is that those arrangements have broader operation points compared to the configuration with less number of engines.

The diesel-mechanical arrangement used in this analysis is the four-diesel engine with two DISO gearboxes (4-DM). The operation point for each loading point can be seen in the appendix D. The interval between the lowest loading point and the highest loading point seems quite broad thanks to the four engines configuration (4-DM). However, one can expect the interval to be shorter when the arrangements with less engine are used. The highest sound pressure level is at 100% gamma. When the gamma is at 10%, the diesel-mechanical plant also gives a high airborne noise level compared to the other transmission types.

The diesel-electrical arrangement used in this analysis is the six-diesel generator sets configuration (6-DE). The interval between the airborne noise level at gamma 10% and 100% is slightly shorter than the diesel-mechanical plant. The diesel-electrical plant sound pressure level at 100% loading point is slightly lower than the diesel-mechanical. The lowest loading point result of the diesel-electric plant is higher than the diesel-mechanical plant. This result is due to the operation mode of the diesel generator set. It needs to work at its nominal speed although the diesel genset works at part load condition. In that sense, even though the diesel generator set used in the diesel-electrical plant has significantly lower brake power, the requirement to work at the nominal speed make the sound pressure level rather high.

The diesel-hybrid plant used in this analysis is the two engines configuration with two electric motors and two-DISO gearboxes (2-DH). It has the broadest range of noise level between 10% and 100% gamma. This broad range is due to the possibility to use only the electric motor in the low load condition. The diesel-hybrid plant noise level at 100% gamma is lower than the diesel-electrical and the diesel-mechanical plant. It is expected since the number of engines and shafts changing result also shows the same condition.

The battery plant used in this analysis is the two motors configuration (2-Bat). There is no need to use the gearbox in this arrangement since the Variable Frequency Driver (VFD) can arrange the speed of the engine. The difference between the sound pressure level of the battery plant at gamma 10% and 100% is quite short compared to the other arrangements. It happens because the electric motor does not have the actual speed or torque as the parameter to calculate the airborne noise source level. Therefore, all the sound pressure levels with the gamma lower than equal to 50% are equal to the sound pressure level of one motor operates at nominal power and speed while the sound pressure levels with the gamma greater than 50% are equal to the sound pressure level of two motors operate at nominal power and speed. The battery plant airborne noise level at the lowest loading point is higher than the diesel-hybrid plant. It is expected because the battery-plant has an electric motor with a larger installed power.

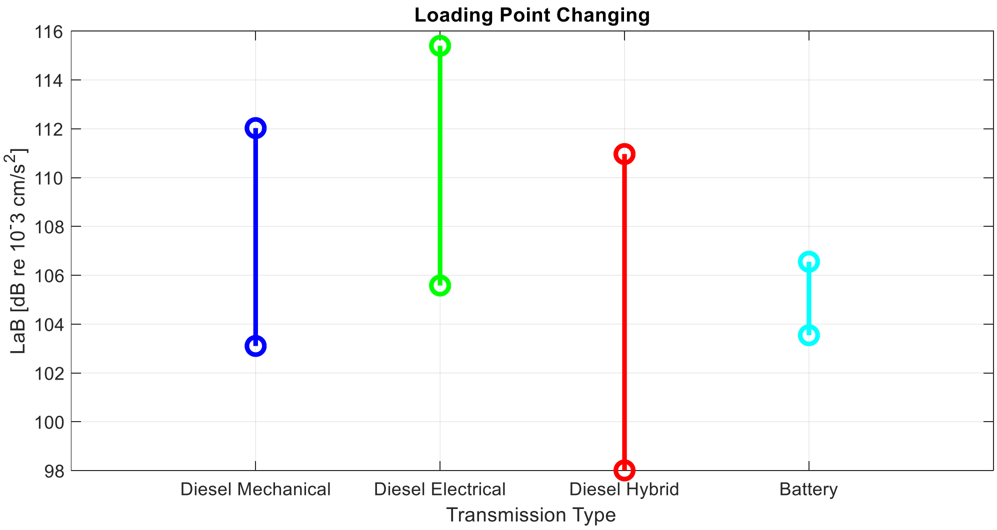


Figure 4.24 Comparison between different transmission types Structure-borne Noise results

The results illustrated in the figure above are the structure-borne noise levels over the changing loading point of various propulsion configurations. The arrangements with the highest number of engines are chosen for every transmission type. The diesel-mechanical structure-borne results have the 10 dB difference between the noise level at the maximum and the minimum loading point. This value is almost the same as the airborne noise level difference between the noise level at the highest and the lowest gamma.

The diesel-electrical plant used in this analysis is the six diesel generator sets configuration (6-DE). The structure-borne noise levels difference between the maximum and the minimum loading point is almost the same as the airborne noise level range. However, due to the overestimate of the generator structure-borne noise source level, the overall level of the diesel-electrical plant is higher than other arrangements. One can expect the diesel-electrical plant has lower structure-borne noise level than these results in case of more accurate generator noise model.

The diesel-hybrid plant used in this analysis is the two diesel engines configuration (2-DH). The structure-borne noise level difference between the maximum and minimum loading point is 12 dB. This value is significantly small compared to the range of the airborne noise level results. The interval in airborne noise results is equal to 19 dBA. It is expected since the generator formula is used to calculate the electric motor structure-borne noise source level. As explained earlier, the generator structure-borne noise level tends to be on the high side. One can argue that the smaller interval in the diesel-hybrid plant is because of the overestimation of the electric motor acceleration level. Nevertheless, the lowest noise level at the lowest loading point is still obtained by using the diesel-hybrid plant.

The battery plant used in this analysis is the two electric motors configuration (2-Bat). The structure-borne noise level at the lowest loading point is higher than the diesel-mechanical vibration level at the lowest gamma. Again, this result is due to the overestimate of the electric motor structure-borne source level. Nevertheless, at the highest loading point, the battery plant generates the lowest structure-borne noise level among other arrangements. The diesel engine generates rather a high vibration level when operating at the nominal power and speed compared to the electric motor.

4.5.3 Varying the Engine Mounting

All analyses that have been done for the structure-borne noise level are assumed to be hard mounted. The results below show the structure-borne noise level when the machinery is resiliently mounted. The characteristics of the comparison are still the same with the hard-mounted equipment. The battery has the lowest vibration level compare to the other transmission types for the number of engines and shafts changing. In the loading point changing results, the lowest noise at the lowest gamma is achieved by the diesel hybrid.

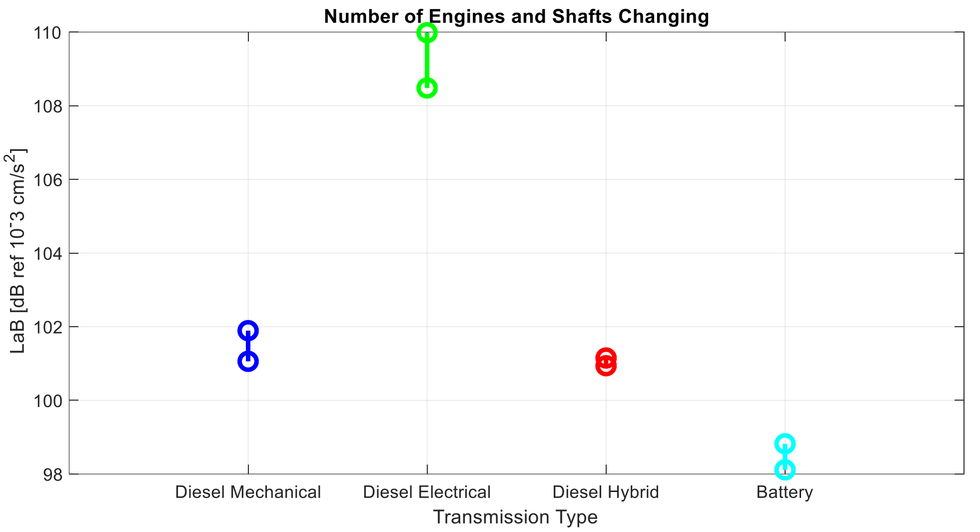


Figure 4.25 Comparison between different transmission types Structure-borne Noise results

5

Sensitivity Analysis

5.1 Overview

This chapter will describe all the essential results of the sensitivity analysis. Various models are being implemented in this project. The sizing model results are compared to the dimensions of the actual engines from the database. The predicted dimensions of various engines then will be used as an input to the noise model. Afterwards, the equivalent noise level is calculated using the predicted and the actual dimensions. This needs to be done in order to understand better the sensitivity of the equipment sizing model towards the noise model. These comparisons are going to be analysed and explained in detail for every equipment.

The sensitivity of the model to the changing installed power will also be analysed. From the previous chapter, the model is implemented only to a ship with an installed power of 5000 kW. It is really important to do this sensitivity test because the behaviour of the parameters such as the number of engines and shafts, the number of rooms and the loading point changing are still unknown for different installed power. In that sense, every chosen installed power needs to be simulated with the evaluation methodology, both for the airborne and the structure-borne noise. Two methods are used to vary the installed power in this analysis, i.e., (1) Changing the number of cylinders and (2) Changing the power per cylinder. It is required to separate these parameters because combining them in one simulation will give conflicting results. In practice, these options are the available choices that the designer can have.

The last analysis is to examine the behaviour of the model over the varying split between the propulsion and the auxiliary power for a certain installed power. In the case study, it is assumed that the split between the propeller and the auxiliary load is 80% and 20% of the installed power respectively. However, in reality, this is not always the case. It depends on the functionality of the vessel. For instance, AHTS vessel commonly has a split of 50% and 50% between the propeller and the auxiliary load while a tanker has a split of 80% and 20% between the propeller and the auxiliary load. It is expected that there will be no effect on the noise level of the diesel electrical plant since in the diesel electrical plant both propulsion and auxiliary load are generated by the diesel generator sets. Nevertheless, the nominal power of the electric motor is changing over the varying split between the propulsion and the auxiliary power.

5.2 Sizing Prediction

5.2.1 Diesel Engine

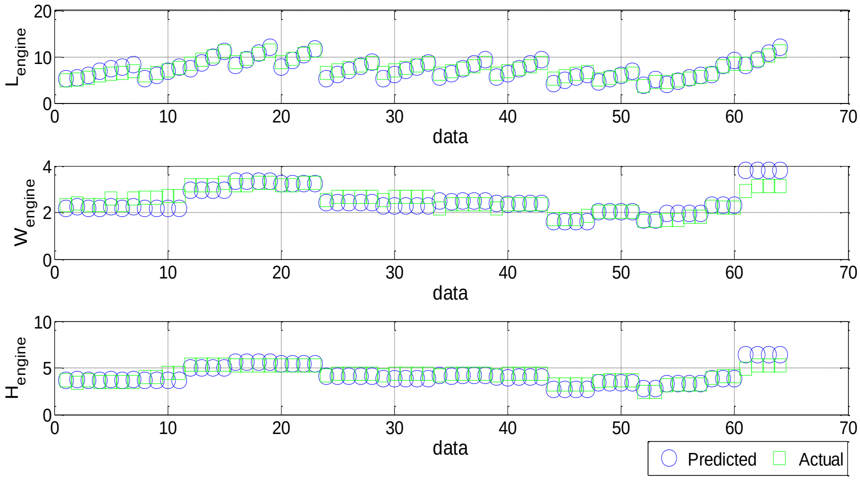


Figure 5.1 Diesel Engine Sizing Results and Comparisons

The results shown above are the four strokes medium speed engines with L arrangement. The y-axes show the dimensions of the diesel engines and the x-axes represent the sequence number of a certain engine in the database (see Appendix A.1). The comparisons between actual and predicted dimensions for other diesel engine types can be found in the appendix B.2 and B.4.

The simulation results show a very good comparison between the actual and the predicted dimensions. The absolute average difference between the actual and predicted is 0.52, 0.21 and 0.35 for the length, width and height respectively. It is expected because the regression analyses show high R squared values. The actual engines data are taken from the Wartsila and MAN catalogue [42] [43]. It is possible to use the coefficients found in this model to predict diesel engine dimensions from different manufacturers,. However, it is better to find the new regression coefficients. This recommendation is mainly because other manufacturers may have different types of construction, which leads to different “disturbance” factors.

The effect of the dimensions difference is then investigated to the sound pressure level calculation. The diesel engine is assumed to be at the centre of an engine room, and there is no other equipment around it. All the boundary surfaces are steel. The room dimensions calculated based on the engine dimensions and the minimum distance from the equipment to the wall. Typical values of minimum distances between the engine and the boundary surfaces are used [42]. The figure below shows the result of SPL calculation from 64 series of engines.

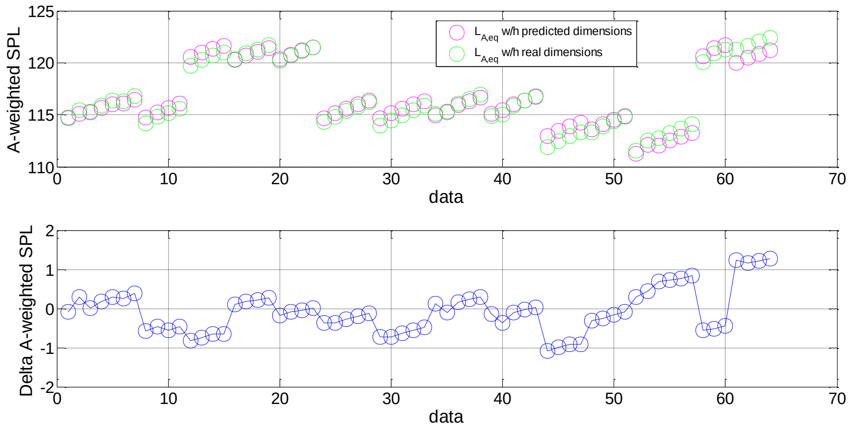


Figure 5.2 Diesel Engine SPL Calculation Results and Comparisons

The lower figure shows the difference between the sound prediction with real dimensions and the sound prediction with predicted dimensions. The maximum delta found from data number 61 to number 64. The specifications of these diesel engines can be found in Appendix A.1. This trend also can be found for the results of width and length of the engine. One could argue that this happens because of the engine speed prediction. The inaccuracy in rpm leads to the deviation in the stroke length of the engine which leads to the overall length and the overall width. Overall, the predicted sound calculations give satisfying results with an absolute average difference of 0.44 dBA.

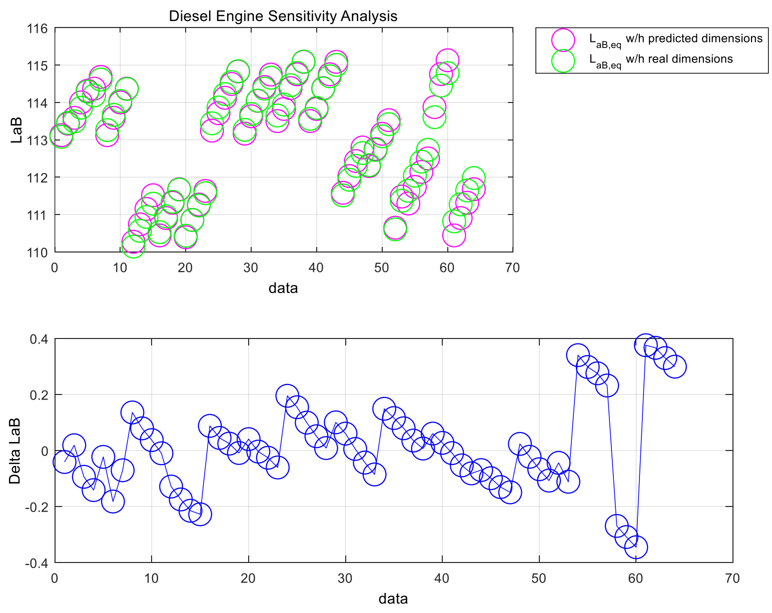


Figure 5.3 Diesel Engine LaB Calculation Results and Comparisons

The structure-borne noise levels with the predicted dimensions also give very good fit with the structure-borne noise levels with the real dimensions.. It is even better than the airborne noise level results. The average difference is 0.01. The maximum delta is also obtained from engine number 61 to 64. It happened because of the same reason as the airborne noise deviations.

5.2.2 Gearbox

The gearbox models are compared with the actual dimensions of Wartsila and MAN gearboxes. Only single stage gearbox is being compared due to the scarcity of double stage gearbox data. Moreover, this thesis also only consider the single stage speed reduction gears. The figures below show the results of SISO and DISO gearboxes. One can see that the predicted dimensions are almost the same as the actual one. Besides the high R squared value for the regression analysis, it is also because there are almost no disturbance factors in gearbox construction type. The construction of gearbox from different manufacturers is similar to each other. This condition is a total opposite of the diesel engines that have many disturbances such as turbocharger, water pump and so on.

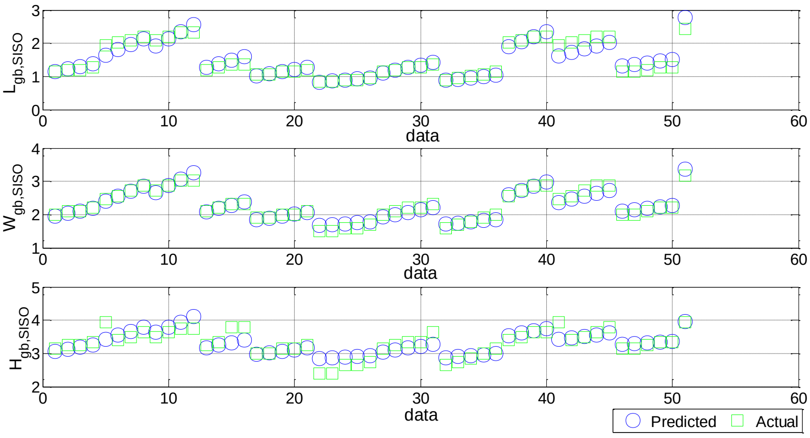


Figure 5.4 SISO Gearbox Sizing Results and Comparisons

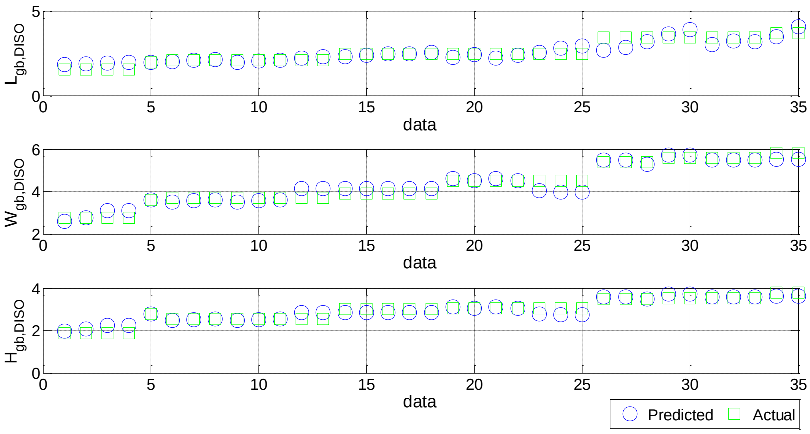


Figure 5.5 DISO Gearbox Sizing Results and Comparisons

The gearbox SPL calculation performed by assuming the distance between the equipment to the boundary surfaces as the same as the diesel engine. There is no typical value from the manufacturer regarding these distances. This lack of a standard is not going to be a problem, as long as the actual gearbox and the predicted one has the same minimum distances then the comparisons are still valid. The graphs below illustrate the comparisons between noise calculations with the predicted and the actual dimensions.

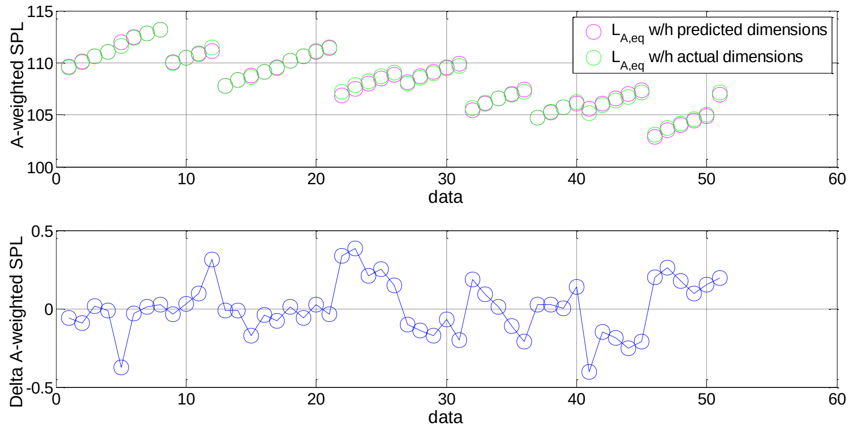


Figure 5.6 SISO Gearbox SPL Calculation Results and Comparisons

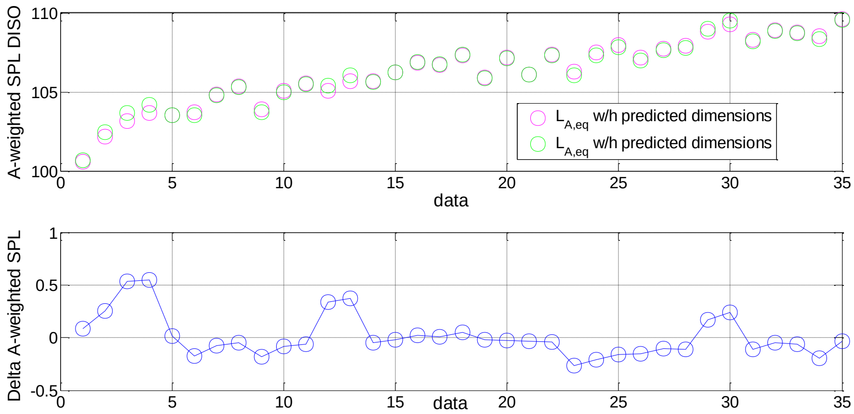


Figure 5.7 DISO Gearbox SPL Calculation Results and Comparisons

For both types of reduction gears, the maximum difference is 0.5 dBA. The effect of the predicted dimensions of gearbox also gives only a small amount of deviation in the structure-borne noise. It can be said that the deviations from the predicted dimensions are really small and good enough for the noise calculation. Therefore, the sizing model of the gearboxes is reliable to be used for the SPL and LaB calculation of a certain propulsion system arrangement.

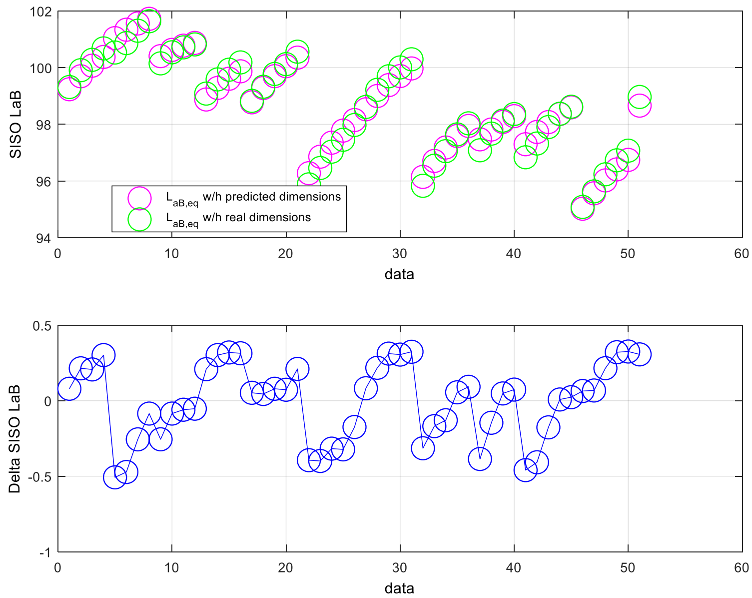


Figure 5.8 SISO Gearbox LaB Calculation Results and Comparisons

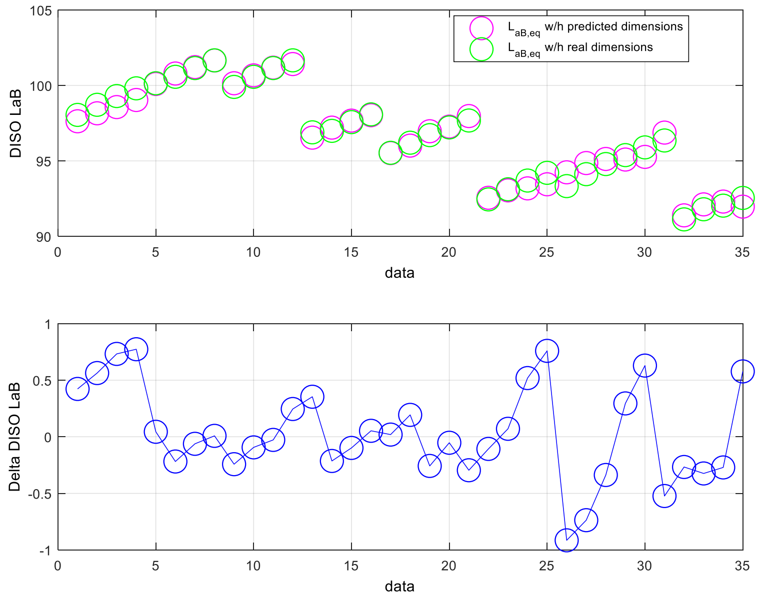


Figure 5.9 DISO Gearbox LaB Calculation Results and Comparisons

5.2.3 Electric Motor

The motors being validated here are the low voltage induction motors manufactured by ABB. There are other types of it but the one presented here is the HXR one. Other types of electric motors will have different constructions; some will have a similar structure. In that sense, it is better to check the overall shape of the machine before deciding if it is better to use the regression coefficients found here or create a new one with a new database. For instance, small electric machines will have an open fan and cooling vanes for the cooling purpose, but a larger one requires a close heat exchanger to cool the machine [32].

The results from the model show a high accuracy. In this analysis, the ratio between the rotor and stator diameter is assumed to be 0.5. However, when predicting a new machine, the s value can be chosen between 0.45 – 0.55.

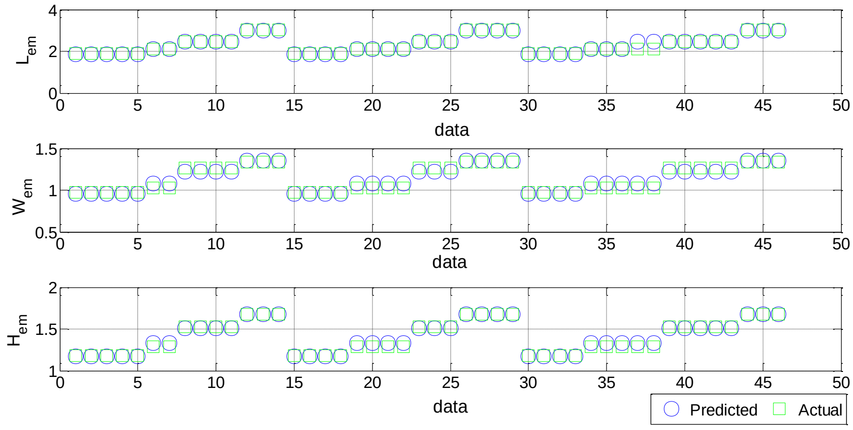


Figure 5.10 Electric Motor Sizing Results and Comparisons

Similar to the other equipment, the noise calculations of the electric motors are performed to see the effect of predicted dimensions to the noise level. Since the fit between the predicted and the actual one is very good, the result of the sound calculations also have the same characteristics. The average difference between those levels is 0.04. The minimum distances between the machine to the wall are assumed to be the same as the one used in diesel engine because such information could not be found in the catalogue.

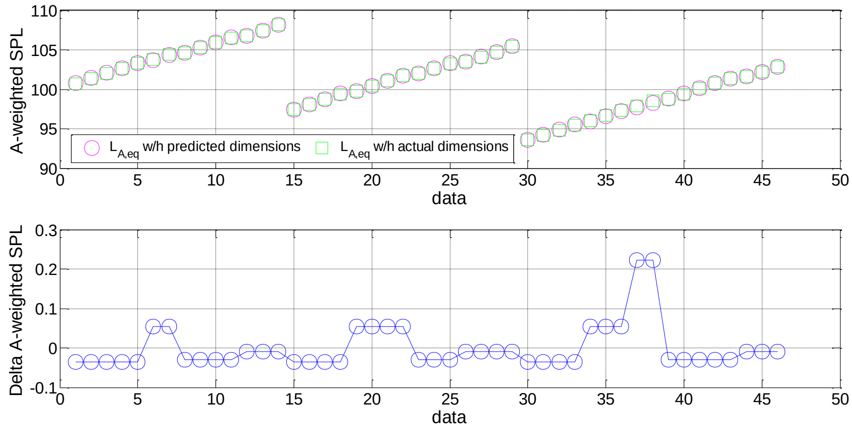


Figure 5.11 Electric Motor SPL Calculation Results and Comparisons

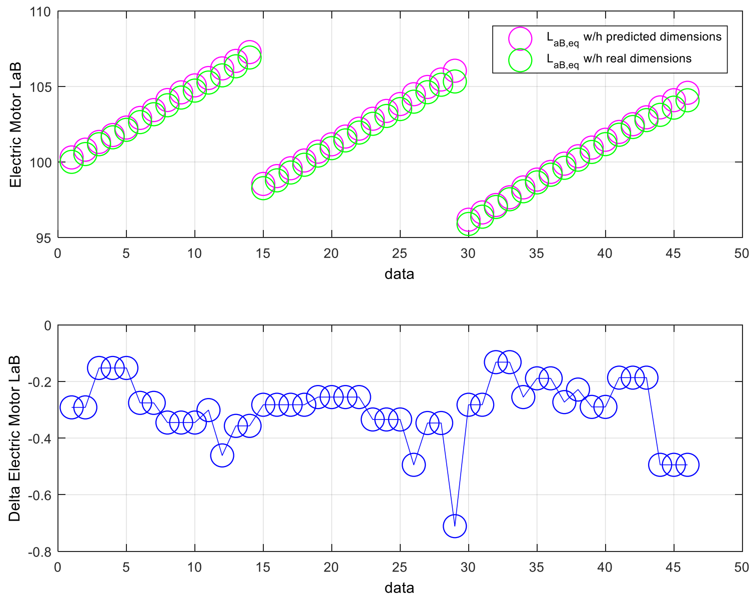


Figure 5.12 Electric Motor LaB Calculation Results and Comparisons

5.2.4 Diesel Generator Set

The results presented here are for the diesel generator with 4-strokes medium speed L-arrangement prime mover. These results show the least fit to the actual dimensions with absolute average differences of 0.79, 0.30 and 0.36 for the length, width and height respectively. The deviations are expected because the model is not taking into account the base frame of the equipment. It is the main disturbance factor to the overall dimensions as explained in the subchapter 3.5.4.

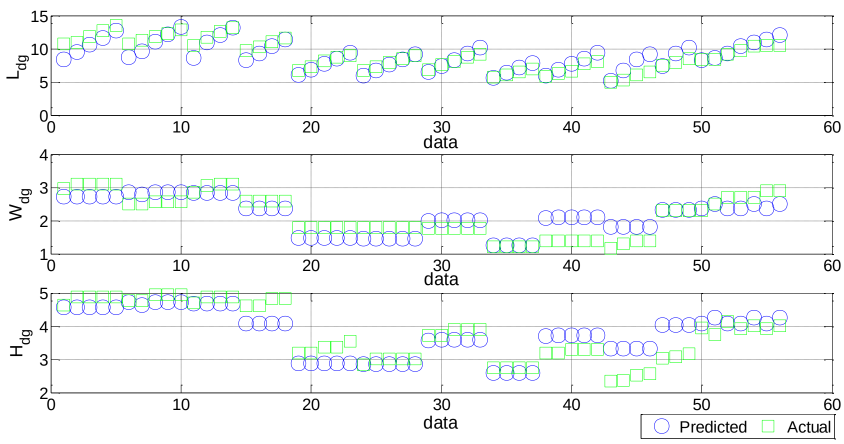


Figure 5.13 Diesel Genset Sizing Results and Comparisons

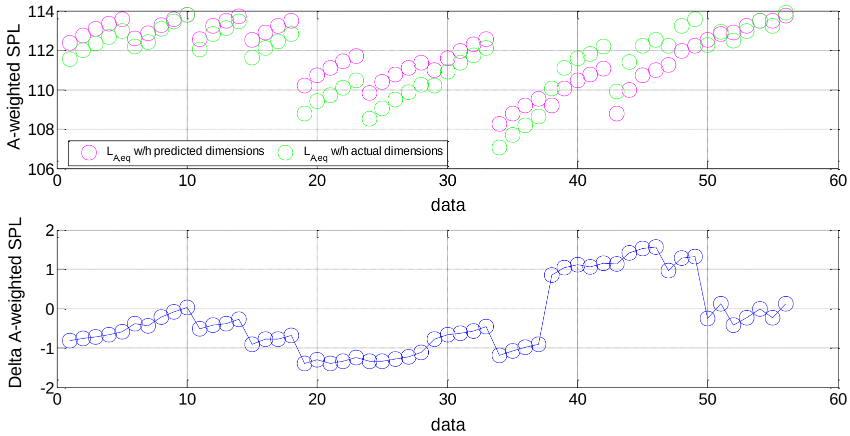


Figure 5.14 Diesel Genset SPL Calculation Results and Comparisons

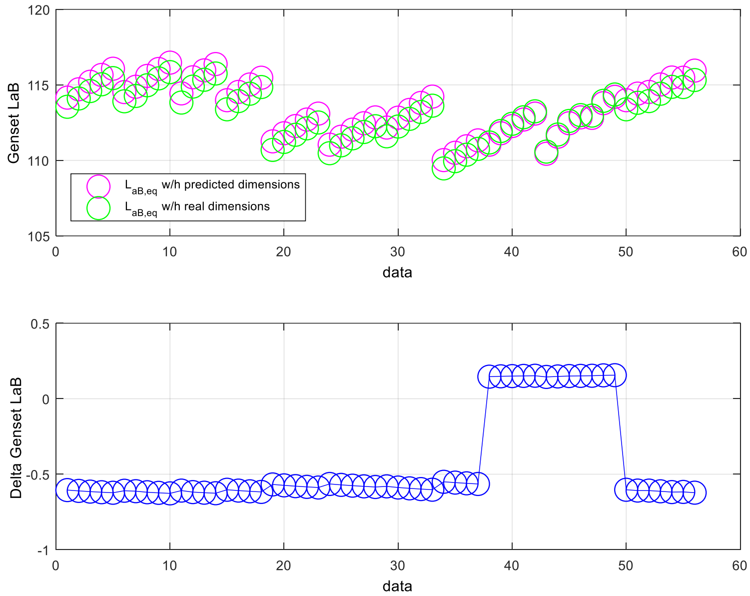


Figure 5.15 Diesel Genset LaB Calculation Results and Comparisons

The relatively high absolute average differences reflected in the SPL calculation. It is evident the difference between the actual and the predicted one. The maximum delta is about 1.5 dBA. For the airborne noise, the diesel gensets have the highest deviation compared to other equipment. However, one could argue that difference is still relevant because the difference is still in the same range with the other equipment. The structure-borne noise with predicted dimensions results gives better fits to the LaB with real dimensions results. The maximum difference is only 0.6 dB. This result means that the diesel genset structure-borne noise is less sensitive to the dimensions of it compared to the airborne noise.

5.3 Varying Installed Power by Changing the Number of Cylinders

This sensitivity analysis will show how the model behaves over the varying installed power. The installed power is one of the ship requirements. One way to increase the power of the engine is to increase the number of cylinders (z) and keep the power per cylinder constant. Two types of cylinder arrangements can be used in this analysis, the L and V arrangement. The maximum number of cylinders for the diesel engine is 20 cylinders, and the minimum number of cylinders is 6. The range of installed power is varying from 3750 kW to 12500 kW. The number of cylinders limits the installed power range. Theoretically, it is possible to have an engine with more than 20 cylinders, but the applicability of it needs a further investigation which is out of the scope of this project.

This analysis will be done on the diesel-mechanical, the diesel-electrical and the diesel-hybrid plant. From the chapter 71,the results of a case study are presented for these transmission types. However, it only gives the results for one type of ship requirements. It is important to investigate if the noticed trends from the case study results are valid for different ship requirements.

The evaluation methodology is implemented together with varying the installed power. All the results are presented in the normalised values. The higher number of engines will be normalised to the lowest one for the number of engines and shafts changing simulation. A certain loading point will be normalised to the maximum loading point for the loading point changing simulation. A positive normalised value means that there is an improvement of acoustical performance while a negative value means that the noise is increasing instead of decreasing.

5.3.1 Diesel Mechanical

Engine Requirements			
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**
3750 kW	**DM-1**	500	6
	DM-2	250	6
	DM-3	250	6
	DM-4	125	6
5000 kW	**DM-1**	500	8
	DM-2	250	8
	DM-3	250	8
	DM-4	125	8
6250 kW	**DM-1**	500	10
	DM-2	250	10
	DM-3	250	10
	DM-4	125	10
8750 kW	**DM-1**	500	14
	DM-2	250	14
	DM-3	250	14
	DM-4	125	14
12500 kW	**DM-1**	500	20
	DM-2	250	20
	DM-3	250	20
	DM-4	125	20

Table 5.1 Input data for Diesel Mechanical analysis

Number of engines and shafts changing

Four diesel-mechanical arrangements are going to be investigated, i.e., one engine with one SISO gearbox (DM-1), two engines with one DISO gearbox (DM-2), two engines with two SISO gearboxes (DM-3) and four engines with two DISO gearboxes (DM-4). From the input of the simulation, it is evident that increasing the installed power will be achieved by changing the number of cylinders and keep the power per cylinder constant. The number of cylinders interval for the L-arrangement is one and for the V-arrangement is two. It is not possible to have an odd number of cylinders for the V-engine since the cylinders need to be in a pair. All the propulsion system configurations have the same number of cylinders for an installed power. For instance, when the installed power is equal to 3750 kW, all the diesel-mechanical arrangements (DM-1, DM-2, DM-3 and DM-4) use the diesel engine with six cylinders. The results are going to be represented in the normalised values, which can be expressed as follows,

$$L^*_{p,eq} = \frac{L_{p,eq@1SISO1DE} - L_{p,eq@multiple\ engines\ arrangement}}{L_{p,eq@1SISO1DE}} \tag{5.1}$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@1SISO1DE} - L_{aB,eq@multiple\ engines\ arrangement}}{L_{aB,eq@1SISO1DE}} \tag{5.2}$$

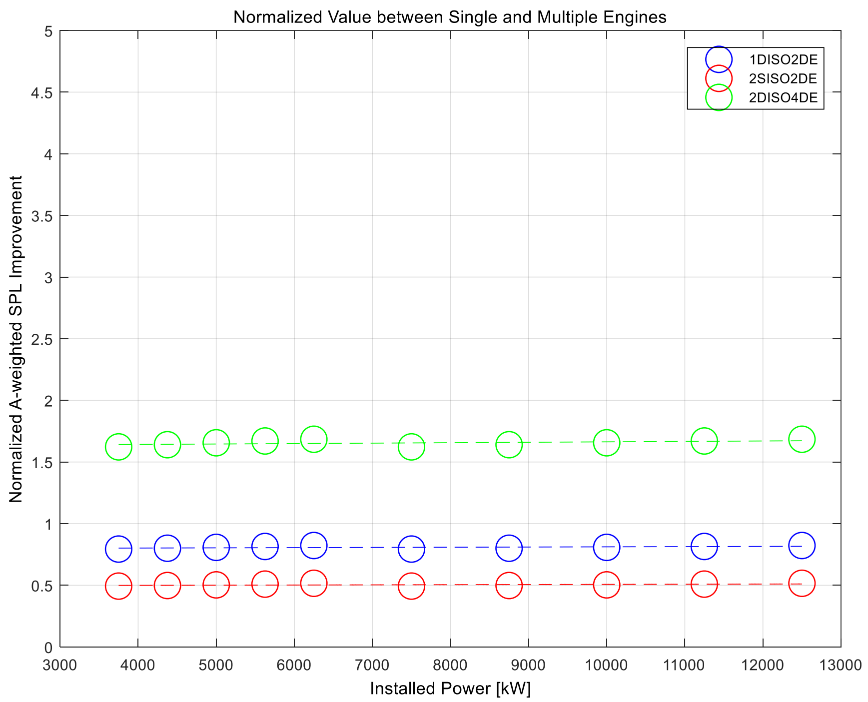


Figure 5.16 Number of engines and shafts changing over varying installed power by changing the number of cylinders Airborne Noise Results

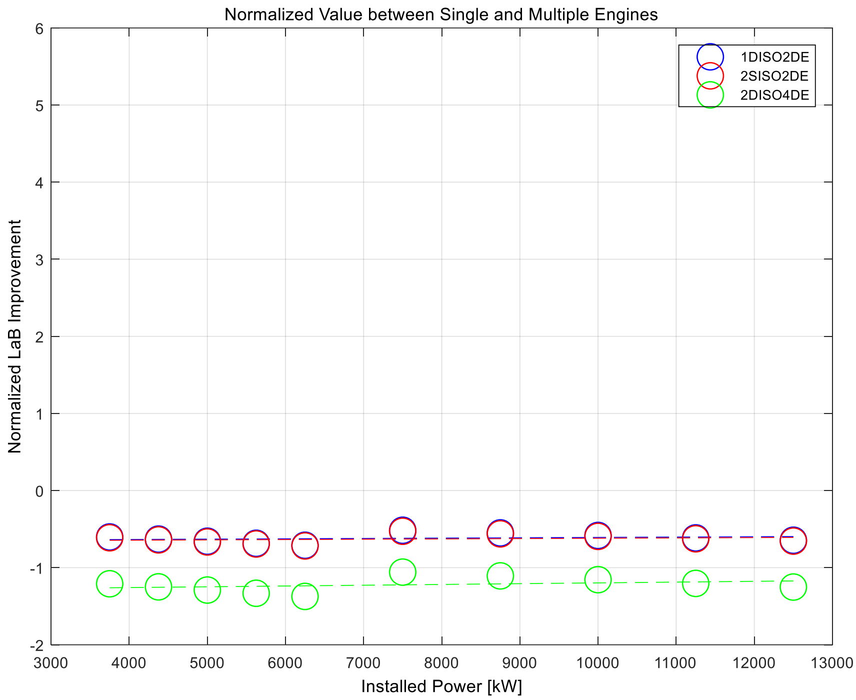


Figure 5.17 Number of engines and shafts changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

The figures above illustrate the change of noise improvement towards the increasing installed power. These results show the propulsion noise level at rated speed and power. It can be seen from these figures that the improvements of airborne and structure-borne noise are almost constant over the varying power demands. The influence of changing the number of cylinders is not exactly zero but it is too small to be considered significant. The number of cylinders is used to determine two variables, which are the power per cylinder and the engine room dimensions.

There are two main reasons why the trend is almost a straight line. Firstly, the power per cylinder is constant over the varying installed power because one increases the nominal power of the engine by increasing the number of cylinders. The nominal speed of the engine stays constant if the power per cylinder remains constant. It can be seen from the results in subchapter 4.2.1 that the nominal engine speed plays a big role in both the airborne and structure-borne noise level. Secondly, increasing the engine brake power by changing the number of cylinders will give an increase in the diesel-engine dimensions. This increase will result in a larger engine room and a higher noise absorption by boundary surfaces. However, the differences in engine room dimensions between different arrangements are almost constant. This is because the increase in the number of cylinders is proportional to every arrangement in each installed power. For instance, at 3750 kW, all the diesel-mechanical arrangements (DM-1, DM-2, DM-3, DM-4) use diesel engine with six cylinders. If the number of cylinders is arbitrary for every arrangement over the varying installed power, one could expect a random improvement trend too.

There is a small decrease between 6250 kW and 7500 kW installed power. The decrease is due to different types of cylinder arrangements. The diesel engines use for all configurations after the installed power of 7500 kW are V-engines. The airborne noise reduction by changing the number of engines is smaller when one uses the V-engine. This is because the V-engine dimensions are more compact compared to the L-Engine. In that sense, the engine room is smaller which leads to lower airborne noise absorption by the boundary surfaces. The structure-borne noise results also show that the improvements of the structure-borne noise levels over the varying power demand are almost constant. A small difference can be found between the installed power before and after 7500 kW.

Loading point changing

The results below show the normalised noise level at two loading points of a particular diesel-mechanical configuration. It is normalised to the noise level at the highest loading point (gamma = 100%). The normalised value can be written as follows,

$$L^*_{p,eq} = \frac{L_{p,eq@\gamma=100\%} - L_{p,eq@\gamma=\cdots\%}}{L_{p,eq@\gamma=100\%}} \tag{5.3}$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@\gamma=100\%} - L_{aB,eq@\gamma=\cdots\%}}{L_{aB,eq@\gamma=100\%}} \tag{5.4}$$

The airborne noise and the structure-borne noise results show that the improvements are slightly declining over the increasing installed power when the gamma is equal to 10%. The slight decreasing trend is because the noise level at 100% loading point is increasing towards the installed power while the difference between the noise level at 100% and 10% stays constant. The difference stays constant due to the same load line characteristic use for the various engines. Therefore, the actual speed at a certain loading point is the same for different engine brake power. For instance, the actual speed of operating diesel engine in the four engines configuration (4-DM) at 10% gamma and 3750 kW installed power will have the same actual speed as the operating diesel engine in four engines configuration at 10% gamma and 12500 kW installed power. At higher loading points, the difference between the noise level in the actual and the nominal load becomes really small thus makes it insignificant to the noise level at 100% loading point. This condition will lead to smaller improvement against the varying installed power. It can be seen from the results of 40% loading point.

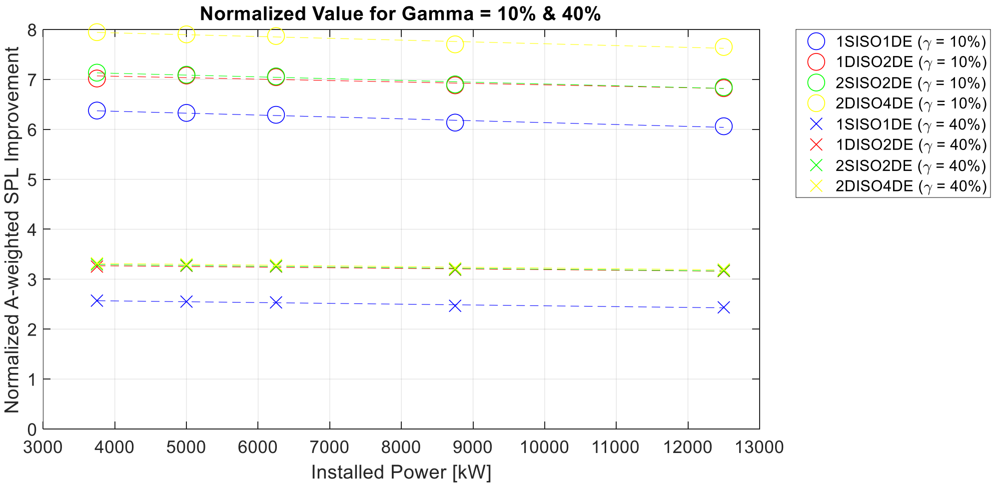


Figure 5.18 Loading point changing over varying installed power by changing the number of cylinders Airborne Noise Results

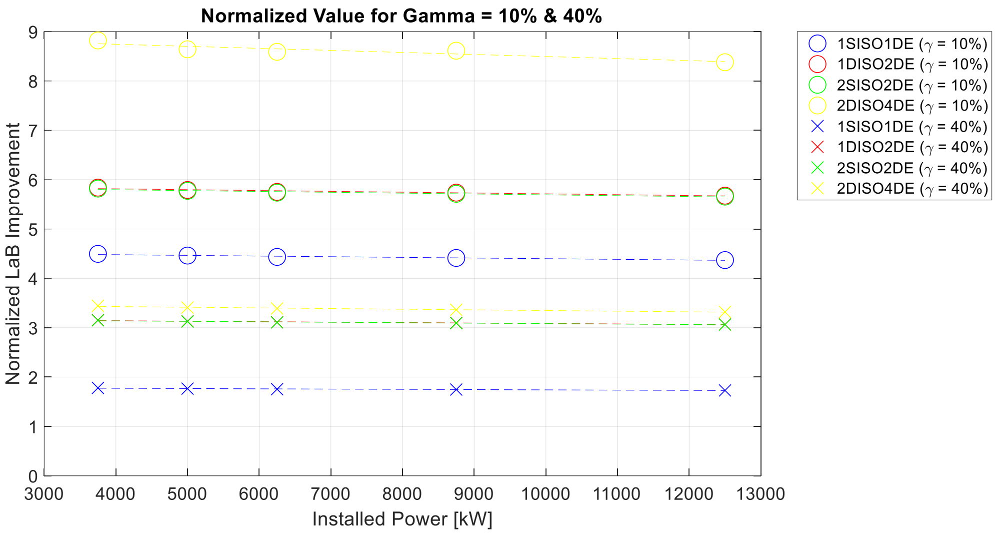


Figure 5.19 Loading point changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

5.3.2 Diesel-Electrical

Engine Requirements			
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**
3750 kW	**DE-1**	535	7
	DE-2	270	7
	DE-3	130	7
	DE-4	100	6
5000 kW	**DE-1**	535	9
	DE-2	270	9
	DE-3	130	9
	DE-4	100	8
6250 kW	**DE-1**	535	10
	DE-2	270	10
	DE-3	130	10
	DE-4	100	9
8750 kW	**DE-1**	535	18
	DE-2	270	18
	DE-3	130	18
	DE-4	100	16
12500 kW	**DE-1**	535	20
	DE-2	270	20
	DE-3	130	20
	DE-4	100	18

Table 5.2 Input data for Diesel-Electrical analysis

Number of engines and shafts changing

The value of the power per cylinder is not a fixed number for the diesel-electrical plant. It will vary more or less 3% from the values in the table. Although it is changing instead of constant, the variation is insignificant and give almost no effect on the results. Four arrangements of the diesel-electrical plant are going to be investigated, i.e., a plant with one diesel generator set, two diesel generator sets, four diesel generator sets and six diesel generator sets. The results will be presented in the normalised values. The multiple-engines arrangements are normalised to the single-engine arrangement, which can be written as follows,

$$L^{*}_{p,eq} = \frac{L_{p,eq@1DG} - L_{p,eq@multiple\ engines\ arrangement}}{L_{p,eq@1DG}} \tag{5.5}$$

$$L^{*}_{aB,eq} = \frac{L_{aB,eq@1DG} - L_{aB,eq@multiple\ engines\ arrangement}}{L_{aB,eq@1DG}} \tag{5.6}$$

The airborne noise results show a declining trend towards the installed power. However, the decrease in the improvement is not very big. The discrepancy before and after 7250 kW is because the change in cylinder arrangement (similar to the diesel mechanical plant). A high improvement can be seen for the six diesel generators arrangement at 6250 kW installed power. The main reason is that at 6250 kW installed power the six engines use the diesel generator sets with L-engine arrangement while the single-engine configuration used the V-engine arrangement. As explained earlier, the V-engine is a more compact engine compared to L-engine which will result in smaller room constant. Therefore, the V-engine produce higher noise compares to the L-engine.

The structure-borne noise normalised values show a little decline over the varying installed power which makes the results are almost constant. A discrepancy can be found again for the six engines arrangement. However, in the structure-borne noise, the V-engine will have lower vibration level because it has a wider dimension. Therefore, the six engines, which use the L-engine at 6250 kW, has lower improvement compare to the others.

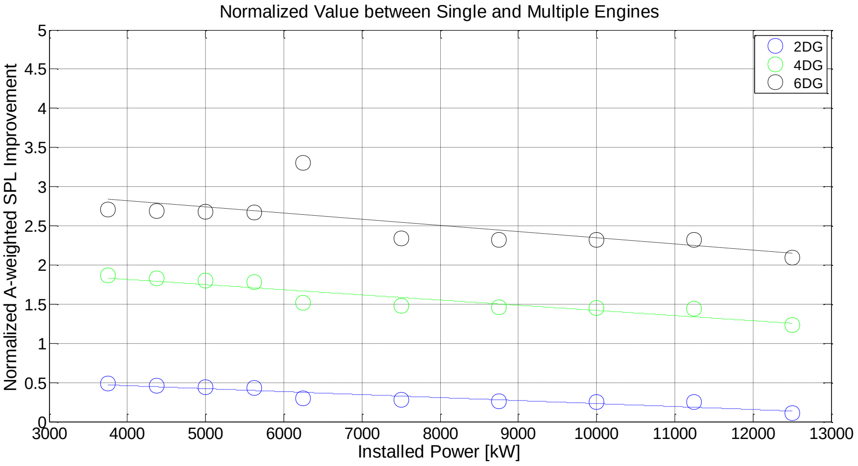


Figure 5.20 Number of engines and shafts changing over varying installed power by changing the number of cylinders Airborne Noise Results

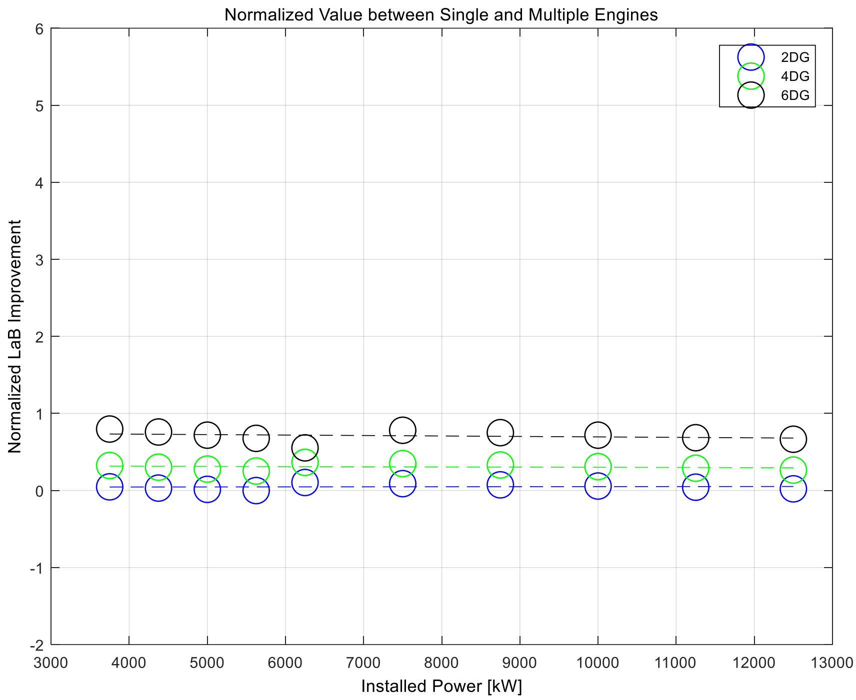


Figure 5.21 Number of engines and shafts changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

Loading point changing

This analysis will be done for four configurations of the diesel-electrical plant. Every arrangement has a different power per cylinder. The power per cylinder will stay constant over the varying installed power, but the number of cylinders will vary. The results below are the normalised values of two loading points (10% and 40%). The equations to get the normalised values are similar to the equation 5.3 and 5.4.

As can be seen from the graphs, both the airborne and the structure-borne noise obtain the highest improvement by applying the maximum number of diesel generator sets. The improvements are almost constant, with a small decreasing trend, over the varying installed power. These results are expected since the power per cylinder is constant for all engines. The next sub-chapter will explain the results of power per cylinder changing.

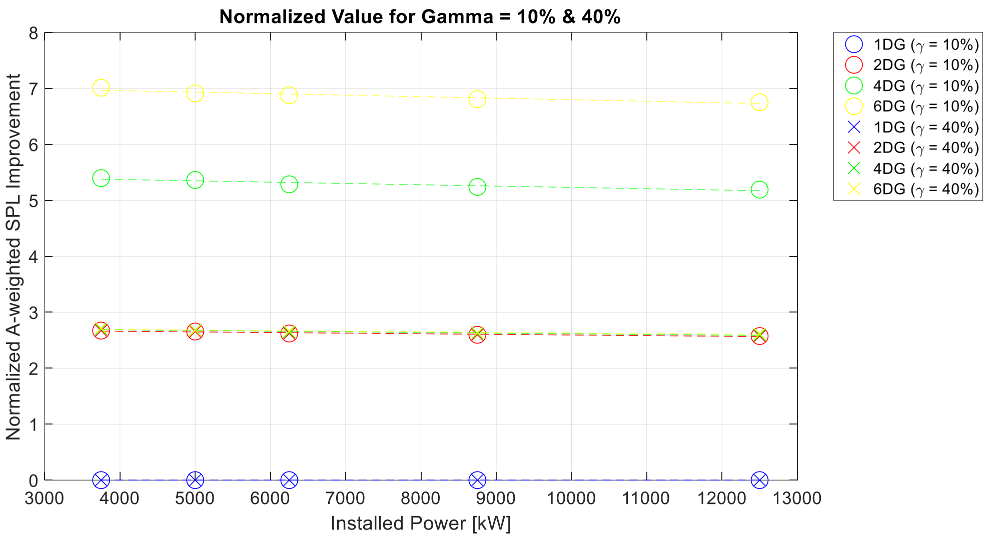


Figure 5.22 Loading Point changing over varying installed power by changing the number of cylinders Airborne Noise Results

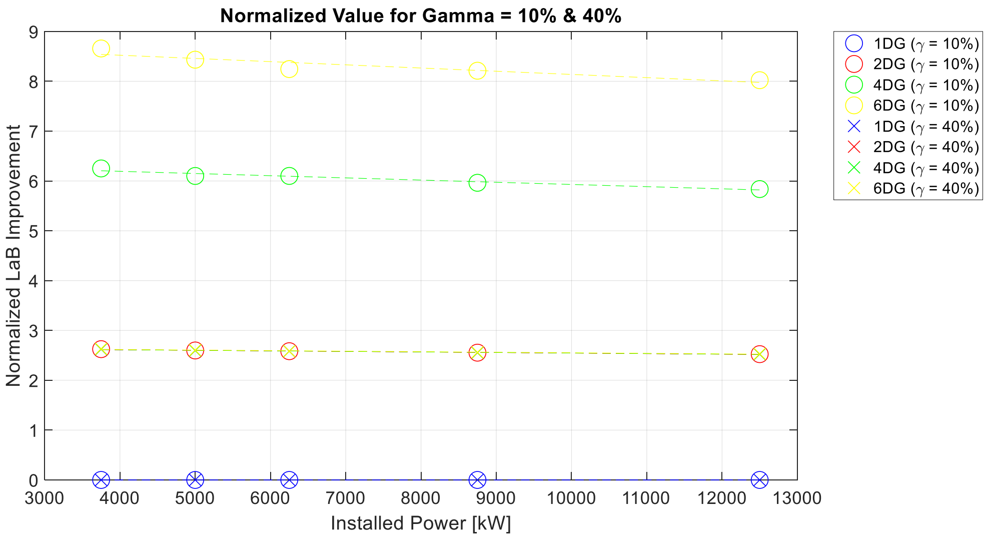


Figure 5.23 Loading Point changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

5.3.3 Diesel-Hybrid

Engine Requirements				
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**	**Delta**
3750 kW	**DH-1**	535	7	0.5
	DH-2	270	7	0.5
5000 kW	**DH-1**	535	9	0.5
	DH-2	270	9	0.5
6250 kW	**DH-1**	535	12	0.5
	DH-2	270	12	0.5
8750 kW	**DH-1**	535	16	0.5
	DH-2	270	16	0.5
12500 kW	**DH-1**	535	20	0.5
	DH-2	270	20	0.5

Table 5.3 Input data for Diesel-Hybrid analysis

Number of engines and shafts changing

Two diesel-hybrid plant arrangements are going to be analysed in this simulation, i.e., a single diesel engine with one electric motor and a DISO gearbox, and two diesel engines with two electric motors and two DISO gearboxes. The electric motor nominal power sets to be minimum which means the electric motor power is enough to boost the diesel engine power to deliver the maximum propulsion demand. The results of this sensitivity analysis will be presented in the normalised values.

$$L^*_{p,eq} = \frac{L_{p,eq@1DISO1DE} - L_{p,eq@multiple\ engines\ arrangement}}{L_{p,eq@1DISO1DE}} \qquad (5.7)$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@1DISO1DE} - L_{aB,eq@multiple\ engines\ arrangement}}{L_{aB,eq@1DISO1DE}} \qquad (5.8)$$

The delta value for the diesel-hybrid in this analysis is 0.5 since one is interested to see how the varying installed power affects the noise level. Unlike the diesel-electric and the diesel-mechanical plant, the diesel hybrid already uses a V-engine at 6250 kW. The discrepancy at that point is due to the usage of V-engine for both single and multiple engines arrangements. For the structure-borne noise results, the V-engine will give a better improvement while it is the other way around in the airborne noise results.

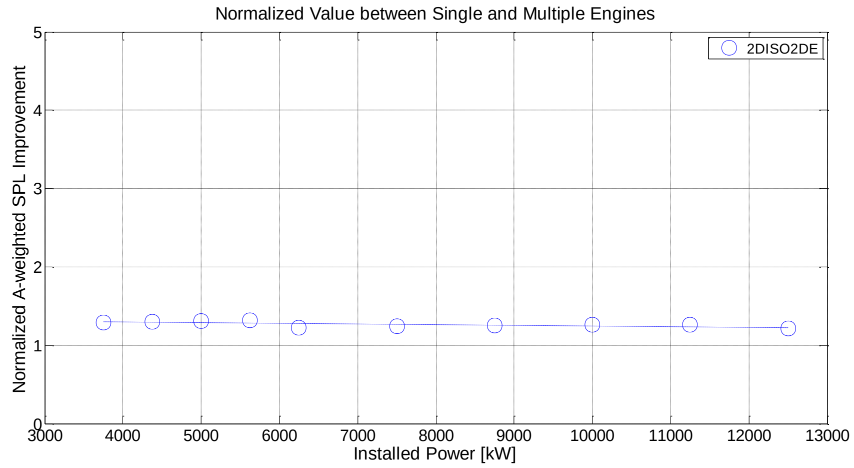


Figure 5.24 Number of engines and shafts changing over varying installed power by changing the number of cylinders Airborne Noise Results

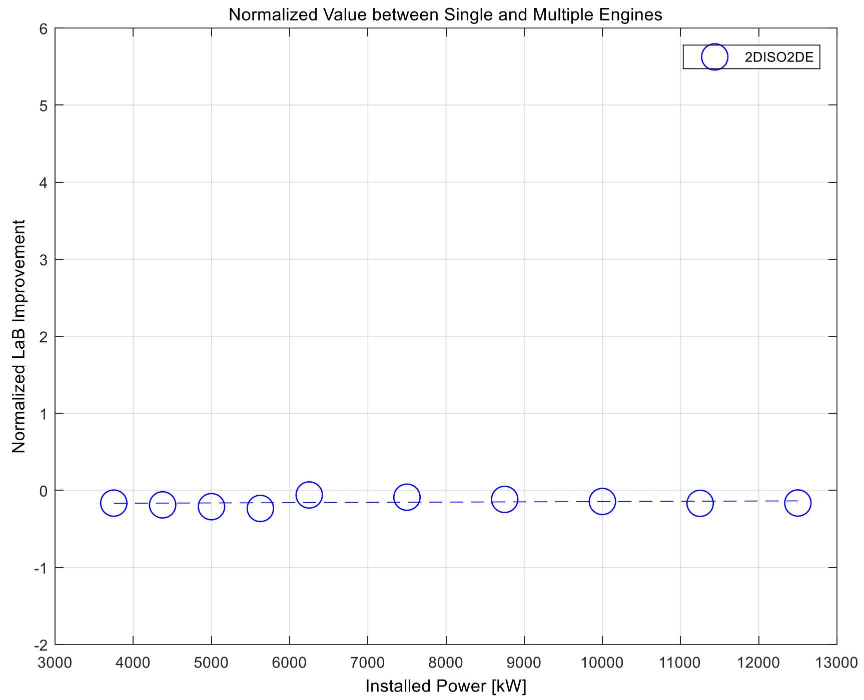


Figure 5.25 Number of engines and shafts changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

Loading point changing

The analysed diesel-hybrid plant here is assumed to have the delta of 0.5. The electric motor nominal power sets to be minimum for every installed power. The gamma is varying from 10% to 100% with the 10% interval. The results of this sensitivity analysis will be presented in the normalised values. It can be calculated using equation 5.3 and 5.4.

The improvements are declining for both airborne and structure-borne noise against the increasing installed power. The decreasing trend towards higher installed power is due to an increase in the electric motor noise level. At gamma 10%, the equipment that generates the airborne and structure-borne noise inside the engine room is only the gearbox and the electric motor.. The diesel engine noise level is also increasing but not as significant as the electric motor. The contribution of the diesel engine direct sound pressure level becomes smaller towards the higher installed power. This smaller contribution is due to the larger dimensions of the diesel engine, which makes the centre point of the engine further from the receiver location. Another loading point with the electric motor as the only propeller driver will result in the same improvement trend. The PTH modes is only available until gamma is equal to 30%. Above that point, the diesel-hybrid arrangement needs to use the combination of the diesel engine and electric motor to drive the propeller.

However, when the loading point is at its maximum, the diesel engine and gearbox will mask the sound of the electric motor. The diesel engine and gearbox noise level are significantly higher compared to the electric motor. The improvement at the maximum loading point is constant over the varying installed power.

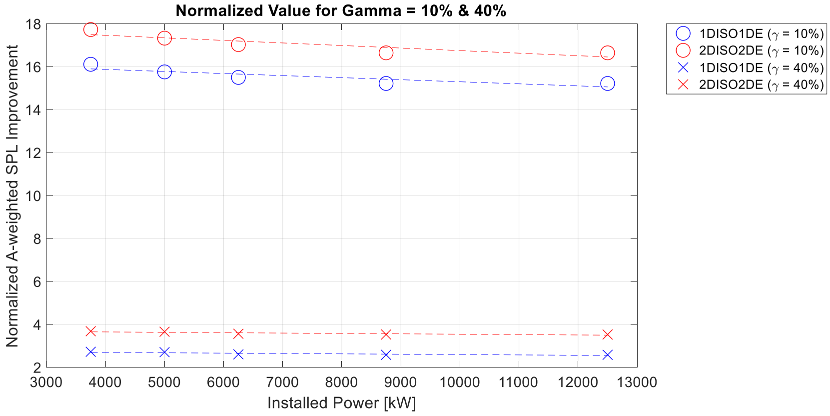


Figure 5.26 Loading Point changing over varying installed power by changing the number of cylinders Airborne Noise Results

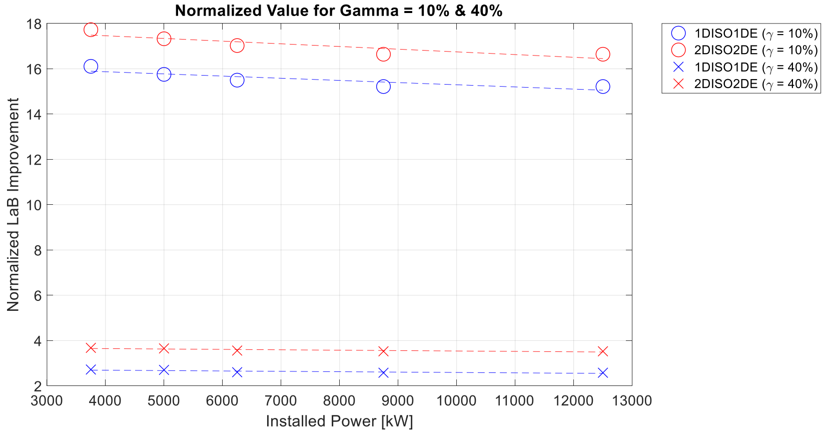


Figure 5.27 Loading Point changing over varying installed power by changing the number of cylinders Structure-borne Noise Results

5.4 Varying Installed Power by Changing the Power per Cylinder

In this analysis, one would keep the number of cylinders constant while the power per cylinder is changing. It is also important to see how the model behave when the designer chooses to increase the installed power by increasing the power per cylinder because it is directly related to the engine speed. From the previous chapter, it is known that the noise level heavily depends on the engine speed. The power per cylinder varies from 150 kW/cyl to 1250 kW/cyl. These values are the typical range for the four-strokes diesel engine. This analysis will be done for all the transmission types. All the results will be presented in the normalised values just like the increasing number of cylinders analysis.

5.4.1 Diesel Mechanical

Engine Requirements			
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**
5000 kW	**DM-1**	250	16
	DM-2	125	16
	DM-3	125	16
	DM-4	125	8
8000 kW	**DM-1**	400	16
	DM-2	200	16
	DM-3	200	16
	DM-4	200	8
10000 kW	**DM-1**	500	16
	DM-2	250	16
	DM-3	250	16
	DM-4	250	8
13000 kW	**DM-1**	650	16
	DM-2	325	16
	DM-3	325	16
	DM-4	325	8
15000 kW	**DM-1**	750	16
	DM-2	375	16
	DM-3	375	16
	DM-4	375	8
18000 kW	**DM-1**	900	16
	DM-2	450	16
	DM-3	450	16
	DM-4	450	8
20000 kW	**DM-1**	1000	16
	DM-2	500	16
	DM-3	500	16
	DM-4	500	8

Table 5.4 Input data for the number of engines and shafts changing

Number of engines and shafts changing

Four diesel-mechanical arrangements are going to be analysed in this analysis. These arrangements are the same as the analysis in subchapter 4.2.2. Two cylinder arrangements are used in this analysis despite the fact that one should keep the number of cylinders constant. The L-engine arrangement will be used for the four engines configuration. It is not possible to use the 16 cylinders in four engines configuration since at the low installed power the power per cylinder becomes lower than 100 kW. Moreover, the limit of the power per cylinder for the four diesel engine strokes makes it not possible to use the 8 cylinders for the single configuration in a high installed power. The results are presented in the normalised values. The equation 5.1 and the equation 5.2 are used to get the normalised values.

The airborne noise results show that the improvement is decreasing logarithmically towards the installed power. It is expected since the power per cylinder is inversely proportional to the nominal engine speed. In that sense, the higher the power per cylinder, the lower the nominal speed and the lower the noise will be.

After the installed power of 13000 kW, there is a huge jump in the airborne noise level improvement. The significant increase is because the single-engine has the nominal speed below 700 rpm due to its high power per cylinder and the multiple engine configurations use the engines with nominal speed above 700 rpm. As explained earlier in chapter 3, the SNAME model predicts significantly higher airborne noise level for the engines with nominal speed below 700 rpm.

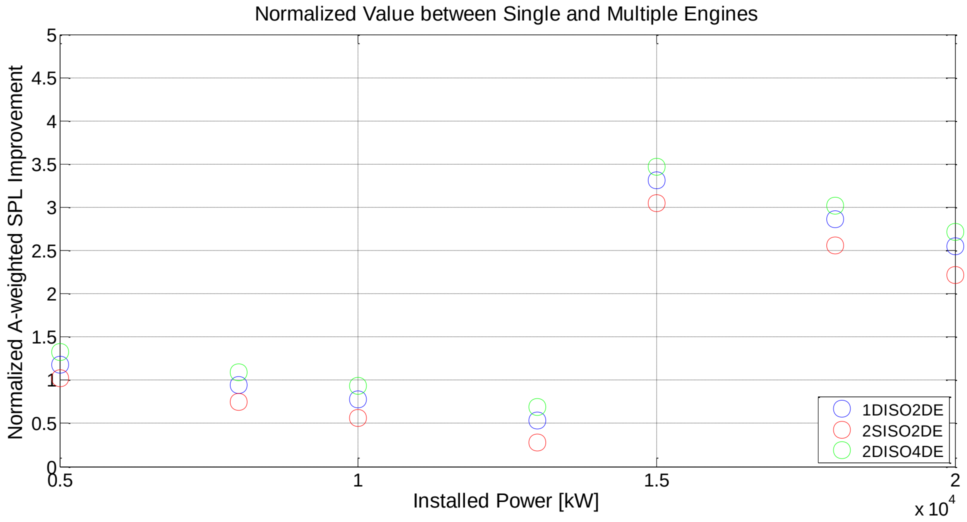


Figure 5.28 Number of engines and shafts changing over varying installed power by changing the power per cylinder Airborne Noise Results

The results of structure-borne noise show constant improvement. However, at 20000 kW there is a huge decline in the improvement. The single-engine at this power demand has the nominal speed below 600 rpm that makes the structure-borne noise source level from the diesel engine is significantly lower than the engine with nominal speed above 600 rpm. On the other hand, all the multiple engine configurations are using the engine with nominal speed above 600 rpm because its power per cylinder is rather low compared to the single engine.

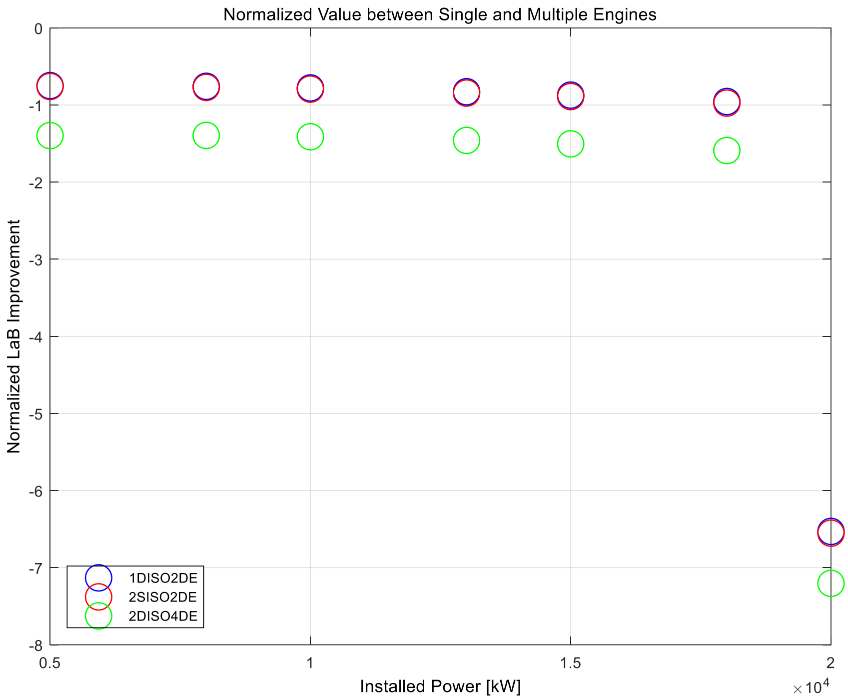


Figure 5.29 Number of engines and shafts changing over varying installed power by changing the power per cylinder Structure-borne Noise Results

Loading point changing

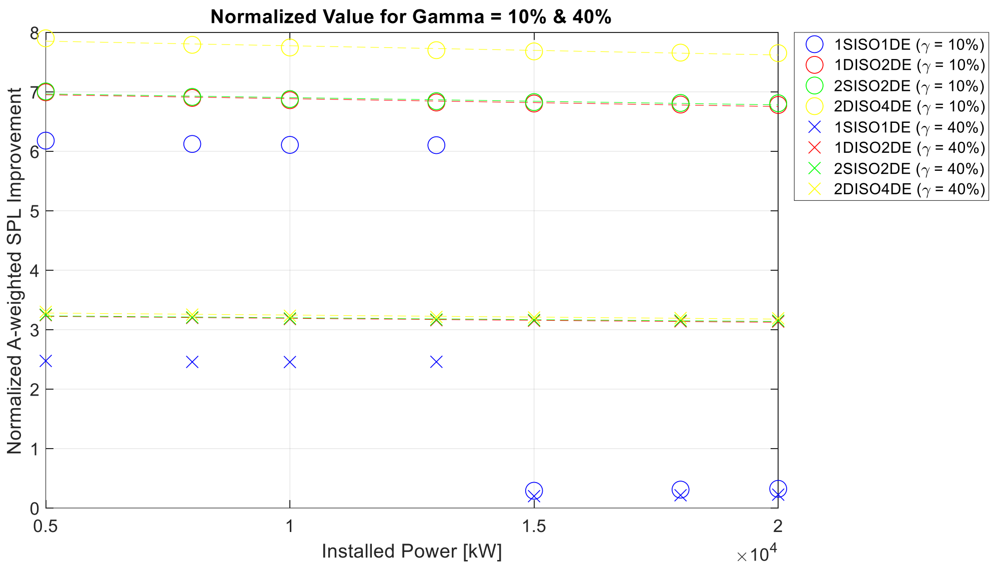


Figure 5.30 Loading point changing over varying installed power by changing the power per cylinder Airborne Noise Results

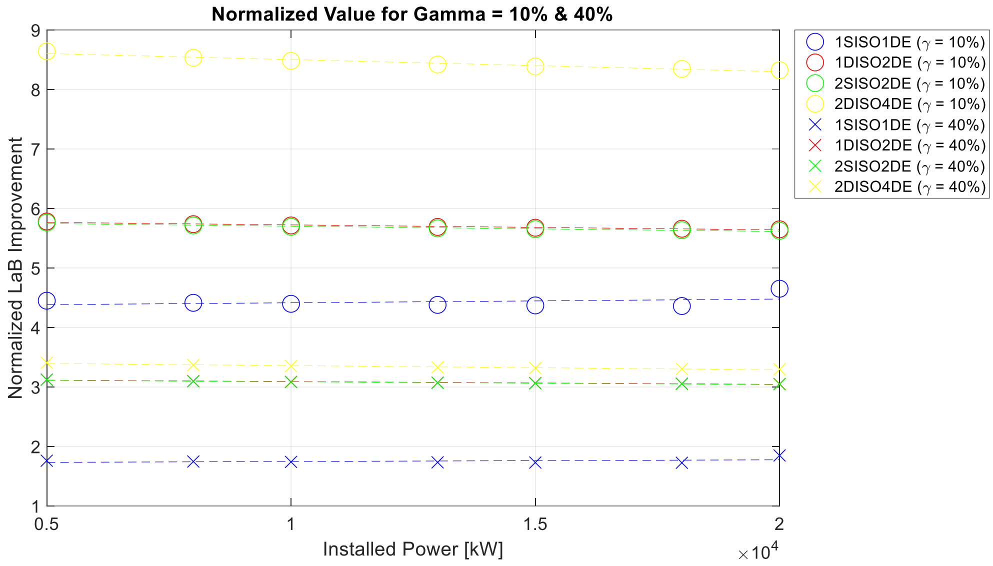


Figure 5.31 Loading point changing over varying installed power by changing the power per cylinder Structure-borne Noise results

The results above show the normalised values of two loading points to the maximum gamma. One can use the equation 5.3 and the equation 5.4 to get the normalised values for the loading point changing analysis. The airborne noise normalised values give a small declining improvement towards the increasing installed power. This trend is due to change of power per cylinder as the installed power is increasing. The power per cylinder affects the engine speed. The higher the power per cylinder the lower the engine nominal speed will be. If two engines with the same power have different nominal speed, the engine with the lower nominal speed will generate lower airborne noise level. This statement applies to the engine that has the nominal speed in the same cluster (above 700 rpm or below 700 rpm). It can be seen that there is a huge decline improvement for the single configuration between 13000 kW and 15000 kW. This decline is because the single engine in 15000 kW has the nominal speed below 700 rpm. The SNAME model distinguishes the empirical formula used to calculate the noise level for an engine with nominal speed above 700 rpm and below 700 rpm. The latter one does not have the actual speed or torque as the parameter to calculate its airborne noise level. Therefore, the airborne noise improvement s of the diesel engine with nominal speed below 700 rpm is almost zero.

The structure-borne noise results show a constant trend for the varying installed power. There is a slightly small improvement for the arrangement with a single engine configuration at 20000 kW installed power. This improvement is due to engine nominal speed belows 600 rpm. In structure-borne noise, a different cluster of speed only gives different constants (for variable A and B) but the basic equation stays the same. For engines with nominal speed below 600 rpm, it has a relatively lower constant compared to the engine with nominal speed above 600 rpm. This is the main reason on why there is a slight increase in the structure-borne noise level improvement at this point.

5.4.2 Diesel Electrical

Engine Requirements			
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**
5000 kW	**DE-1**	312.5	16
	DE-2	160	16
	DE-3	160	8
	DE-4	140	6
8000 kW	**DE-1**	500	16
	DE-2	250	16
	DE-3	250	16
	DE-4	225	8
10000 kW	**DE-1**	625	16
	DE-2	312.5	16
	DE-3	312.5	16
	DE-4	280	8
13000 kW	**DE-1**	812.5	16
	DE-2	410	16
	DE-3	410	16
	DE-4	370	8
15000 kW	**DE-1**	937.5	16
	DE-2	470	16
	DE-3	470	16
	DE-4	420	8
18000 kW	**DE-1**	1125	16
	DE-2	625	16
	DE-3	625	16
	DE-4	555	8
20000 kW	**DE-1**	1250	16
	DE-2	625	16
	DE-3	625	16
	DE-4	555	8

Table 5.5 Input data for Diesel-Electric analysis

Number of engines and shafts changing

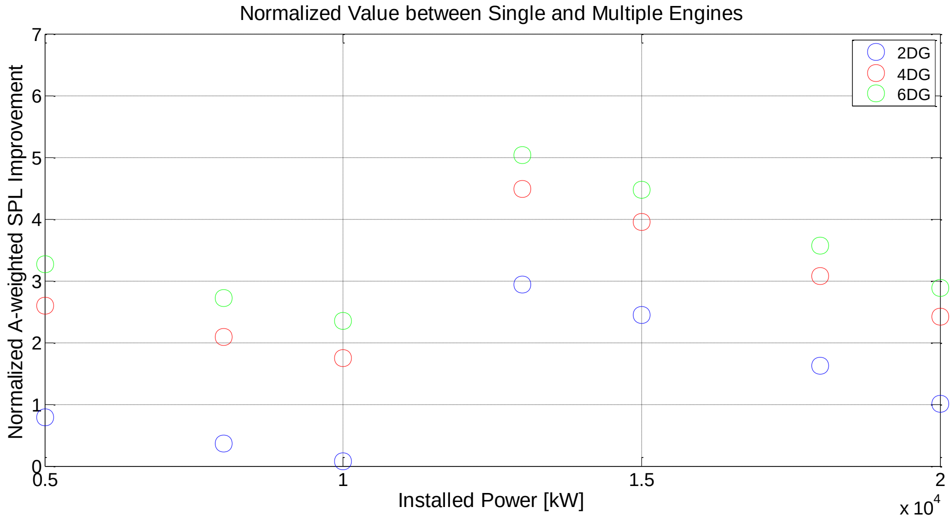


Figure 5.32 Number of engines and shafts changing over varying installed power by changing the power per cylinder Airborne Noise results

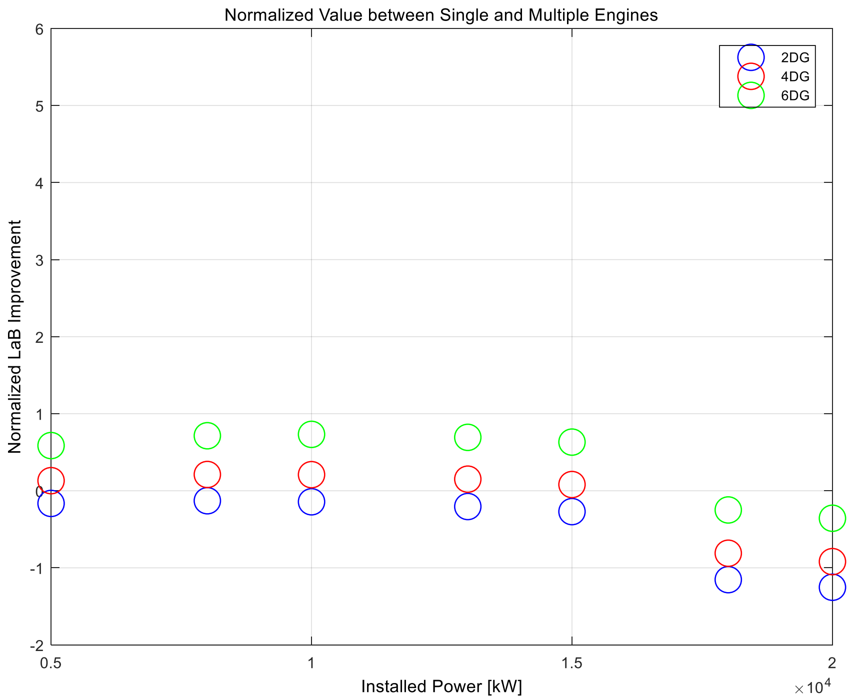


Figure 5.33 Number of engines and shafts changing over varying installed power by changing the power per cylinder Structure-borne Noise results

Four diesel-electrical plant arrangements are being investigated in this analysis. These arrangements are the same as the arrangements used in the subchapter 4.3.2. The arrangement with six engines will use the engine with L arrangement. It is not possible to implement V engine with 16 cylinders for six engines arrangement in the low installed power region. The reason is that the power per cylinder become too small in that low region. The normalised values are obtained by using the equation 5.5 and the equation 5.6.

The diesel-electrical plant airborne noise results also have the similar characteristic as the diesel-mechanical plant. There is a big difference between the installed power of 10000 kW and 13000 kW because the diesel generator set nominal speed for the latter one is below 700 rpm. The power per cylinder of the single-engine at 13000 kW installed power is 812.5 kW. On the other hand, the structure-borne noise results show a constant trend from 5000 kW to 15000 kW. The engine speed for the single engine in this range is above 600 rpm. Therefore, the difference is not evident. However, when the installed power is at 17000 kW, one can see there is a big decrease in the improvement because the engine speed becomes lower than 600 rpm. Contrary to the airborne noise, the structure-borne noise has a lower empirical constant when the speed becomes lower

Loading point changing

The input data for the simulation are presented in the tables above. The number of cylinders is kept constant while the installed power is increasing. The configuration with one, two and four engines use the engine with 16 cylinders while the six engines arrangement uses only 8 cylinders. This constraint is due to the restriction of minimum power per cylinder. Using six engines with 16 cylinders result in very low power per cylinder on the low installed power region. The normalised values are obtained from the equation 5.3 and the equation 5.4.

The improvement values for both the airborne and the structure-borne noise are constant against the changing installed power. However, for the six engines arrangement, there is a small decline towards the increasing installed power. This decreasing trend is due to the increase of the engine room dimensions becomes more significant than the noise source of the diesel generator sets itself. It will create the room constant for airborne noise and the distance travel by the structure-borne noise higher which results in lower noise.

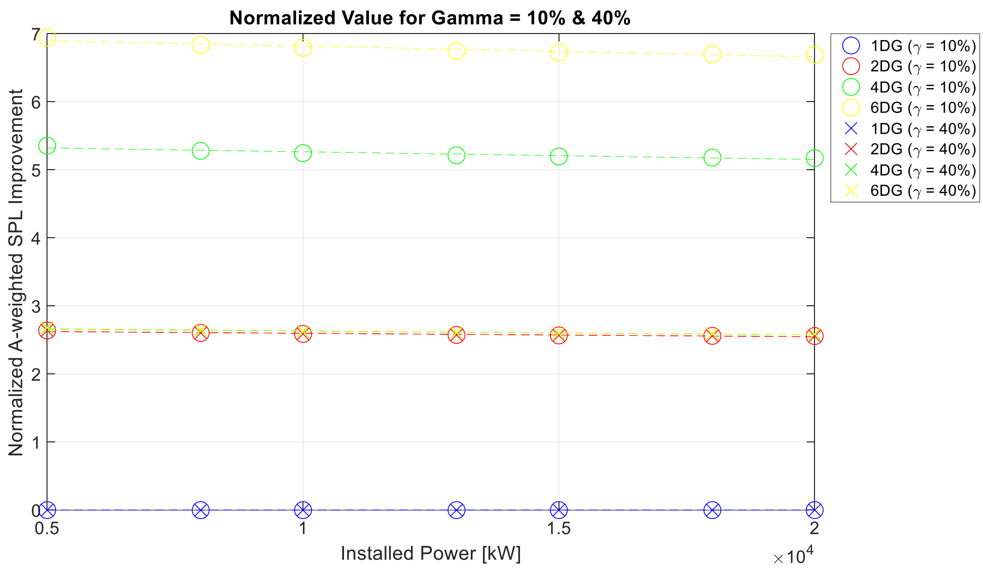


Figure 5.34 Loading Point changing over varying installed power by changing the power per cylinder Airborne Noise results

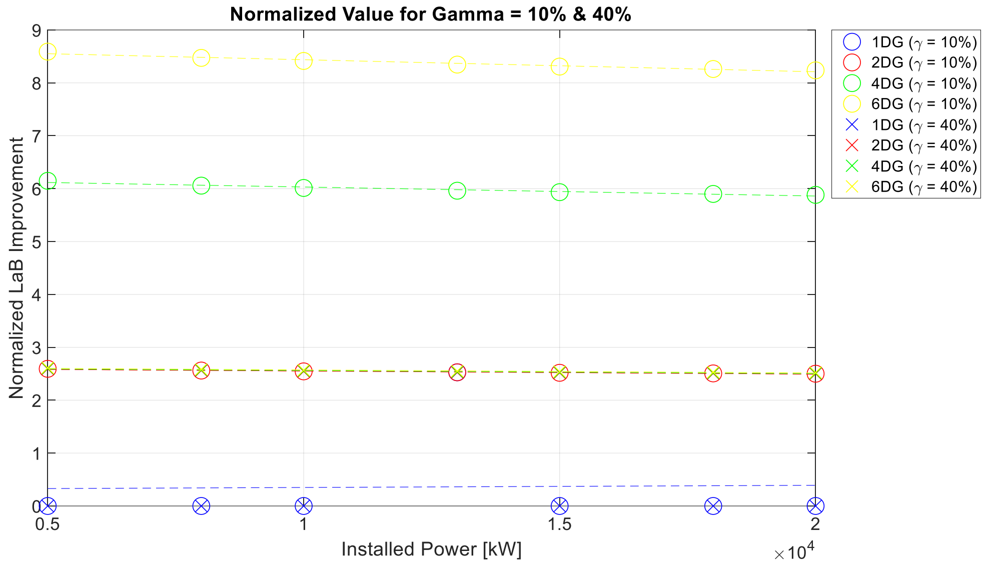


Figure 5.35 Loading Point changing over varying installed power by changing the power per cylinder Structure-borne Noise results

5.4.3 Diesel Hybrid

Engine Requirements			
Installed Power	**Arrangements**	**Pi (kW/cyl)**	**Z**
5000 kW	**DH-1**	312.5	8
	DH-2	160	8
8000 kW	**DH-1**	500	8
	DH-2	250	8
10000 kW	**DH-1**	625	8
	DH-2	312.5	8
13000 kW	**DH-1**	812.5	8
	DH-2	410	8
15000 kW	**DH-1**	937.5	8
	DH-2	470	8
18000 kW	**DH-1**	1125	8
	DH-2	625	8
20000 kW	**DH-1**	1250	8
	DH-2	625	8

Table 5.6 Input data for Diesel-Hybrid analysis

Number of engines and shafts changing

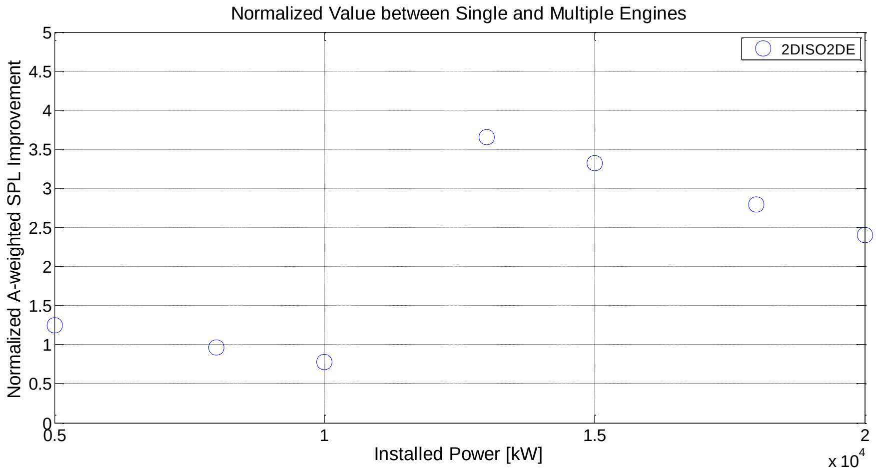


Figure 5.36 Number of engines and shafts changing over varying installed power by changing the power per cylinder Airborne Noise results

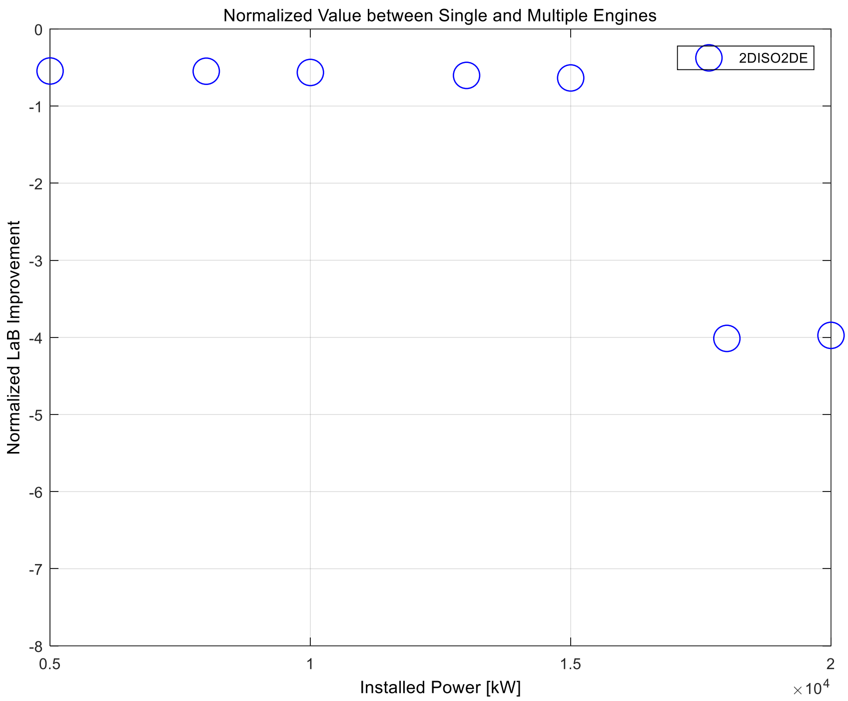


Figure 5.37 Number of engines and shafts changing over varying installed power by changing the power per cylinder Structure-borne Noise results

Two arrangements are analysed in the diesel-hybrid plant. In the diesel-hybrid plant, there is no need to distinguish the cylinder arrangement between the analysed diesel-hybrid configurations. All the arrangements are using L-cylinder arrangement with 8 cylinders. The equations to get the normalised values can be found in equation 5.7 and equation 5.8.

The results of diesel-hybrid also show the similar trend as the diesel-mechanical and diesel-electrical. The airborne noise shows a declining trend because the power per cylinder is increasing which leads to decrease in diesel engine nominal speed. However, in the structure-borne noise when the power per cylinder is increasing the improvement stays almost constant. The huge decline happens because the diesel engine nominal speed is below 600 rpm.

Loading point changing

The input data for the simulation are presented above. The delta chosen is 0.5. It will be kept constant over the varying installed power. The range of power per cylinder is 300 – 1250 kW/cyl for the single-engine configuration and 150 – 625 kW/cyl for the multiple engines configuration. The results are presented in the normalised values. Equations 5.3 and equation 5.4 are used to calculate the normalised values of every installed power.

The results of one engine configuration show a big jump between two installed power both for the airborne and structure-borne noise. This significant increase is due to the similar reason as in the diesel-mechanical plant. The engine used in the 13000 kW has a nominal speed below 700 rpm because the power per cylinder is high. One needs to lower the nominal speed because the mean effective pressure should be kept on the typical value. The change in nominal speed will result in a very high jump between the 10000 kW and 13000 kW for airborne noise improvement. Furthermore, the big jump also happens in the structure-borne noise results. The diesel engine used after 15000 kW has a nominal speed below 600 rpm that makes the structure-borne noise improvement suddenly drop.

The results of two engines configuration show a declining trend towards the increasing installed power. The single-engine configuration shows the same trend too, except that it has the jump between 10000 and 13000 kW. The reason for the declining trend is due to the increase of the electric motor noise level as explained earlier in the subchapter 5.3.3.

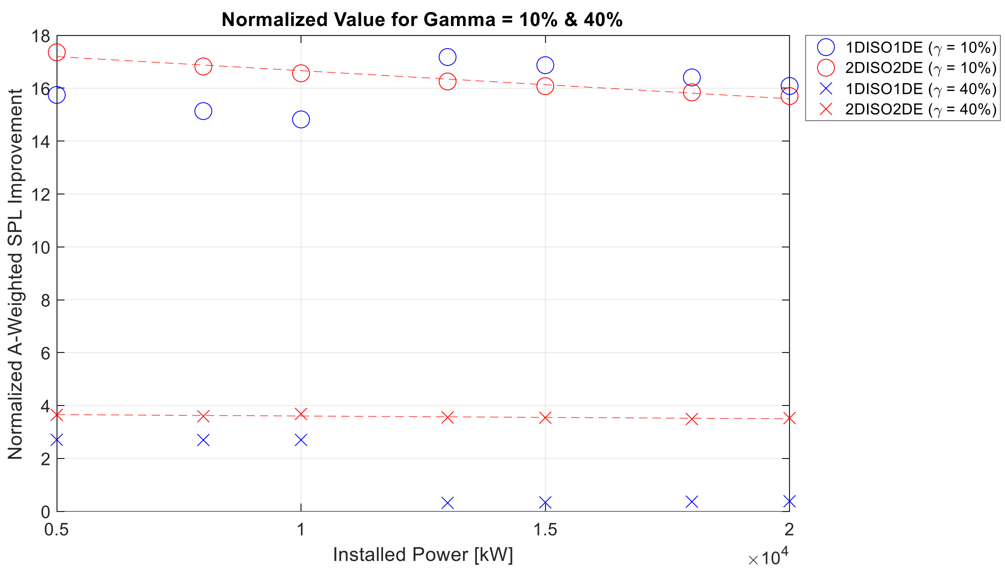


Figure 5.38 Loading Point changing over varying installed power by changing the power per cylinder Airborne Noise results

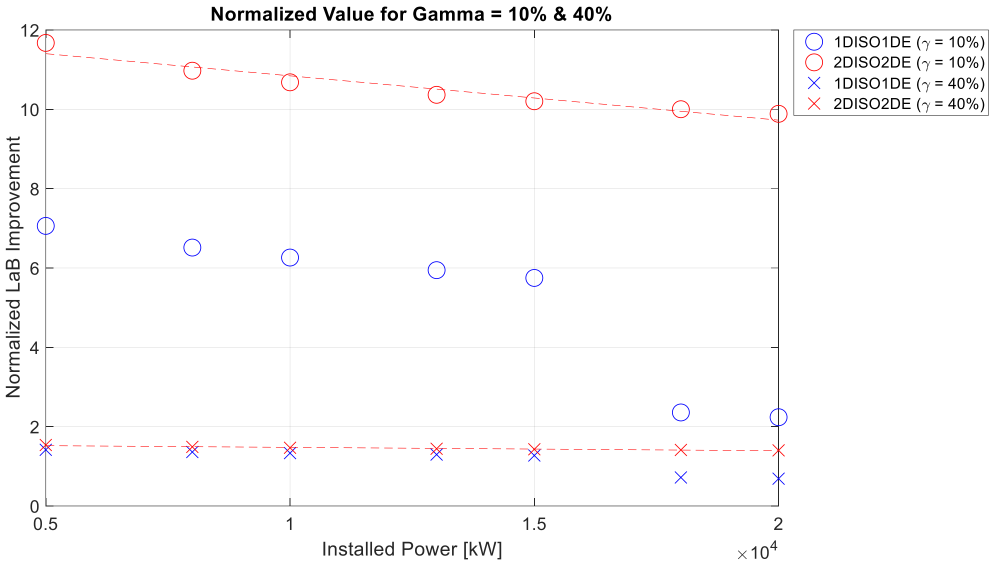


Figure 5.39 Loading Point changing over varying installed power by changing the power per cylinder Structure-borne Noise results

5.5 Varying Installed Power by Changing the Number of Cylinders and Keeping the Propulsion System Configuration Constant

This analysis is similar to the sensitivity analysis done in sub-chapter 5.3. However, instead of normalising each of multiple configurations to the single-engine configuration, this analysis will normalise the noise level of an arrangement at a certain installed power to its noise level at 5000 kW installed power. It has to be done in order to see how the increase of the installed power affects the propulsion system noise level by increasing the number of cylinders. The noise level results are calculated when the propulsion systems work at nominal loading point (100%).The propulsion configurations for every transmission type are the same as the configurations used in the subchapter 5.3. Furthermore, the number of cylinders of the engine per configuration is also the same as in subchapter 5.3.

5.5.1 Diesel Mechanical

There are four types of configurations for the Diesel-Mechanical Plant. Every configuration will be normalised to its configuration noise level at 5000 kW installed power. The normalised value can be written as follows,

$$L^*_{p,eq} = \frac{L_{p,eq@Ptot=5000kW} - L_{p,eq@Ptot=\cdots}}{L_{p,eq@Ptot=5000kW}} \quad (5.9)$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@Ptot=5000kW} - L_{aB,eq@Ptot=\cdots}}{L_{aB,eq@Ptot=5000kW}} \quad (5.10)$$

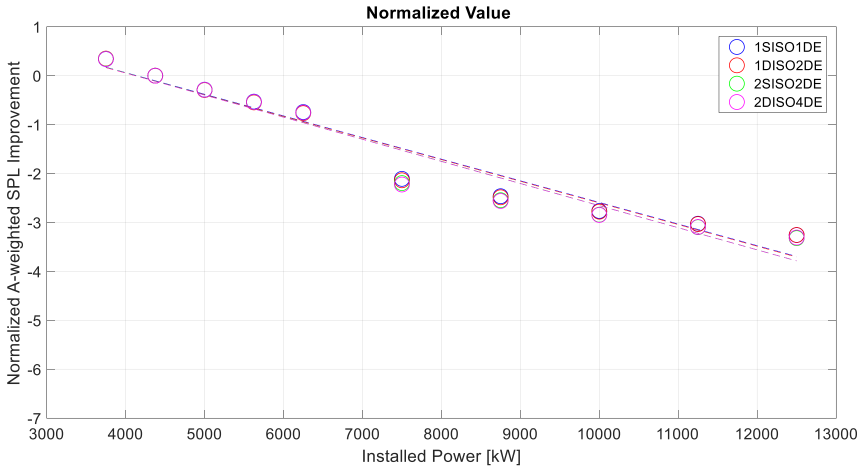


Figure 5.40 Normalised values of diesel-mechanical configurations over varying installed power by changing the number of cylinders Airborne Noise results

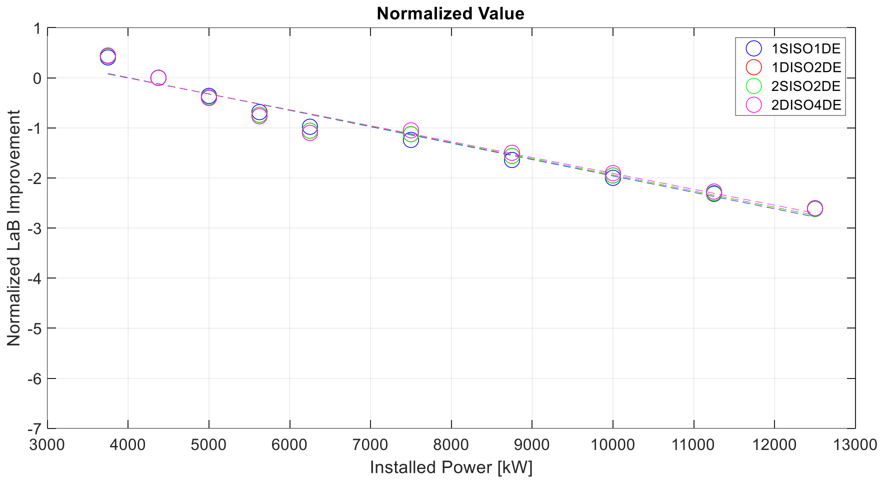


Figure 5.41 Normalised values of diesel-mechanical configurations over varying installed power by changing the number of cylinders Structure-borne Noise results

The airborne noise and the structure-borne results show that the outputs are affected by the total installed power. Obviously, the configuration with higher installed power will always give higher noise. It is expected because the nominal power of the diesel engine and the gearbox are one of the main variables in the empirical formula to calculate the equipment noise level. Every configuration more or less has the same normalised values when the installed power is increased.

The improvements of the airborne and the structure-borne noise by increasing the installed power from 3750 kW to 6250 kW for each of transmission type are linear. The engines used in this range are diesel engines with L-cylinder arrangement. However, there is a significant airborne noise decline between the installed power of 6250 kW and 7500 kW. This decline is because starting from the installed power of 7500 kW the diesel engines with V-cylinder arrangement are used. The diesel engine with V-cylinder arrangement produces higher airborne noise compared to the L-cylinder arrangement.

5.5.2 Diesel Electrical

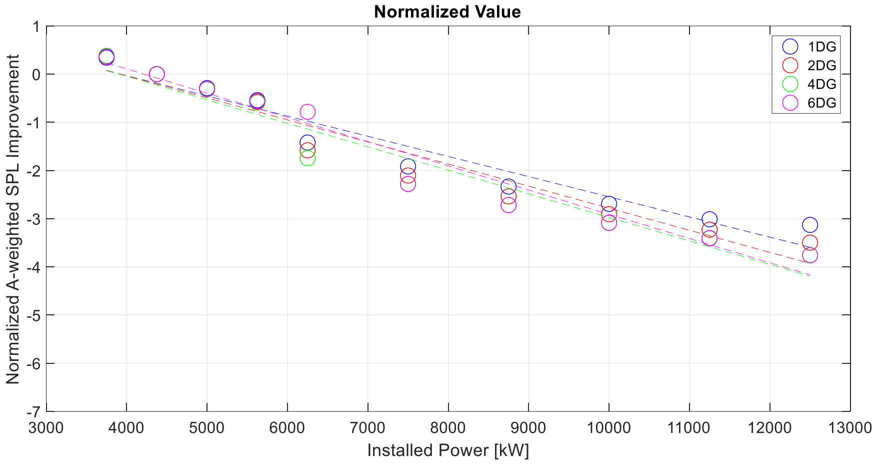


Figure 5.42 Normalised values of diesel-electrical configurations over varying installed power by changing the number of cylinders Airborne Noise results

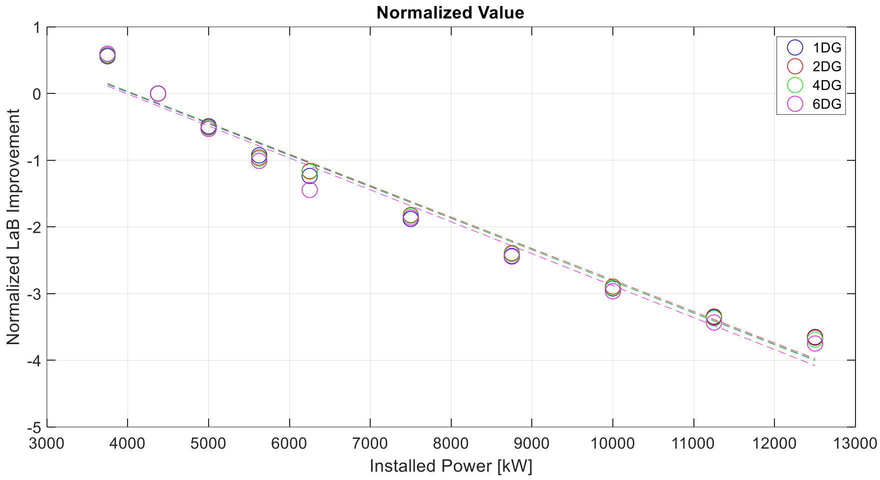


Figure 5.43 Normalised values of diesel-electrical configurations over varying installed power by changing the number of cylinders Structure-borne Noise results

The normalised values of the diesel-electrical results are achieved by using the equation 5.9 and 5.10. The airborne noise and structure-borne results show that the outputs are affected by the total installed power, similar to the diesel-mechanical. The airborne noise could be higher if the configuration used the diesel engine with V-cylinder arrangement instead of the L one. On the other hand, the diesel engine with V-cylinder arrangement gives lower noise in the structure-borne results.

5.5.3 Diesel Hybrid

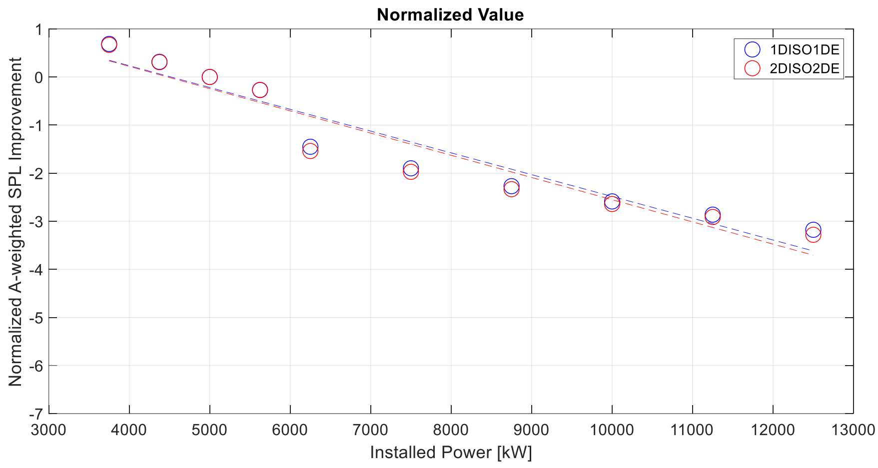


Figure 5.44 Normalised values of diesel-hybrid configurations over varying installed power by changing the number of cylinders Airborne Noise results

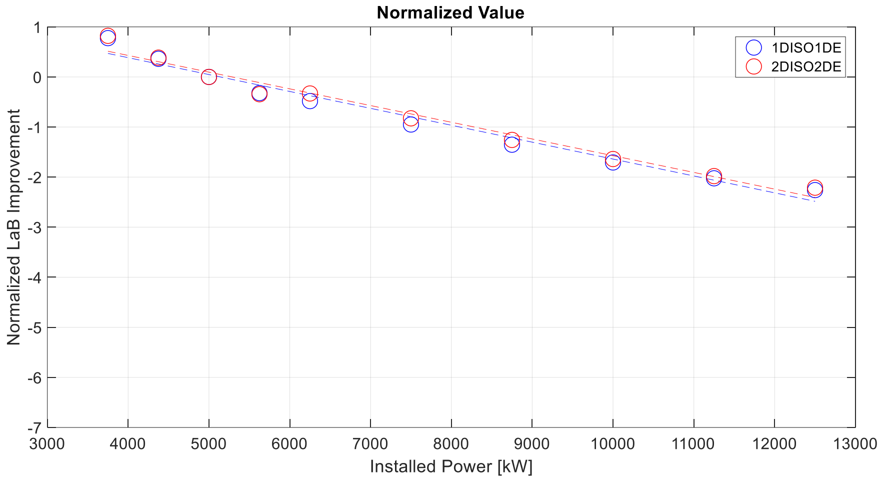


Figure 5.45 Normalised values of diesel-hybrid configurations over varying installed power by changing the number of cylinders Structure-borne Noise results

The normalised values of the diesel-hybrid results are achieved by using the equation (5.9) and (5.10). The airborne noise and structure-borne results show that the outputs are affected by the total installed power, similar to the diesel-mechanical. A higher installed power means higher nominal power for the diesel engine(s), the diesel generator set(s), the reduction gear(s) and the electric motor(s). This is why above 5000 kW the normalised values are negative since they have higher power which will lead to the higher noise level. The improvement by changing the installed power is the same for each of the configurations.

5.6 Varying Installed Power by Changing the Power per Cylinder and Keeping the Propulsion System Configuration Constant

The analysis in the previous subchapter is repeated here, but instead of increasing the number of cylinders, one increases the power per cylinder to increase the total installed power. It is important to do this analysis in order to understand the consequence of increasing the installed power by changing the power per cylinder for a propulsion system configuration. The noise level results are calculated when the propulsion systems work at nominal loading point (100%).The propulsion system arrangements for every transmission type are the same as the configurations used in the subchapter 5.4. The power per cylinder of the engine(s) per configuration is also similar as in subchapter 5.4.

5.6.1 Diesel Mechanical

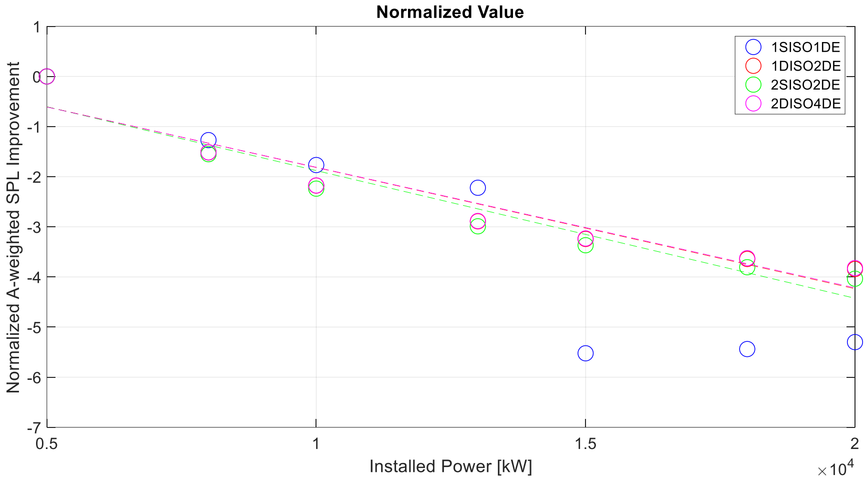


Figure 5.46 Normalised values of diesel-mechanical configurations over varying installed power by changing the power per cylinder Airborne Noise results

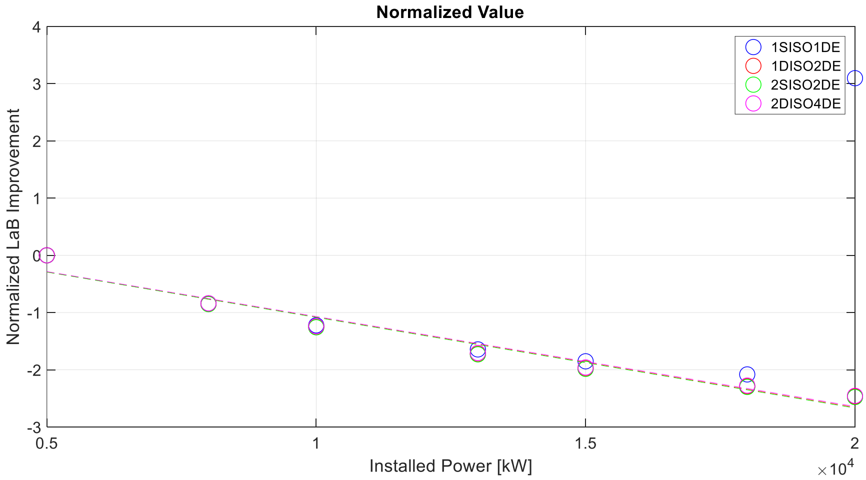


Figure 5.47 Normalised values of diesel-mechanical configurations over varying installed power by changing the power per cylinder Structure-borne Noise results

The normalised values are calculated by using the equation 5.9 and 5.10. Every installed power is normalised to the noise level at 5000 kW. The airborne noise and the structure-borne noise results show that increasing the installed power by changing the power per cylinder will also increase the noise level. The airborne noise normalised values between different configurations are almost the same for the installed power ranging from 5000 kW to 13000 kW,. However, after the 13000 kW installed power, the arrangement with single engine shows a significant decrease. This reduction is due to the engines used on 1-DM configuration (after the installed power of 13000 kW) have nominal speed below 700 rpm. Furthermore, the airborne noise results after 15000 kW show a slightly increasing trend. It means that although the installed power is increased, the noise level is decreased. It is somehow not very intuitive, but this trend shows that the SNAME Noise model for a diesel engine with nominal speed below 700 rpm is more sensitive to the change of nominal speed rather than the nominal power. When the power per cylinder becomes higher, the nominal speed of the engine becomes lower.

The structure-borne noise has a huge jump in the installed power of 20000 kW. This is because for the single-engine configuration at 20000 kW the diesel engine used has a nominal speed lower than 600 rpm. The single configuration at 20000 kW shows a positive normalised value which means that it has lower structure-borne noise level compared to the single configuration at 5000 kW thanks to its lower speed.

5.6.2 Diesel Electrical

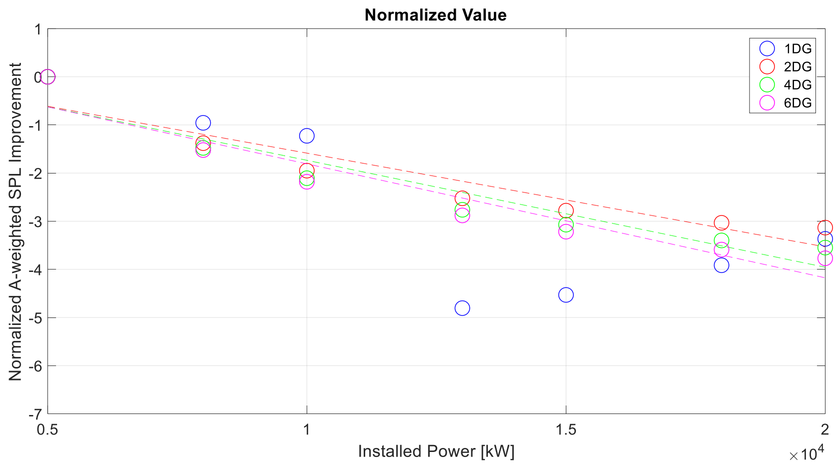


Figure 5.48 Normalised values of diesel-electrical configurations over varying installed power by changing the power per cylinder Airborne Noise results

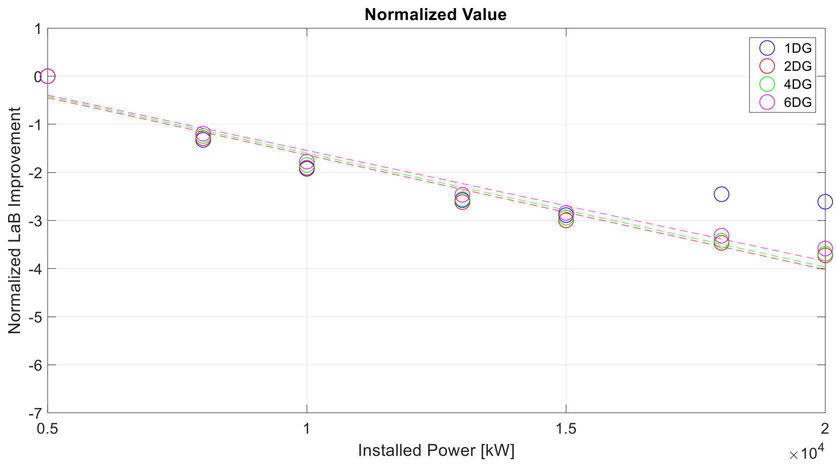


Figure 5.49 Normalised values of diesel-electrical configurations over varying installed power by changing the power per cylinder Structure-borne Noise results

The normalised values are calculated by the equation (5.9) and (5.10). Increasing the installed power will increase the noise level of the diesel-electrical plant. The normalised values for the multiple engines configuration show a linear trend both for the airborne and the structure-borne noise. Moreover, the normalised values per multiple engines configuration are similar to each other for every installed power. This shows that the multiple engine configurations give similar increases in noise levels. It is important to note that the increase in noise levels here means the increase of noise level relative to the noise level at 5000 installed power. However, for the single-engine configurations the results show a similar characteristic as the diesel-mechanical plant. For the airborne noise, when the diesel engine has a nominal speed below 700 rpm there will be a huge decline in the normalised values. On the other hand, for the structure-borne noise, there will be an increase in the normalised values for the propulsion systems that use diesel engines with nominal speed below 600 rpm.

5.6.3 Diesel Hybrid

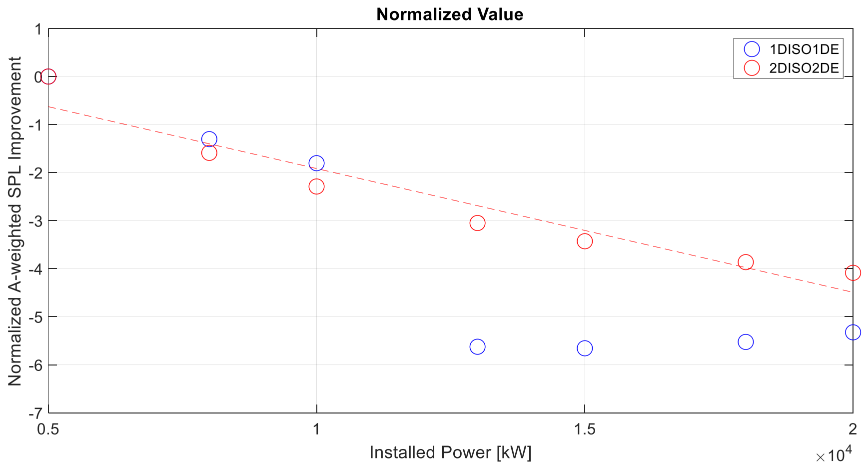


Figure 5.50 Normalised values of diesel-hybrid configurations over varying installed power by changing the power per cylinder Airborne Noise results

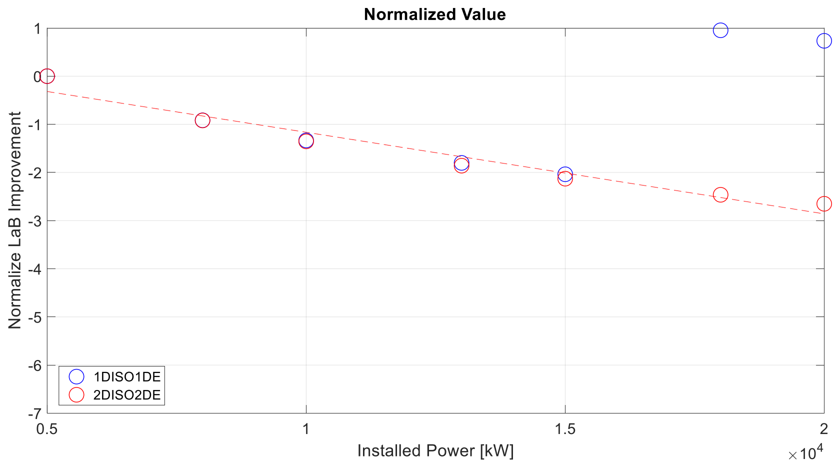


Figure 5.51 Normalised values of diesel-hybrid configurations over varying installed power by changing the power per cylinder Structure-borne Noise results

The normalised values are calculated by the equation 5.9 and 5.10. The multiple engines configuration and the single-engine configuration have the same normalised values until a certain installed power. It means that the increase in noise levels are similar but of course the overall noise level will be different. The huge decline in single engine configuration airborne noise result is due to the nominal speed of the diesel engine used is below 700 rpm. Furthermore, the huge increase in the structure-borne noise results is because the diesel engines have a nominal speed below 600 rpm.

5.7 Varying Propulsion and Auxiliary Load Split

The previous analysis assumes that the split between propulsion and auxiliary is always 80% and 20% respectively. This analysis will vary the split between the propulsion and auxiliary load to see how the model behave when this parameter is changed. Therefore, the propulsion power can be written as follows,

$$P_{prop,\beta} = P_{tot} * \beta \tag{5.9}$$

Where β is the split factor between the propulsion and auxiliary load. If the beta is equal to zero it means that there is no propulsion load required. On the other hand, if the beta is equal to one the ship does not require the auxiliary load. The auxiliary power can be expressed as,

$$P_{aux,\beta} = P_{tot} * (1 - \beta) \tag{5.10}$$

The beta will vary from 0.1 to 0.9 with an interval of 0.1. The results are presented in the normalised values. It can be written as follows,

$$L^*_{p,eq} = \frac{L_{p,eq@\beta=100\%} - L_{p,eq@\beta=10\%}}{L_{p,eq@\beta=100\%}} \tag{5.11}$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@\beta=100\%} - L_{aB,eq@\beta=10\%}}{L_{aB,eq@\beta=100\%}} \tag{5.12}$$

The noise level at a certain beta will be normalised to the noise level at the maximum beta. The analysis is done for the installed power of 10000 kW. The power per cylinder remains constant when the arrangement is changing. The reason is that one would like to see only the effect of beta. It can be seen from the previous results that the model is rather sensitive to the power per cylinder.

5.7.1 Diesel Mechanical

Engine Requirements			
No	**Variables**	**Value**	
1	**z (DM-1)**	16	Cylinders
2	**z (DM-2 & DM-3)**	8	Cylinders
3	**z (DM-4)**	6	Cylinders
4	**Pi (DM-1)**	500	kW/cyl
5	**Pi (DM-2 & DM-3)**	500	kW/cyl
6	**Pi (DM-4)**	330	kW/cyl
7	**Beta**	[0.1:0.1:0.9]	

Table 5.7 Input for Diesel-Mechanical beta analysis

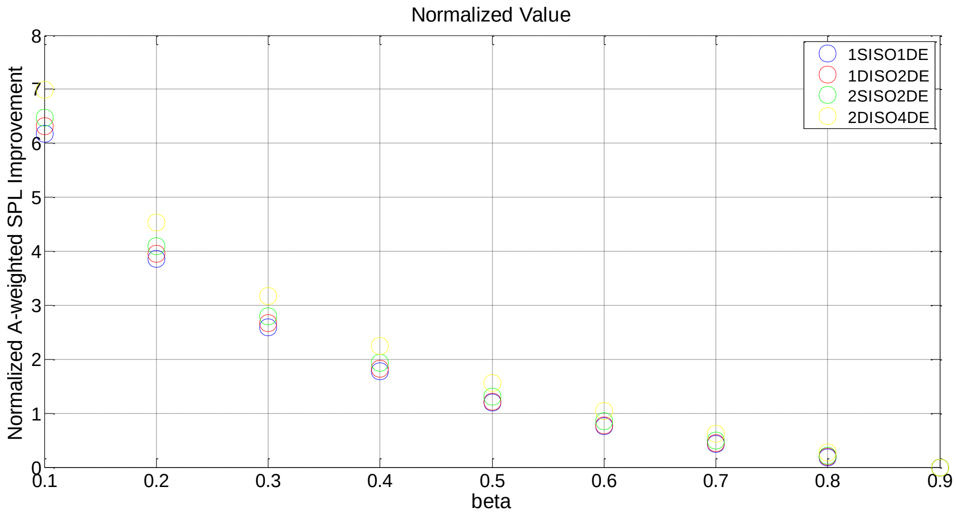


Figure 5.52 Beta changing Airborne Noise results

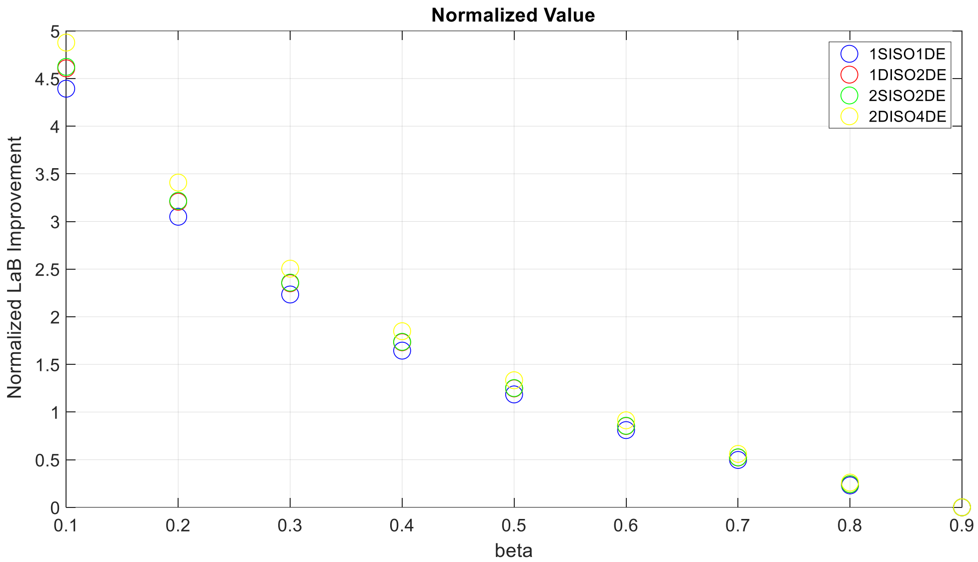


Figure 5.53 Beta changing Structure-Borne noise results

The results for both airborne and structure-borne noise of the propulsion systems show that the highest improvement is achieved when the beta is equal to 0.1. The propulsion load is low when the beta at its lowest point. Consequently, the diesel engine installed in the propulsion driver room has significantly lower brake power compare to the engine when the beta is maximum. Obviously, the arrangement with smaller diesel engine brake power will have lower noise level. When the beta is equal to 0.9, the improvement is zero because everything is normalised to the noise level at maximum beta.

5.7.2 Diesel Electrical

Engine Requirements			
No	**Variables**	**Value**	
1	**z (DE-1)**	18	Cylinders
2	**z (DE-2)**	9	Cylinders
3	**z (DE-3)**	8	Cylinders
4	**z (DE-4)**	8	Cylinders
5	**Pi (DE-1)**	555	kW/cyl
6	**Pi (DE-2)**	555	kW/cyl
7	**Pi (DE-3)**	312.5	kW/cyl
8	**Pi (DE-4)**	210	kW/cyl
9	**Beta**	[0.1:0.1:0.9]	

Table 5.8 Input for Diesel-Electric beta analysis

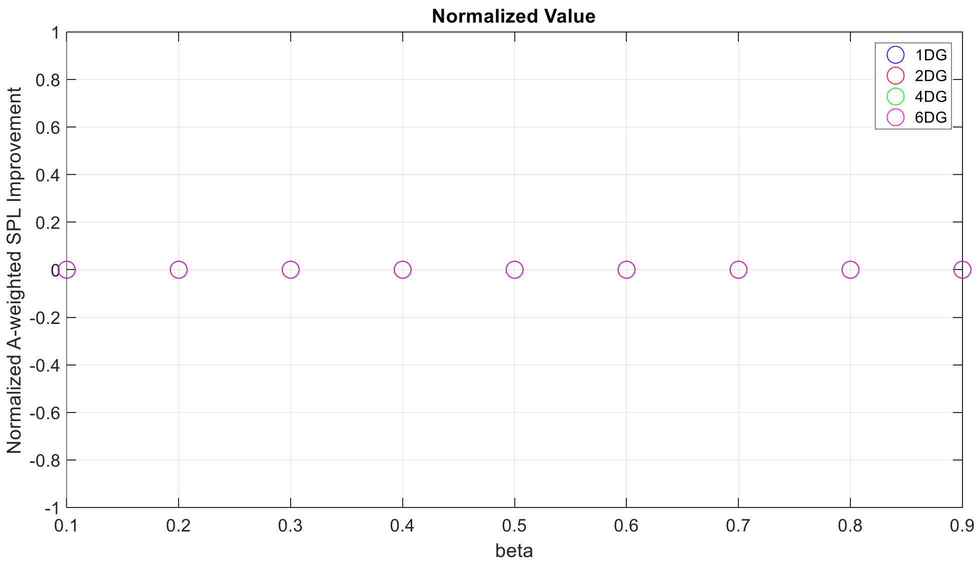


Figure 5.54 Beta changing Airborne Noise results

The diesel-electrical plant uses the diesel generator sets to provide both the propulsion and the auxiliary load. In that sense, even if the split between the propulsion and auxiliary changes, the power that needs to be generated by the diesel generator sets stay the same. The results do not show any variation in the changing beta. The normalised values are always zero since the installed power in all beta is the same.

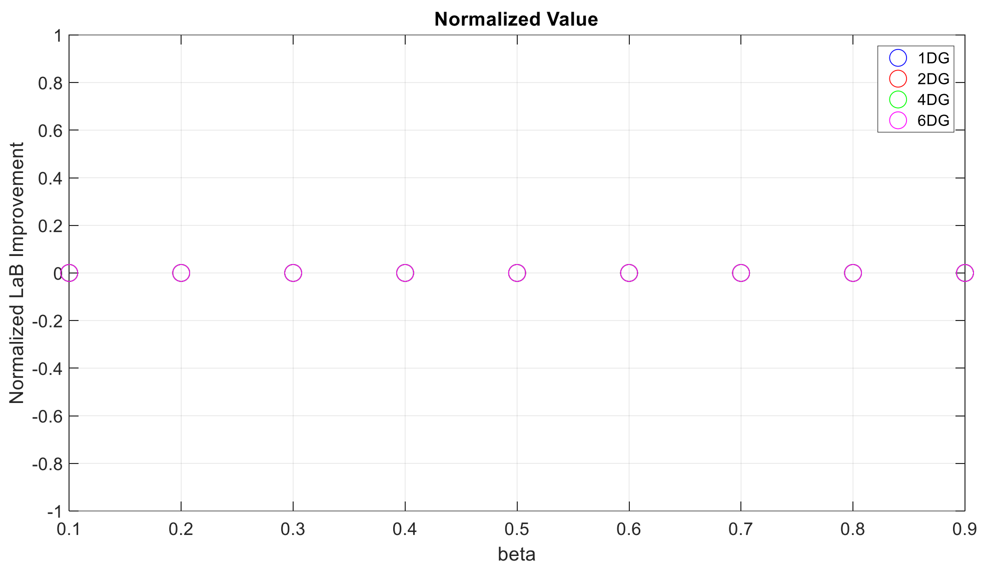


Figure 5.55 Beta changing Structure-borne Noise results

5.7.3 Diesel Hybrid

Engine Requirements			
No	**Variables**	**Value**	
1	**z (DH-1)**	10	Cylinders
2	**z (DH-2)**	8	Cylinders
3	**Pi (DH-1)**	500	kW/cyl
4	**Pi (DH-2)**	312.5	kW/cyl
5	**Beta**	[0.1:0.1:0.9]	
6	**Delta**	0.5	

Table 5.9 Input for Diesel-Hybrid beta analysis

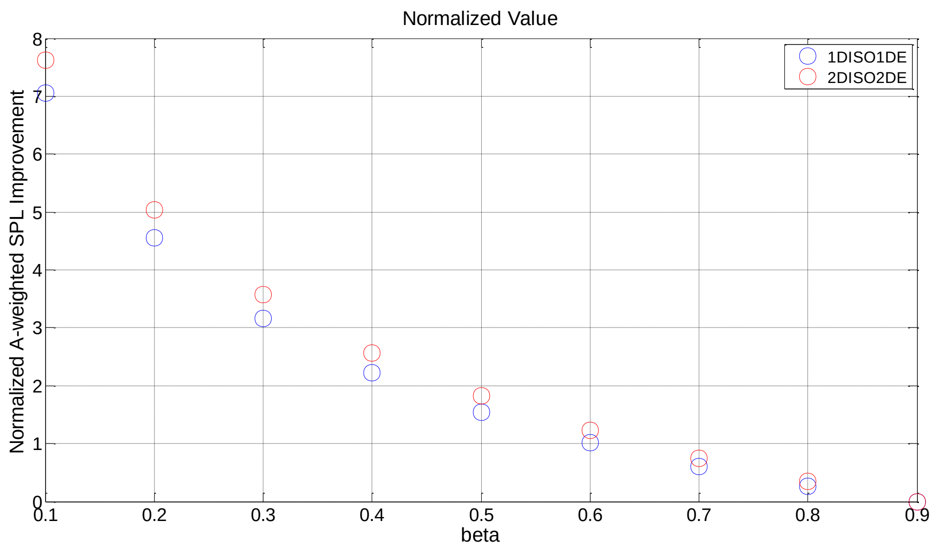


Figure 5.56 Beta changing Airborne Noise results

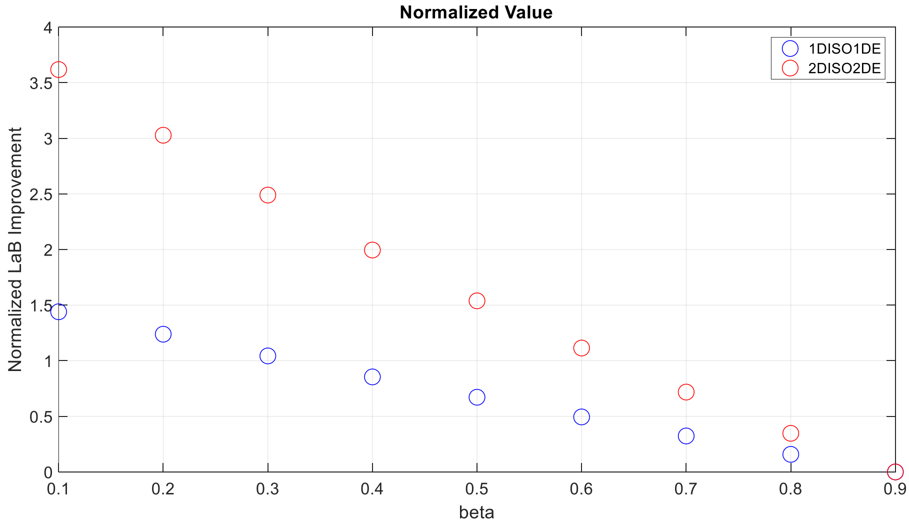


Figure 5.57 Beta changing Structure-borne Noise results

The diesel-hybrid plant has the same characteristic as the diesel-mechanical plant. The noise improvements for the propulsion systems are declines toward higher beta. The decreasing beta is directly proportional to the diesel engine power and the electric motor power. This relation will result in lower noise level as one can see from the graph above.

5.8 Verification of SNAME Noise Level

The noise level predicted by SNAME will be verified to the noise level from the engine catalogue. The diesel engine airborne noise levels are compared to the diesel engine manufactured by MAN and Rolls-Royce while the structure-borne noise levels are compared to the Wartsila Engine. MAN and Rolls-Royce engine catalogues do not give any information about the structure-borne noise level of its engine. The predicted electric motor noise levels are compared to the noise level from the ABB induction motor catalogue. However, there is no information regarding the structure-borne noise level of the electric motor from the catalogue. Therefore, only the airborne noise is verified for this equipment. The predicted diesel generator set airborne noise levels are compared to the measurement of Rolls Royce diesel generator set. The catalogue also does not provide the structure-borne noise information. Furthermore, the gearbox noise levels are compared to the value from the TNO report.

5.8.1 Diesel Engine

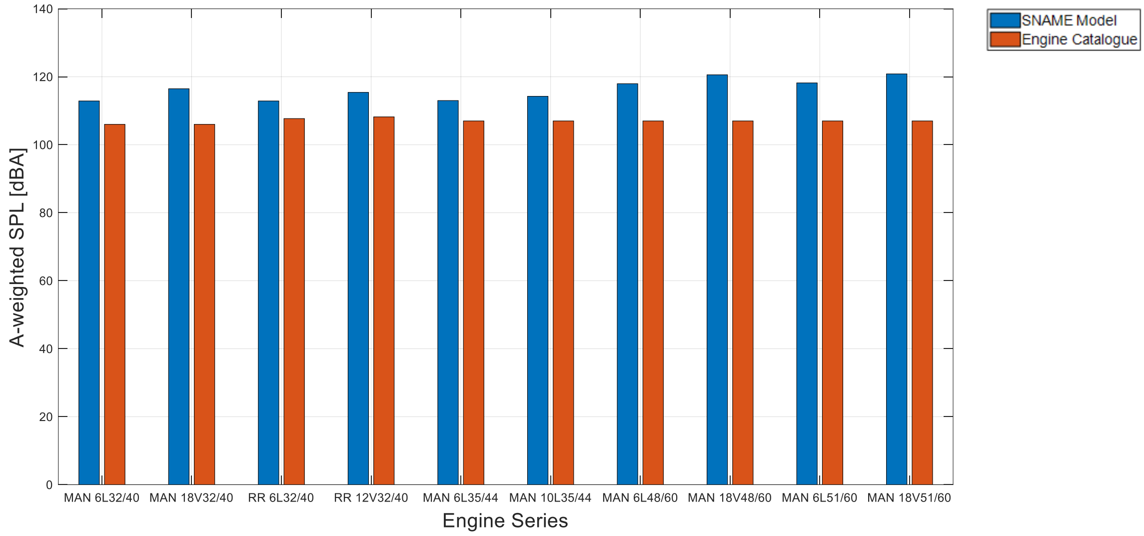


Figure 5.58 Diesel Engine Airborne Noise verification results

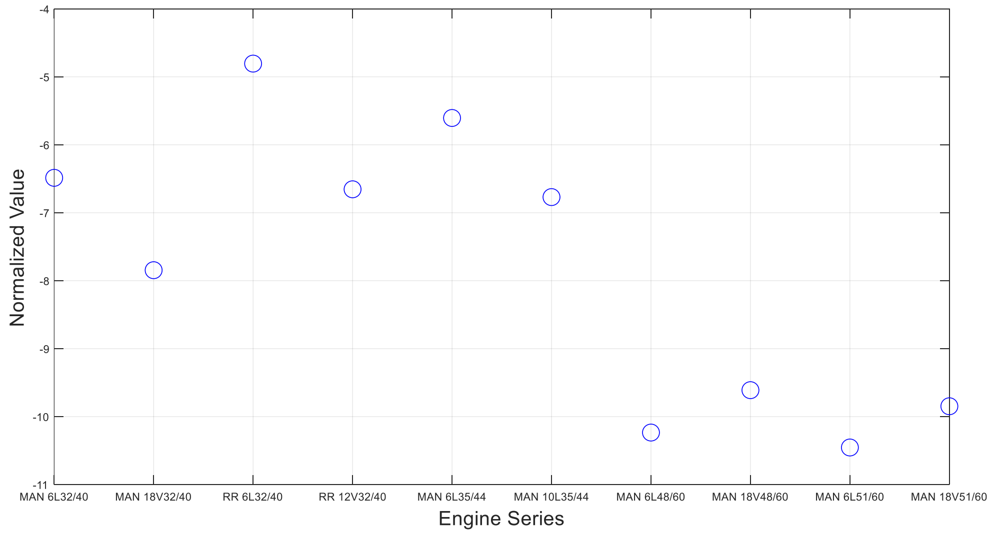


Figure 5.59 Normalised Diesel Engine Airborne Noise verification results

$$L^*_{p,eq} = \frac{L_{p,eq@measurement} - L_{p,eq@SNAME}}{L_{p,eq@measurement}} \tag{5.13}$$

$$L^*_{aB,eq} = \frac{L_{aB,eq@measurement} - L_{aB,eq@SNAME}}{L_{aB,eq@measurement}} \tag{5.14}$$

The airborne noise and the structure-borne noise levels of diesel engines from the SNAME prediction and the engine catalogues are compared here. The values from the engine catalogues are typical values. It does not depend on the number of cylinders of the engine and the power of the engine. It only depends on the engine bore diameter engine and the power per cylinder. The measurements of sound pressure levels from MAN are taken one meter away from the engine surface with 20 measurement points. The points are distributed evenly around the engine according to ISO 6798. There is no room correction is performed. It means that the unfavourable effects of a room's acoustics affect the airborne noise level measurement by MAN. However, there is no information regarding the specification of the test room uses by MAN to measure the airborne noise level. The SNAME sound pressure levels are also calculated by using one meter as the distance from the receiver to the diesel engine. Since MAN catalogues do not describe the specification of the room, it is assumed the boundary surfaces material is steel. The measurement uses to define the A-weighted sound pressure level is dBA with the reference pressure of 20 μPa.

From the figures above it can be seen that the SNAME prediction is always higher compared to the noise level from the catalogue. For the MAN 6L32/40 the difference between the predicted and the noise from the MAN catalogue is approximately 6 dBA. However, for MAN 18V32/40 the difference is higher. It is expected since the MAN noise levels are the typical values while the SNAME prediction is a function of the nominal power, the nominal speed and the actual speed. For MAN 48/60 and MAN 51/60 the difference between the SNAME value and the engine catalogue value becomes higher, approximately 11 dBA. One could argue this is due to the conservative character of the SNAME airborne noise model for a diesel engine with nominal speed below 700 rpm. Both MAN 48/60 and MAN 51/60 have a nominal speed below 700 rpm. This overestimation of the diesel engine airborne noise levels with engine speed below 700 rpm affect the results of the varying power per cylinder sensitivity analysis.

Another thing that is relevant to the overestimation of the airborne noise level is the room constant. It is possible that the test room used by MAN has higher noise absorption value. The most common type of test room to measure the sound pressure level of equipment is an anechoic room. It has a very high absorption value that creates the contribution by the reverberant sound pressure level is negligible. However, there is no information provided in the catalogue whether they use the anechoic room. The prediction by SNAME assumed the wall of the test room is steel which has a very low noise absorption value.

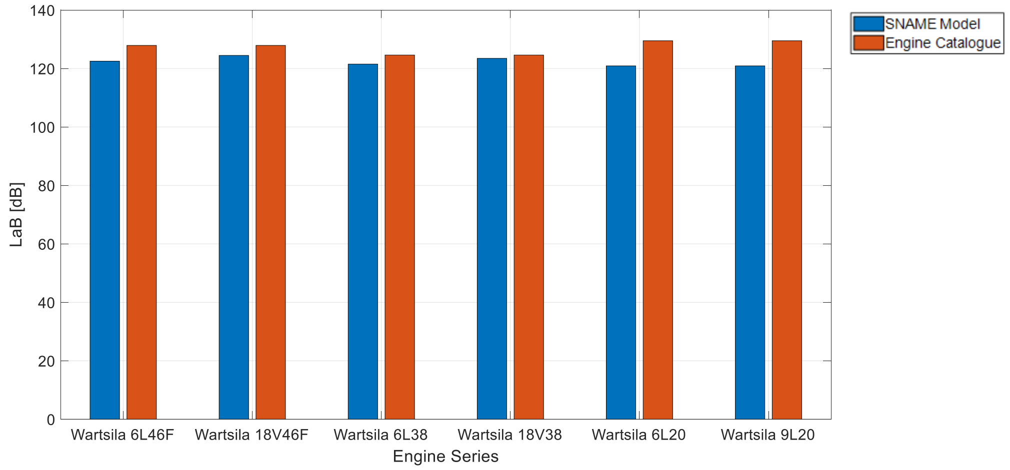


Figure 5.60 Diesel Engine Structure-borne Noise verification results

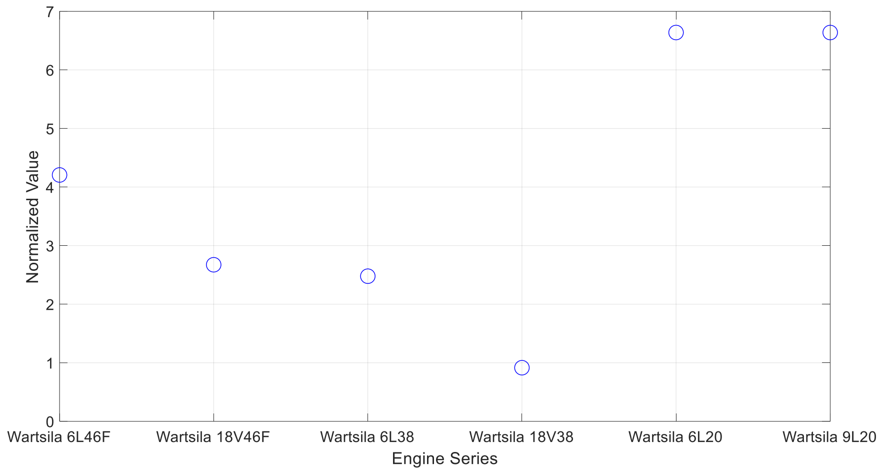


Figure 5.61 Normalised Diesel Engine Structure-borne Noise verification results

The SNAME structure-borne noise levels from the Wartsila catalogue are compared to the structure-borne noise levels predicted by SNAME Model. The values from Wartsila catalogue are also typical values. It only depends on the power per cylinder and the bore diameter. The structure-borne noise is measured above the mounting, which means the free vibration of the diesel engine. The measure of the structure-borne noise level is in dB with a reference value of $10^{-3}\ cm^2/sec$.

From the figures above, it shows that the SNAME underestimates the structure-borne noise levels. The structure-borne noise results from SNAME show a better fit with the values from catalogue compared to the verification of airborne noise results. The differences between the predicted structure-borne noise and the measured structure-borne noise of W18V46F are lower compared to W6L46F. It happens because the values from the catalogue are typical values. All the engines investigated in this analysis have nominal speed above 600 rpm. The accuracy of the SNAME prediction below 600 rpm is still unknown at this point.

5.8.2 Electric Motor

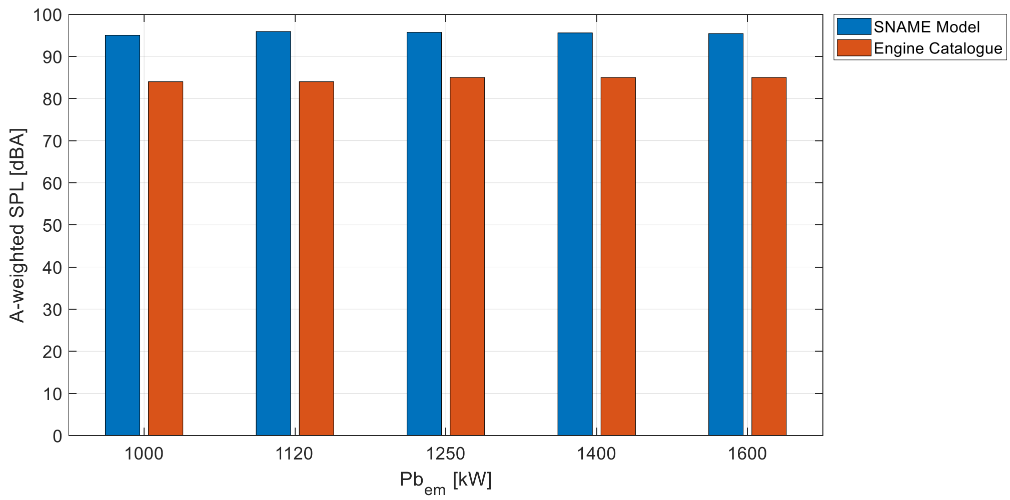


Figure 5.62 Electric Motor Airborne Noise verification results

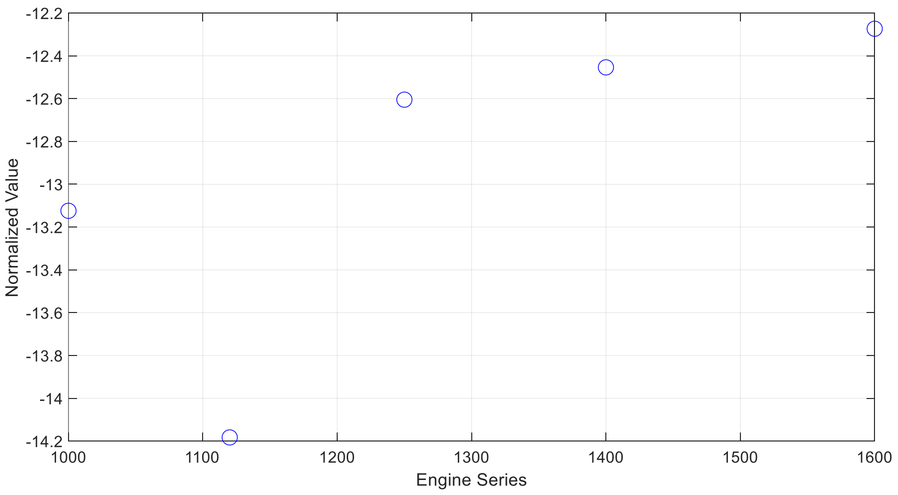


Figure 5.63 Normalised Electric Motor Airborne Noise verification results

The measured sound pressure levels are taken from the ABB Low Voltage Induction Motor for Process Application. The measurements are done by placing the microphone at various positions around the motor to measure sound radiation in different directions. The distance between the motor surfaces and the microphone is one meter. The sound pressure levels from the catalogue have a tolerance of 3 dB due to the variation of the noise level in different directions. The test room uses by ABB is designed to eliminate the reflected noise and external sources. The sound pressure levels of the electric motor by SNAME are calculated by using one meter as the distance between the receiver and the noise source. Since there is no reflection noise on the measurement condition, only the contribution from the direct sound pressure level is considered. The measure uses to define the A-weighted sound pressure level is dBA with the reference pressure of 20 μPa.

The results from the SNAME Model are significantly higher compared to the measurement values. One can argue because the SNAME model is designed specifically to calculate the noise level of an electric motor for the propulsion driver while the value from the catalogue is for the electric motor in process industry. However, the construction and the noise-driven factors for these types of motors are similar. Therefore, it is acceptable to use the noise level value of the process motor as a comparison.

The airborne noise level difference between the measurement value and the SNAME prediction is between 10-15 dBA. This high difference affects the prediction of overall airborne noise levels of the diesel-electric plant, the diesel-hybrid plant and the battery plant on this project. One could expect a lower overall airborne noise level from those transmission types if a more accurate model is used to predict the electric motor airborne noise level. In that sense, there is a potential to have a higher reduction noise by using the battery plant. The diesel-electrical plant and the diesel-hybrid plant could also have a decrease in overall noise but it would not be as significant as the battery plant since the diesel-engine noise contribution is still there.

The measurement data of the electric motor structure-borne noise level is not provided in the catalogue. Furthermore, it is hard to find the data from papers or journals since the authors keep the structure-borne noise level data as confidential. However, it is expected the prediction of electric motor structure-borne noise level is higher than its real acceleration level because the generator formula is used to calculate the structure-borne noise level. As explained in the subchapter 4.5.1, the generator formula is not very accurate since the contribution from the diesel engine contaminates its measurement.

5.8.3 Diesel Generator Set

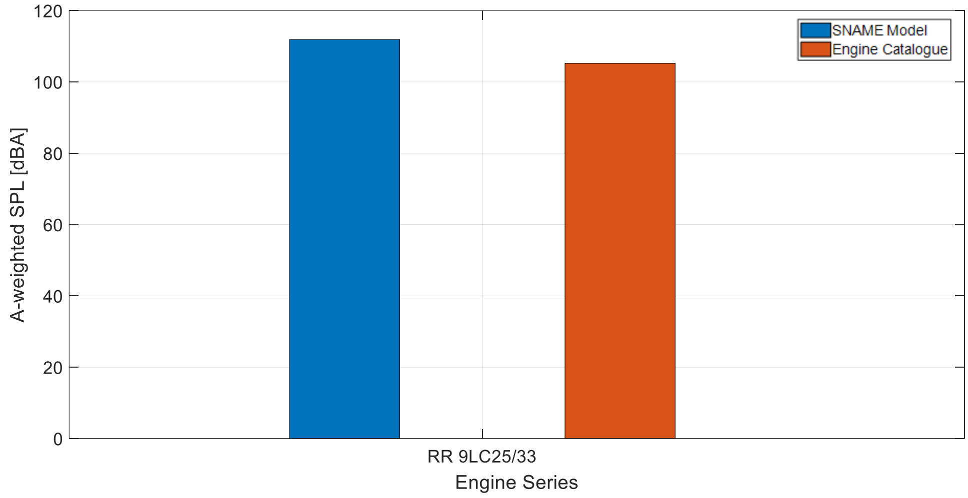


Figure 5.64 Diesel Generator Sets Airborne Noise verification Results

The airborne noise level of diesel generator set from the SNAME prediction and the engine catalogue are compared here. The value from the catalogue is taken from Rolls Royce generator sets project guide. The noise level from the Rolls Royce catalogue is not a typical value. The sound pressure level from Rolls-Royce is measured one meter away from the engine surface with 8 measurement points. The points are distributed evenly around the engine. Similar to the MAN catalogue, Rolls-Royce does not give information regarding the specification of the test room uses to measure the airborne noise level. The SNAME sound pressure levels are calculated by using one meter as the distance from the receiver to the diesel generator set. It is assumed the boundary surfaces material of the room is steel. The measurement uses to define the A-weighted sound pressure level is dBA with the reference pressure of 20 μPa.

The SNAME model prediction is slightly higher than the measurement data from Rolls-Royce. The normalised value is -6.3 which means the prediction overestimate the sound pressure level of the diesel generator set by 6.7 dBA . The difference is almost the same as the diesel engine. It is expected because the diesel engine airborne noise is masking the generator airborne noise. One can argue that its prime mover dominates the overall noise of a diesel generator set.

The structure-borne noise of the diesel-generator set is not provided in the catalogue. Therefore, the verification cannot be done for the diesel genset structure-borne noise level. However, since the diesel-engine also dominate the acceleration level of the whole generator set, one could expect the difference between the measurement data and the SNAME prediction for the diesel generator set is similar to the diesel engine.

5.8.4 Gearbox

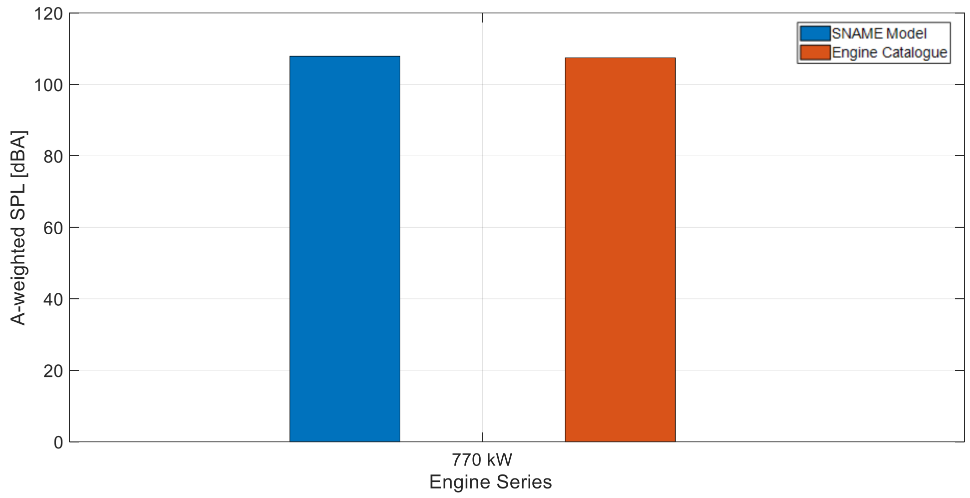


Figure 5.65 Gearbox Airborne Noise verification Results

The SNAME gearbox sound pressure level is compared to the gearbox sound pressure level from a report by TNO [44] . The values from TNO are not a measurement but also a prediction. It is then used to calculate the cabin noise of a yacht, which then compared to the measurement value. The TNO values of sound pressure levels in the receiver cabin show a very good agreement to the measurements. The deviations are really small, between 0.5-2.5 dB(A). The reason why one uses the sound pressure level of the TNO report is that the gearbox measurement data is not available at the moment.

The prediction by TNO calculates only the direct sound pressure level of the gearbox. The receiver is 1 m away from the noise source. Therefore, the calculation by SNAME also recreates the same condition as the TNO calculation. The gear manufacturing tolerance is D1 since the report from TNO states that the gearbox has poor quality.

The SNAME has a slightly higher A-weighted sound pressure level compared to the TNO Cabin. The normalised value is -0.5 which means the SNAME model overestimates the gearbox sound pressure level by 0.5 dBA. This overestimation will not affect the results of the overall sound pressure level of the diesel-mechanical plant and the diesel-hybrid plant since the deviation is low.

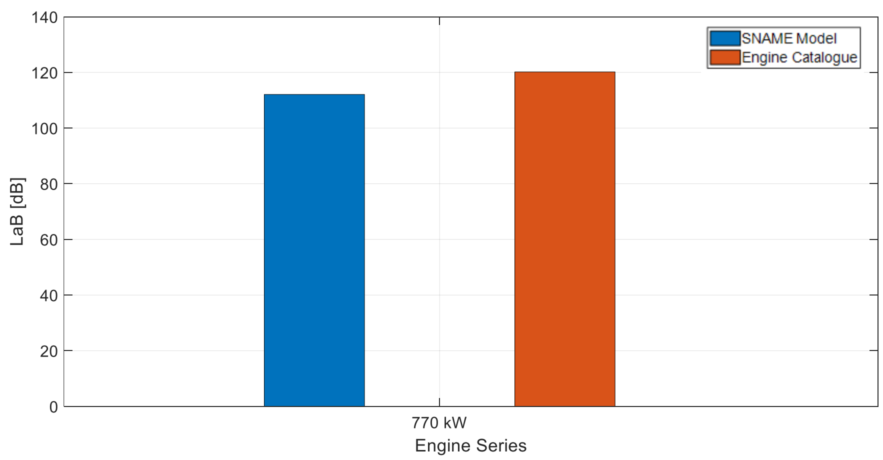


Figure 5.66 Gearbox Structure-borne Noise verification Results

The structure-borne noise level from SNAME also compared to the value of TNO Cabin. The value from TNO cabin calculates the acceleration level above the seating of the gearbox. Therefore, there is no transmission loss by the mounting and the foundation.

It can be seen from the figure that the SNAME model underestimates the structure-borne noise level of the gearbox. The difference between them is 8.1 dB or 6.7 in normalised value. The underestimation of the gearbox structure-borne noise level is similar to the diesel engine thus it will not affect the noise level of the diesel-mechanical plant and the diesel-hybrid plant.

6

Conclusions & Recommendations

6.1 Introduction

The proposed methodology is intended to give a deeper insight into the acoustical matters for the ship designer when designing the propulsion system. It is important to understand the consequences of early design choices toward the noise generated by the power configurations. The chosen design options are the transmission types, number of engines and number of compartments. The transmission types have the biggest influence on noise level compared to other parameters. Therefore, it needs to be analysed separately from other parameters. The engineer can use this project as a general design guideline in the preliminary design phase which will be beneficial in the later design stage. However, the method is limited only to the airborne and structure-borne noise excitation of the propulsion system equipment. It is also important to perform further research on the effects of early design choices toward the soundproofing potential.

In the first chapter of this project, the research questions are asked to determine the research steps. This final chapter will elaborate the summary of the research performed and describe what the main findings are. The main conclusions will be discussed in the sub-chapter 6.2 along with the answers to the research questions. Afterwards, sub-chapter 6.3 gives recommendations for further study and improvement.

6.2 Conclusions

From the first chapter, the research questions are asked to achieve the objectives of the project. The answers to these questions will be given in the following sub-chapters. The answers are based on the literature review and results that have been done in this project.

What is the most effective way to model the noise pollution from a propulsion system?

There are several steps need to be done to determine the noise of a propulsion system. It starts with selecting the propulsion equipment that has the significant noise source. There are lots of equipment installed in the power configuration but not all of them give a significant contribution to the overall noise level. This circumstance happens because the overall noise level is a log summation. The equipment that is being considered as noise sources depends on the transmission types. In general, the devices that have high noise level (both for airborne and structure-borne noise) are the diesel engine, the reduction gear, the diesel-generator set and the electric motor. In this project, the SNAME model is used to predict the noise level of these devices. Moreover, the SNAME model is also used to predict the transmission losses from the source to the receiver location. It is possible to use analytical model, such as FEA and SEA, to predict the noise level of the propulsion system. However, these models require a high level of details, which is not ideal for this project. There are some drawbacks when using the SNAME model. The nominal speed of the engine categorises the formula of diesel engine source levels, for both the airborne and the structure-borne noise. This clustering will create a big discrepancy, more or less 5 dBA, in the transition region between two categories as can be seen in the results of engine speed changing simulation. Another drawback is that the prediction of generator source level is not accurate enough since the contribution of its prime mover contaminates the measurement results.

After the noise source level is determined, the equipment and the engine room model are necessary to be developed. The dimensions of the engine room depend on the dimensions of the equipment and the equipment arrangement. It is important to develop these models because one would like to know the noise level at the location of interest inside the engine room. The airborne and the structure-borne transmission losses from the noise source to the receiver is a function of the engine dimensions and the compartment dimensions. Draw attention to the regression coefficients derived from the equipment-sizing model. Those coefficients are not valid for manufacturers that differ from the database. It should be noted that each manufacturer has its typical shape of the engine. Nevertheless, the methodology developed to build the sizing model can be implemented to predict the size of equipment from various manufacturers.

How will the early design choices affect the noise excitation and propagation of a propulsion system?

- *How do different propulsion system configurations that have similar transmission type perform in the area of noise pollution?*

The assumptions are made at the beginning of this thesis to determine which aspects of the propulsion system are relevant for the noise level. These aspects are the transmission types, the number of engines, the number of shafts, the number of compartments and the loading factors. The evaluation methodology is created based on these aspects to determine the effect of each parameter. The variations in these choices have different effects on the airborne noise and the structure-borne noise level. The following explanation will be divided into two sections, one for the airborne noise and the other one is for the structure-borne noise.

Airborne Noise

- Increasing the number of engines and shafts can decrease the airborne noise level. It depends on the transmission type and the number of engines itself. This statement applies to all the transmission types that are investigated in this project. The lowest sound pressure level is always generated by the arrangement with the largest number of engines, thanks to the higher room absorption of multiple engines configuration and low noise source level of the small engine.
- Increasing the number of rooms will decrease the airborne noise level at the location of interest between 1.4 until 2.1 dBA depending on the number

of engines and transmission types. This statement is also true for all the transmission types. The contribution of airborne noise from the adjacent engine room is negligible because the transmission loss from the bulkhead is high. However, the noise improvement should be compromised by providing extra width to the room dimensions.

- Decreasing the loading factors will lead to lower airborne noise levels because the engine does not work at its nominal speed and power. For all the transmission types, a configuration with the largest number of engines will give a better improvement due to the possibility to use only some of the small engines rather than to use one big engine in the part load condition. High noise reduction in the low load condition can be found in the diesel-hybrid. There is an option to go with PTH mode when sailing for relatively low power demand.

Structure-borne Noise

- Increasing the number of engines and shafts has different consequences for every transmission type. The diesel-mechanical plant structure-borne noise level is higher when the arrangement has more engines. The single-engine arrangement generates the lowest vibration level. The diesel-hybrid plant also gives higher structure-borne noise level for multiple engines arrangement. Nevertheless, the difference between the single engine and the multiple engines configuration is small. On the other hand, the diesel-electrical plant gives an improvement to the noise level when the number of engines is added.
- Increasing the number of compartment gives a very good improvement to all transmission types. The improvements could achieve 5.8 dB, 5.5 dB and 2.4 dB for the diesel-mechanical, the diesel-electrical and the diesel-hybrid plant respectively. These high values are due to the extra damping by the additional bulkhead between the adjacent rooms. The material of the bulkhead is assumed to be steel. However, if the material of the bulkhead is changed to another material with less density, it is expected that the improvement obtained by increasing the number of compartments will be smaller than the results presented in this project. In practice, it is unlikely to have the bulkhead weaker than steel due to safety reasons.
- Lower loading factors also lead to lower structure-borne noise levels. When the loading factor is below 40%, the multiple engines configuration gives much lower vibration levels compared to single engine configuration. Although with more engines the diesel-mechanical and the diesel-hybrid plant have a higher vibration level at the maximum loading point. There is, therefore, a trade-off when using the multiple engines configuration for the diesel-mechanical and the diesel-hybrid plant. Nevertheless, the increase in the structure-borne noise level at the maximum loading point is insignificant compared to the improvement that one can get in part load condition.

- *How do different transmission types perform in the area of noise pollution?*

Four transmission types are being compared in this thesis, the diesel-mechanical, the diesel-electrical, the diesel-hybrid and the battery plant. When varying the number of engines, the results show that the battery gives the least noise pollution for both the airborne noise and structure-borne noise. This result is expected since in the battery plant it is assumed that the only noise source is the electric motor. The diesel mechanical plant generates the highest noise level. The noise reduction that can be achieved by using battery plant instead of diesel mechanical is 20 dBA and 5.4 dB for airborne and structure-borne noise respectively. Intuitively, the airborne noise and the structure-borne noise improvement should have a higher value since the battery is compared to heavy equipment such as diesel engines and reduction gears. Nevertheless, when looking into more detail, the electric motor produces quite high airborne noise in the high-frequency region, which makes the log summation over the entire range of octave band frequencies relatively high. It is important to note that the airborne noise calculation is quite conservative for the electric motor. For the structure-borne noise, the lower improvement of battery plant is possibly due to the overestimate value of the electric motor structure-borne noise source level.

The structure-borne noise results of the four transmission types with the resilient mountings are also investigated. This needs to be done in order to show if the noise reductions by changing the transmission types are still valid for different types of mountings. Similar to the hard-mounted results, the battery plant has the lowest structure-borne noise compared to the other transmission types. However, the difference between the battery plant and the diesel mechanical plant become smaller compared to the difference in the hard mounting analysis. The same thing also happens to the diesel-hybrid plant thanks to the low-frequency mounting on the gearbox(s) and electric motor(s). Therefore, by installing the low frequency mounting to all the main noise sources, the diesel-mechanical and the diesel-hybrid will get more structure-borne noise reductions compared to the battery-plant.

6.3 Recommendations

This thesis has been investigated many aspects, but still, many more things can be done This sub-chapter will elaborate more on the recommendations on what can be done in the future works. It can be split into four categories, i.e., further study in the equipment sizing model, further study in the noise model, further study in the evaluation methodology and the applicability of the approach developed in this project to the ship design process. More topics in the noise of the propulsion system that should be investigated are also mentioned here. The following sub-sections will describe the recommendations in more details.

6.3.1 Equipment Sizing Model

- Extend the database of the equipment from different manufacturers. It will create the model becomes more versatile.
- The engine room is assumed to be a simple cubic compartment. Create a more realistic model for the engine room with all the outfits. In this way, the room absorption can be predicted much more accurate.

6.3.2 Noise Model

- Develop the transition region for the diesel engine airborne and structure-borne noise source instead of clustering it by the range of speed. This improvement needs to be done to avoid the big noise level gap between two adjacent engine speeds in a different speed range.
- Include the actual speed as a parameter to calculate the diesel engine airborne noise in every nominal speed region. The SNAME model only uses actual speed as a parameter for an engine with the nominal speed above 700 rpm. It is important to have the actual speed as the parameter in the empirical formula to have a more accurate noise result in the loading point changing analysis.
- The electrical motor and generator only have the nominal speed and nominal power as input parameters to calculate their noise source level. It is important to have the empirical formula as a function of actual speed. In that sense, the noise level of the electric motor and generator in part load condition would be more reliable.
- Develop a better empirical constant for the generator structure-borne noise source levels. The equation of SNAME model gives a very conservative value, which results in overestimation in the structure-borne noise level.
- The transmission path model can be improved by using an analytical model instead of the empirical model. In this way, the calculation will take a bit longer, but the analysis can be extended to the whole ship.
- It is assumed that the battery does not produce airborne noise and structure-borne noise at all. However, not enough research has been performed on this topic in order to fully support this assumption. A further study is needed to investigate the battery airborne and structure-borne noise level

6.3.3 Evaluation Methodology

- The effect of early design choices for the noise of ship propulsion system is determined in this project. However, the effects of these choices on the underwater noise are still unknown. Further analysis to investigate the consequences of early design choices for the propulsion system on the underwater noise needs to be done. There is a big correlation between the structure-borne noise and the underwater noise.

- The electric motor used in this analysis is assumed to be an AC induction motor. Extend the analysis to a larger variety of electric motor needs to be done. There is a potential to reduce the noise by using a different type of electric motor. A study [37] shows that DC motor will give less noise compared to AC motor.
- The prime mover is assumed only the diesel engine in this project. Further investigation for other types of prime movers need to be done. There is a potential to reduce the noise by using a different type of prime mover. An experiment by [45] shows that dual fuel engine gives less noise when operates with LNG compared to operate with liquid marine fuels due to a better combustion performance.
- The structure of the ship is assumed to be the same for every transmission type. It would be interesting to investigate the effect of the structure variations toward the noise of propulsion system.
- Investigate the potential to implement the mounting between different arrangements. There is a chance that the configuration with multiple engines will get more noise reductions compared to the single-engine configuration. It is possible to have a large mounting for all engines in the multiple engines configuration. In the part load condition, where only some of the engines need to operate, the mounting will give a larger reduction in the overall noise level. A further study needs to be done in this field.

6.3.4 Applicability to Ship Design Process

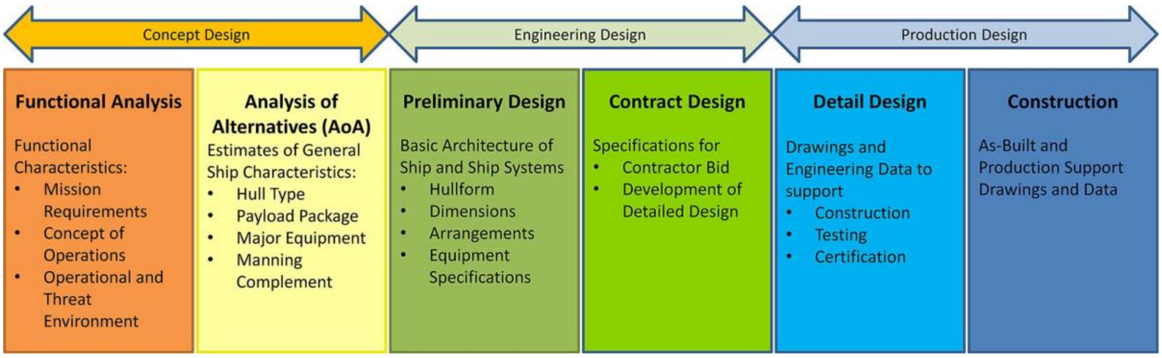


Figure 6.1 Ship Design Stages [7]

The ship design process can be classified into three main stages, namely the concept design, the engineering design and the production design. Based on the figure above, the preliminary design phase is the step in the engineering design where the basic architecture of the ship and the ship systems are established.

The challenging task of the preliminary design phase is that the decisions made at this point will give high impact to many aspects of the ship when the uncertainty is high too. One of the aspects is the noise generated by the propulsion system. This thesis aims to guide the designer in this phase to choose the propulsion system based on the noise considerations. The ship requirements and the early design choices that are used in the evaluation methodology are available options that can be easily modified in the preliminary design phase. It is not feasible to change these choices once the design process arrives at the production design stage.

Perhaps, the ship designer can create a better decision in the preliminary phase when designing the ship propulsion system using the methodology that is developed in this study. Moreover, the information that is provided in this thesis gives clearer acoustical consequences when a decision regarding the propulsion system design is made. In addition, the stricter regulations for the on-board and radiated noise also urge the implementation of this thesis into the ship design process. It is also possible the noise reduction that is achieved by using the early design options will also reduce the propagated noise to the other cabin inside the ship since the early choices reduce the noise level from the sources.

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A

Equipment Database

In this appendix, the database being used in the sizing model are presented. First, the Diesel Engines and Diesel Generator Sets use the MAN and Wartsila engine catalogue. Second, the electric motors use the induction catalogue motor from ABB. Last, the Reduction Gear Database is built by using the gearbox catalogue from RENK.

A.1 Diesel Engine Database

No	Manufacturer	Type	No.of Cyl	Power		alpha	Speed	L	W	H	Db	Ls	Vs	p_me	cm
				[kW]	[hp]		[rpm]	[mm]	[mm]	[mm]	[mm]	[mm]	[m3]	[bar]	[m/s]
1	Wartsila	W 6L32	6	3000	4020	0	750	4980	2305	3715	320	400	0.03	24.87	10
2	Wartsila	W 6L32	6	3480	4663.2	0	750	5130	2380	3450	320	400	0.03	28.85	10
3	Wartsila	W 7L32	7	3500	4690	0	750	5470	2305	3715	320	400	0.03	24.87	10
4	Wartsila	W 8L32	8	4000	5360	0	750	5960	2305	3515	320	400	0.03	24.87	10
5	Wartsila	W 8L32	8	4640	6217.6	0	750	6379	2610	3530	320	400	0.03	28.85	10
6	Wartsila	W 9L32	9	4500	6030	0	750	6450	2305	3515	320	400	0.03	24.87	10
7	Wartsila	W 9L32	9	5220	6994.8	0	750	6869	2610	3530	320	400	0.03	28.85	10

8	MAN	6L32	6	3000	4020	0	750	5940	2630	4010	320	400	0.03	24.9	10
9	MAN	7L32	7	3500	4690	0	750	6470	2630	4010	320	400	0.03	24.9	10
10	MAN	8L32	8	4000	5360	0	750	7000	2715	4490	320	400	0.03	24.9	10
11	MAN	9L32	9	4500	6030	0	750	7530	2715	4490	320	400	0.03	24.9	10
12	MAN	6L51/60	6	6300	8442	0	500	8494	3165	5340	510	600	0.12	20.6	10
13	MAN	7L51/60	7	7350	9849	0	500	9314	3165	5340	510	600	0.12	20.6	10
14	MAN	8L51/60	8	8400	11256	0	500	10134	3165	5340	510	600	0.12	20.6	10
15	MAN	9L51/60	9	9455	12669.7	0	500	11160	3283	5340	510	600	0.12	20.6	10
16	MAN	6L48/60CR	6	7200	9648	0	514	8760	3165	5300	480	600	0.11	25.8	10.28
17	MAN	7L48/60CR	7	8400	11256	0	514	9580	3165	5300	480	600	0.11	25.8	10.28
18	MAN	8L48/60CR	8	9600	12864	0	514	10540	3280	5300	480	600	0.11	25.8	10.28
19	MAN	9L48/60CR	9	10800	14472	0	514	11360	3280	5300	480	600	0.11	25.8	10.28
20	MAN	6L48/60B	6	6900	9246	0	514	8760	3165	5300	480	600	0.11	24.73	10.28
21	MAN	7L48/60B	7	8050	10787	0	514	9580	3165	5300	480	600	0.11	24.73	10.28
22	MAN	8L48/60B	8	9200	12328	0	514	10540	3280	5300	480	600	0.11	24.73	10.28
23	MAN	9L48/60B	9	10350	13869	0	514	11360	3280	5300	480	600	0.11	24.73	10.28
24	MAN	6L35/44DF	6	3180	4261.2	0	750	6485	2539	4163	350	440	0.04	20	11
25	MAN	7L35/44DF	7	3710	4971.4	0	750	7015	2678	4369	350	440	0.04	20	11
26	MAN	8L35/44DF	8	4240	5681.6	0	750	7545	2678	4369	350	440	0.04	20	11
27	MAN	9L35/44DF	9	4770	6391.8	0	750	8075	2678	4369	350	440	0.04	20	11
28	MAN	10L35/44DF	10	5300	7102	0	750	8605	2678	4369	350	440	0.04	20	11

29	MAN	6L35/44DF	6	3060	4100.4	0	720	6485	2539	4163	350	440	0.04	20.1	10.56
30	MAN	7L35/44DF	7	3570	4783.8	0	720	7015	2678	4369	350	440	0.04	20.1	10.56
31	MAN	8L35/44DF	8	4080	5467.2	0	720	7545	2678	4369	350	440	0.04	20.1	10.56
32	MAN	9L35/44DF	9	4590	6150.6	0	720	8075	2678	4369	350	440	0.04	20.1	10.56
33	MAN	10L35/44DF	10	5100	6834	0	720	8605	2678	4369	350	440	0.04	20.1	10.56
34	MAN	6L32/44CR	6	3600	4824	0	750	6312	2174	4163	320	440	0.04	27.1	11
35	MAN	7L32/44CR	7	3920	5252.8	0	750	6924	2359	4369	320	440	0.04	27.1	11
36	MAN	8L32/44CR	8	4800	6432	0	750	7454	2359	4369	320	440	0.04	27.1	11
37	MAN	9L32/44CR	9	5400	7236	0	750	7984	2359	4369	320	440	0.04	27.1	11
38	MAN	10L32/44CR	10	6000	8040	0	750	8603	2359	4369	320	440	0.04	27.1	11
39	MAN	6L32/44CR	6	3600	4824	0	720	6312	2174	4163	320	440	0.04	28.3	10.56
40	MAN	7L32/44CR	7	3920	5252.8	0	720	6924	2359	4369	320	440	0.04	28.3	10.56
41	MAN	8L32/44CR	8	4800	6432	0	720	7454	2359	4369	320	440	0.04	28.3	10.56
42	MAN	9L32/44CR	9	5400	7236	0	720	7984	2359	4369	320	440	0.04	28.3	10.56
43	MAN	10L32/44CR	10	6000	8040	0	720	8603	2359	4369	320	440	0.04	28.3	10.56
44	MAN	6L28/32A	6	1470	1969.8	0	775	5330	1732	3186	280	320	0.02	19.3	8.27
45	MAN	7L28/32A	7	1715	2298.1	0	775	5810	1732	3186	280	320	0.02	19.3	8.27
46	MAN	8L28/32A	8	1960	2626.4	0	775	6290	1732	3186	280	320	0.02	19.3	8.27
47	MAN	9L28/32A	9	2205	2954.7	0	775	6770	1844	3242	280	320	0.02	19.3	8.27
48	MAN	6L27/38	6	2040	2733.6	0	800	5070	2035	3555	280	380	0.02	23.5	10.13
49	MAN	7L27/38	7	2380	3189.2	0	800	5515	2035	3687	270	380	0.02	23.5	10.13

50	MAN	8L27/38	8	2720	3644.8	0	800	5960	2035	3687	270	380	0.02	23.5	10.13
51	MAN	9L27/38	9	3060	4100.4	0	800	6405	2035	3687	270	380	0.02	23.5	10.13
52	MAN	6L23/30A	6	960	1286.4	0	900	3737	1660	2467	230	300	0.01	17.1	9
53	MAN	8L23/30A	8	1280	1715.2	0	900	4477	1660	2467	230	300	0.01	17.1	9
54	MAN	6L21/31	6	1290	1728.6	0	1000	4544	1695	3113	210	310	0.01	24	10.33
55	MAN	7L21/31	7	1505	2016.7	0	1000	4899	1695	3267	210	310	0.01	24	10.33
56	MAN	8L21/31	8	1720	2304.8	0	1000	5254	1820	3267	210	310	0.01	24	10.33
57	MAN	9L21/31	9	1935	2592.9	0	1000	5609	1820	3267	210	310	0.01	24	10.33
58	Wartsila	W 6L38	6	4350	5829	0	600	6345	2244	3945	380	475	0.05	26.92	9.5
59	Wartsila	W 8L38	8	5800	7772	0	600	7961	2209	4112	380	475	0.05	26.92	9.5
60	Wartsila	W 9L38	9	6525	8743.5	0	600	8561	2209	4112	380	475	0.05	26.92	9.5
61	Wartsila	W 6L46F	6	7200	9648	0	600	8470	2905	4930	460	580	0.1	24.9	11.6
62	Wartsila	W 7L46F	7	8400	11256	0	600	9435	3130	5230	460	580	0.1	24.9	11.6
63	Wartsila	W 8L46F	8	9600	12864	0	600	10255	3130	5230	460	580	0.1	24.9	11.6
64	Wartsila	W 9L46F	9	10800	14472	0	600	11075	3130	5230	460	580	0.1	24.9	11.6
65	MAN	12V32	12	6000	8040	45	750	6915	3140	4100	320	400	0.03	24.9	10
66	MAN	14V32	14	7000	9380	45	750	7545	3140	4100	320	400	0.03	24.9	10
67	MAN	16V32	16	8000	10720	45	750	8365	3730	4420	320	400	0.03	24.9	10
68	MAN	18V32	18	9000	12060	45	750	8995	3730	4420	320	400	0.03	24.9	10
69	Wartsila	W 12V32	12	6000	8040	45	750	6935	3020	4190	320	400	0.03	24.87	10
70	Wartsila	W 12V32	12	6960	9326.4	45	750	6865	2900	3640	320	400	0.03	28.85	10

71	Wartsila	W 16V32	16	8000	10720	45	750	8060	3020	3955	320	400	0.03	24.87	10
72	Wartsila	W 16V32	16	9280	12435.2	45	750	7905	3325	3805	320	400	0.03	28.85	10
73	Wartsila	W 18V32	18	9000	12060	45	750	8620	3020	3955	320	400	0.03	24.87	10
74	MAN	12V32/44CR	12	7200	9648	45	750	7195	3100	4039	320	440	0.04	27.13	11
75	MAN	14V32/44CR	14	7840	10505.6	45	750	7970	3100	4262	320	440	0.04	25.32	11
76	MAN	16V32/44CR	16	9600	12864	45	750	8600	3100	4262	320	440	0.04	27.13	11
77	MAN	18V32/44CR	18	10800	14472	45	750	9230	3100	4262	320	440	0.04	27.13	11
78	MAN	20V32/44CR	20	12000	16080	45	750	9860	3100	4262	320	440	0.04	27.13	11
79	MAN	12V32/44CR	12	7200	9648	45	720	7195	3100	4039	320	440	0.04	28.26	10.56
80	MAN	14V32/44CR	14	7840	10505.6	45	720	7970	3100	4262	320	440	0.04	26.38	10.56
81	MAN	16V32/44CR	16	9600	12864	45	720	8600	3100	4262	320	440	0.04	28.26	10.56
82	MAN	18V32/44CR	18	10800	14472	45	720	9230	3100	4262	320	440	0.04	28.26	10.56
83	MAN	20V32/44CR	20	12000	16080	45	720	9860	3100	4262	320	440	0.04	28.26	10.56
84	MAN	12V28/33D STC	12	5460	7316.4	45	1000	6207	2473	3734	280	330	0.02	26.87	11
85	MAN	16V28/33D STC	16	7280	9755.2	45	1000	7127	2473	3734	280	330	0.02	26.87	11
86	MAN	20V28/33D STC	20	9100	12194	45	1000	8047	2473	3734	280	330	0.02	26.87	11
87	Wartsila	W 12V38	12	8700	11658	45	600	7461	3030	4516	380	475	0.05	26.92	9.5
88	Wartsila	W 16V38	16	11600	15544	45	600	9018	3030	4717	380	475	0.05	26.92	9.5
89	MAN	12V51/60	12	11700	15678	50	500	10254	4713	5517	510	600	0.12	19.09	10
90	MAN	14V51/60	14	13650	18291	50	500	11254	4713	5517	510	600	0.12	19.09	10
91	MAN	16V51/60	16	15600	20904	50	500	12254	4713	5517	510	600	0.12	19.09	10

92	MAN	18V51/60	18	17550	23517	50	500	13644	4713	5517	510	600	0.12	19.09	10
93	MAN	12V48/60	12	14400	19296	50	514	10790	4730	5550	480	600	0.11	25.8	10.28
94	MAN	14V48/60	14	16800	22512	50	514	11790	4730	5550	480	600	0.11	25.8	10.28
95	MAN	16V48/60	16	19200	25728	50	514	13140	4730	5550	480	600	0.11	25.8	10.28
96	MAN	18V48/60	18	21600	28944	50	514	14140	4730	5550	480	600	0.11	25.8	10.28
97	MAN	12V48/60B	12	13800	18492	50	514	10790	4730	5500	480	600	0.11	24.73	10.28
98	MAN	14V48/60 B	14	16100	21574	50	514	11790	4730	5500	480	600	0.11	24.73	10.28
99	MAN	16V48/60 B	16	18400	24656	50	514	13140	4730	5500	480	600	0.11	24.73	10.28
100	MAN	18V48/60 B	18	20700	27738	50	514	14140	4730	5500	480	600	0.11	24.73	10.28
101	Wartsila	W 12V46F	12	14400	19296	50	600	10945	4040	5385	460	580	0.1	24.9	11.6
102	Wartsila	W 14V46F	14	16800	22512	50	600	11728	4678	5854	460	580	0.1	24.9	11.6
103	Wartsila	W 16V46F	16	19200	25728	50	600	12871	4678	5854	460	580	0.1	24.9	11.6
104	Wartsila	W 8V31	8	4880	6539.2	50	750	6175	3113	4701	310	430	0.03	30.07	10.75
105	Wartsila	W 10V31	10	6100	8174	50	750	6813	3113	4701	310	430	0.03	30.07	10.75
106	Wartsila	W 12V31	12	7320	9808.8	50	750	7900	3500	4124	310	430	0.03	30.07	10.75
107	Wartsila	W 14V31	14	8540	11443.6	50	750	8540	3500	4124	310	430	0.03	30.07	10.75
108	Wartsila	W 16V31	16	9760	13078.4	50	750	9130	3500	4124	310	430	0.03	30.07	10.75
109	Wartsila	W 12V26	12	3900	5226	55	900	5442	2552	2860	260	320	0.02	25.51	9.6
110	Wartsila	W 16V26	16	5200	6968	55	900	6223	2552	2860	260	320	0.02	25.51	9.6
111	MAN	G95ME-C9-5	5	34350	46029	0	80	11468	5380	17585	950	3460	2.45	21.01	9.23
112	MAN	G95ME-C9-6	6	41220	55234.8	0	80	13042	5380	17585	950	3460	2.45	21.01	9.23

113	MAN	G95ME-C9-7	7	48090	64440.6	0	80	14616	5380	17585	950	3460	2.45	21.01	9.23
114	MAN	G95ME-C9-8	8	54960	73646.4	0	80	16190	5380	17585	950	3460	2.45	21.01	9.23
115	MAN	G95ME-C9-9	9	61830	82852.2	0	80	17804	5380	17585	950	3460	2.45	21.01	9.23
116	MAN	G95ME-C9-10	10	68700	92058	0	80	19779	5380	17585	950	3460	2.45	21.01	9.23
117	MAN	G95ME-C9-11	11	75570	101263.8	0	80	21489	5380	17585	950	3460	2.45	21.01	9.23
118	MAN	G95ME-C9-12	12	82440	110469.6	0	80	23159	5380	17585	950	3460	2.45	21.01	9.23
119	MAN	G90ME-C9-5	5	31200	41808	0	84	9920	5034	15860	900	3260	2.07	21.49	9.13
120	MAN	G90ME-C9-6	6	37440	50169.6	0	84	11410	5034	15860	900	3260	2.07	21.49	9.13
121	MAN	G90ME-C9-7	7	43680	58531.2	0	84	12900	5034	15860	900	3260	2.07	21.49	9.13
122	MAN	G90ME-C9-8	8	49920	66892.8	0	84	14390	5034	15860	900	3260	2.07	21.49	9.13
123	MAN	G90ME-C9-9	9	56160	75254.4	0	84	25880	5034	15860	900	3260	2.07	21.49	9.13
124	MAN	G90ME-C9-10	10	62400	83616	0	84	18040	5034	15860	900	3260	2.07	21.49	9.13
125	MAN	G90ME-C9-11	11	68640	91977.6	0	84	19530	5034	15860	900	3260	2.07	21.49	9.13
126	MAN	G90ME-C9-12	12	74880	100339.2	0	84	21020	5034	15860	900	3260	2.07	21.49	9.13
127	MAN	S90ME-C10-5	5	30500	40870	0	84	10312	5450	16775	900	3260	2.07	21.01	9.13
128	MAN	S90ME-C10-6	6	36600	49044	0	84	11902	5450	16775	900	3260	2.07	21.01	9.13
129	MAN	S90ME-C10-7	7	42700	57218	0	84	13492	5450	16775	900	3260	2.07	21.01	9.13
130	MAN	S90ME-C10-8	8	48800	65392	0	84	16135	5450	16775	900	3260	2.07	21.01	9.13
131	MAN	S90ME-C10-9	9	54900	73566	0	84	17725	5450	16775	900	3260	2.07	21.01	9.13
132	MAN	S90ME-C10-10	10	61000	81740	0	84	19315	5450	16775	900	3260	2.07	21.01	9.13
133	MAN	S90ME-C10-11	11	67100	89914	0	84	20905	5450	16775	900	3260	2.07	21.01	9.13

134	MAN	S90ME-C10-12	12	73200	98088	0	84	22495	5450	16775	900	3260	2.07	21.01	9.13
135	MAN	G80ME-C9-6	6	28260	37868.4	0	72	10735	5680	17785	800	3720	1.87	20.99	8.93
136	MAN	G80ME-C9-7	7	32970	44179.8	0	72	12135	5680	17785	800	3720	1.87	20.99	8.93
137	MAN	G80ME-C9-8	8	37680	50491.2	0	72	13535	5680	17785	800	3720	1.87	20.99	8.93
138	MAN	G80ME-C9-9	9	42390	56802.6	0	72	15880	5680	17785	800	3720	1.87	20.99	8.93
139	MAN	S80ME-C9-6	6	27060	36260.4	0	78	10100	5374	15390	800	3450	1.73	20.01	8.97
140	MAN	S80ME-C9-7	7	31570	42303.8	0	78	11434	5374	15390	800	3450	1.73	20.01	8.97
141	MAN	S80ME-C9-8	8	36080	48347.2	0	78	12768	5374	15390	800	3450	1.73	20.01	8.97
142	MAN	S80ME-C9-9	9	40590	54390.6	0	78	14102	5374	15390	800	3450	1.73	20.01	8.97
143	MAN	G70ME-C9-5	5	18200	24388	0	83	8486	4900	14550	700	3256	1.25	21	9.01
144	MAN	G70ME-C9-6	6	21840	29265.6	0	83	9596	4900	14550	700	3256	1.25	21	9.01
145	MAN	G70ME-C9-7	7	25480	34143.2	0	83	10856	4900	14550	700	3256	1.25	21	9.01
146	MAN	G70ME-C9-8	8	29120	39020.8	0	83	12116	4900	14550	700	3256	1.25	21	9.01
147	MAN	S70ME-C10-5	5	17150	22981	0	91	7464	4122	13270	700	2800	1.08	20.99	8.49
148	MAN	S70ME-C10-6	6	20580	27577.2	0	91	8562	4122	13270	700	2800	1.08	20.99	8.49
149	MAN	S70ME-C10-7	7	24010	32173.4	0	91	9660	4122	13270	700	2800	1.08	20.99	8.49
150	MAN	S70ME-C10-8	8	27440	36769.6	0	91	10758	4122	13270	700	2800	1.08	20.99	8.49
151	MAN	S70ME-C8-5	5	16350	21909	0	91	7781	4454	13021	700	2800	1.08	20.01	8.49
152	MAN	S70ME-C8-6	6	19620	26290.8	0	91	8971	4454	13021	700	2800	1.08	20.01	8.49
153	MAN	S70ME-C8-7	7	22890	30672.6	0	91	10161	4454	13021	700	2800	1.08	20.01	8.49
154	MAN	S70ME-C8-8	8	26160	35054.4	0	91	11351	4454	13021	700	2800	1.08	20.01	8.49

155	MAN	S65ME-C8-5	5	14350	19229	0	95	7148	4170	12435	650	2730	0.91	20.01	8.65
156	MAN	S65ME-C8-6	6	17220	23074.8	0	95	8232	4170	12435	650	2730	0.91	20.01	8.65
157	MAN	S65ME-C8-7	7	20090	26920.6	0	95	9316	4170	12435	650	2730	0.91	20.01	8.65
158	MAN	S65ME-C8-8	8	22960	30766.4	0	95	10400	4170	12435	650	2730	0.91	20.01	8.65
159	MAN	G60ME-C9-5	5	13400	17956	0	97	7390	4220	12575	600	2790	0.79	21.01	9.02
160	MAN	G60ME-C9-6	6	16080	21547.2	0	97	8470	4220	12575	600	2790	0.79	21.01	9.02
161	MAN	G60ME-C9-7	7	18760	25138.4	0	97	9550	4220	12575	600	2790	0.79	21.01	9.02
162	MAN	G60ME-C9-8	8	21440	28729.6	0	97	10630	4220	12575	600	2790	0.79	21.01	9.02
163	MAN	S60ME-C8-5	5	11900	15946	0	105	6668	3840	11075	600	2400	0.68	20.04	8.4
164	MAN	S60ME-C8-6	6	14280	19135.2	0	105	7688	3840	11075	600	2400	0.68	20.04	8.4
165	MAN	S60ME-C8-7	7	16660	22324.4	0	105	8708	3840	11075	600	2400	0.68	20.04	8.4
166	MAN	S60ME-C8-8	8	19040	25513.6	0	105	9728	3840	11075	600	2400	0.68	20.04	8.4
167	MAN	G50ME-C9-5	5	8600	11524	0	100	6260	3652	10980	500	2500	0.49	21.02	8.33
168	MAN	G50ME-C9-6	6	10320	13828.8	0	100	7132	3652	10980	500	2500	0.49	21.02	8.33
169	MAN	G50ME-C9-7	7	12040	16133.6	0	100	8004	3652	10980	500	2500	0.49	21.02	8.33
170	MAN	G50ME-C9-8	8	13760	18438.4	0	100	8876	3652	10980	500	2500	0.49	21.02	8.33
171	MAN	G50ME-C9-9	9	15480	20743.2	0	100	9748	3652	10980	500	2500	0.49	21.02	8.33
172	MAN	S50ME-C9-5	5	8900	11926	0	117	6073	3290	10090	500	2214	0.43	21	8.63
173	MAN	S50ME-C9-6	6	10680	14311.2	0	117	6948	3290	10090	500	2214	0.43	21	8.63
174	MAN	S50ME-C9-7	7	12460	16696.4	0	117	7823	3290	10090	500	2214	0.43	21	8.63
175	MAN	S50ME-C9-8	8	14240	19081.6	0	117	8698	3290	10090	500	2214	0.43	21	8.63

176	MAN	S50ME-C9-9	9	16020	21466.8	0	117	9573	3290	10090	500	2214	0.43	21	8.63
177	MAN	S50ME-C8-5	5	8300	11122	0	127	5924	3150	9335	500	2000	0.39	19.97	8.47
178	MAN	S50ME-C8-6	6	9960	13346.4	0	127	6774	3150	9335	500	2000	0.39	19.97	8.47
179	MAN	S50ME-C8-7	7	11620	15570.8	0	127	7624	3150	9335	500	2000	0.39	19.97	8.47
180	MAN	S50ME-C8-8	8	13280	17795.2	0	127	8474	3150	9335	500	2000	0.39	19.97	8.47
181	MAN	S50ME-C8-9	9	14940	20019.6	0	127	9324	3150	9335	500	2000	0.39	19.97	8.47

Table A.0.1 Diesel Engine Database

A.2 Gearbox Database :

No	Type	M_input	Lgb	Wgb	Hgb	M_prop	i	Pinput	Speed
1		133.76	1.14	1.99	3.15	381.23	2.85	7200	514
2		156.06	1.2	2.11	3.25	444.77	2.85	8400	514
3		178.35	1.2	2.11	3.25	508.3	2.85	9600	514
4		200.65	1.27	2.22	3.35	571.84	2.85	10800	514
5		267.53	1.94	2.46	3.95	762.46	2.85	14400	514
6		312.12	2.03	2.56	3.4	889.53	2.85	16800	514
7		356.71	2.1	2.72	3.5	1016.61	2.85	19200	514
8		401.29	2.19	2.88	3.65	1143.69	2.85	21600	514
9		267.53	2.1	2.72	3.5	981.83	3.67	14400	514
10		312.12	2.19	2.88	3.65	1145.47	3.67	16800	514
11		356.71	2.33	3.02	3.75	1309.11	3.67	19200	514
12		401.29	2.33	3.02	3.75	1472.75	3.67	21600	514
13		133.76	1.2	2.11	3.25	490.92	3.67	7200	514
14		156.06	1.27	2.22	3.35	572.73	3.67	8400	514
15		178.35	1.37	2.32	3.8	654.55	3.67	9600	514
16		200.65	1.37	2.32	3.8	736.37	3.67	10800	514
17		76.39	1.07	1.91	3	286.48	3.75	6000	750
18		89.13	1.07	1.91	3	334.23	3.75	7000	750
19		101.86	1.14	1.99	3.15	381.97	3.75	8000	750
20		114.59	1.14	1.99	3.15	429.72	3.75	9000	750
21		127.32	1.2	2.11	3.25	477.46	3.75	10000	750
22		38.2	0.85	1.51	2.4	143.24	3.75	3000	750
23		44.56	0.85	1.51	2.4	167.11	3.75	3500	750
24		50.93	0.9	1.59	2.65	190.99	3.75	4000	750
25		57.3	0.9	1.59	2.65	214.86	3.75	4500	750
26		63.66	0.95	1.7	2.75	238.73	3.75	5000	750
27		76.39	1.14	1.99	3.15	358.1	4.6875	6000	750
28		89.13	1.2	2.11	3.25	417.78	4.6875	7000	750
29		101.86	1.27	2.22	3.35	477.46	4.6875	8000	750
30		114.59	1.27	2.22	3.35	537.15	4.6875	9000	750
31		127.32	1.37	2.32	3.65	596.83	4.6875	10000	750

32		38.2	0.9	1.59	2.65	179.05	4.6875	3000	750
33		44.56	0.95	1.7	2.75	208.89	4.6875	3500	750
34		50.93	1.02	1.78	2.85	238.73	4.6875	4000	750
35		57.3	1.07	1.91	3	268.57	4.6875	4500	750
36									
37		133.76	2.03	2.56	3.4	687.55	5.14	7200	514
38		156.06	2.1	2.72	3.5	802.14	5.14	8400	514
39		178.35	2.19	2.88	3.65	916.73	5.14	9600	514
40		200.65	2.19	2.88	3.65	1031.32	5.14	10800	514
41		76.39	1.94	2.46	3.95	477.46	6.25	6000	750
42		89.13	2.03	2.56	3.4	557.04	6.25	7000	750
43		101.86	2.1	2.72	3.5	636.62	6.25	8000	750
44		114.59	2.19	2.88	3.65	716.2	6.25	9000	750
45		127.32	2.19	2.88	3.8	795.77	6.25	10000	750
46		38.2	1.14	1.99	3.15	238.73	6.25	3000	750
47		44.56	1.14	1.99	3.15	278.52	6.25	3500	750
48		50.93	1.2	2.11	3.25	318.31	6.25	4000	750
49		57.3	1.27	2.22	3.35	358.1	6.25	4500	750
50		63.66	1.27	2.22	3.35	397.89	6.25	5000	750
51		267.53	2.44	3.18	3.95	1375.1	5.14	14400	514

Table A.2 Gearbox Database

A.3 Low Voltage Electric Motor Database:

No	P_output [kW]	Motor Type	rpm	Power Factor	Rotor Inertia	Motor Weight	SPL	Dimensions					Dr[mm]	Lr [mm]	vt [m/s]	tau_EM [kN/m2]
				100% Load	kgm^2	kg	dB(A)	Ls	Ds	Lem	Wem	Hem				
1	355	HXR400LB4	1492	0.84	10.3	2540	79	1120	710	1900	975	1185	355	1120	27.73	10.25
2	400	HXR400LC4	1491	0.86	11.4	2660	79	1120	710	1900	975	1185	355	1120	27.71	11.55
3	450	HXR400LD4	1491	0.86	12.5	2800	79	1120	710	1900	975	1185	355	1120	27.71	13.00
4	500	HXR400LF4	1491	0.88	14.7	3060	79	1120	710	1900	975	1185	355	1120	27.71	14.45
5	560	HXR400LH4	1493	0.87	16.9	3330	79	1120	710	1900	975	1185	355	1120	27.75	16.16
6	630	HXR450LG4	1493	0.89	26.8	4220	80	1300	800	2120	1025	1290	400	1300	31.27	12.33
7	710	HXR450LH4	1494	0.88	28.6	4390	80	1300	800	2120	1025	1290	400	1300	31.29	13.89
8	800	HXR500LG4	1494	0.88	44.1	5840	82	1570	900	2455	1265	1530	450	1570	35.20	10.24
9	900	HXR500LH4	1493	0.89	46.9	6040	82	1570	900	2455	1265	1530	450	1570	35.18	11.53
10	1000	HXR500LJ4	1495	0.87	49.8	6220	82	1570	900	2455	1265	1530	450	1570	35.23	12.79
11	1120	HXR500LN4	1493	0.9	61.1	7010	82	1570	900	2455	1265	1530	450	1570	35.18	14.35
12	1250	HXR560LJ4	1493	0.89	83.3	9290	84	1940	1000	3025	1340	1675	500	1940	39.09	10.49
13	1400	HXR560LM4	1495	0.88	98	10060	84	1940	1000	3025	1340	1675	500	1940	39.14	11.74
14	1600	HXR560LT4	1496	0.89	142.3	12170	84	1940	1000	3025	1340	1675	500	1940	39.17	13.41
15	315	HXR400LC6	994	0.78	16.9	2610	80	1120	710	1900	975	1185	355	1120	18.48	13.65
16	355	HXR400LD6	994	0.79	18.6	2740	80	1120	710	1900	975	1185	355	1120	18.48	15.38
17	400	HXR400LF6	994	0.81	21.9	3000	80	1120	710	1900	975	1185	355	1120	18.48	17.33
18	450	HXR400LH6	995	0.82	25.3	3260	80	1120	710	1900	975	1185	355	1120	18.49	19.49
19	500	HXR450LF6	995	0.83	35.4	4030	82	1300	800	2120	1025	1290	400	1300	20.84	14.69
20	560	HXR450LG6	995	0.83	38.1	4200	82	1300	800	2120	1025	1290	400	1300	20.84	16.44
21	630	HXR450LJ6	995	0.85	43.5	4530	82	1300	800	2120	1025	1290	400	1300	20.84	18.50
22	710	HXR450LL6	995	0.85	48.9	4860	82	1300	800	2120	1025	1290	400	1300	20.84	20.85
23	800	HXR500LG6	994	0.84	57.8	5750	84	1570	900	2455	1265	1530	450	1570	23.42	15.38
24	900	HXR500LK6	996	0.83	69.4	6320	84	1570	900	2455	1265	1530	450	1570	23.47	17.28
25	1000	HXR500LN6	996	0.83	81.1	6920	84	1570	900	2455	1265	1530	450	1570	23.47	19.20

26	1120	HXR560LK6	995	0.86	120.1	9370	85	1940	1000	3025	1340	1675	500	1940	26.05	14.10
27	1250	HXR560LL6	996	0.85	127	9630	85	1940	1000	3025	1340	1675	500	1940	26.08	15.73
28	1400	HXR560LQ6	997	0.85	154.5	10640	85	1940	1000	3025	1340	1675	500	1940	26.10	17.61
29	1600	HXR560LU6	997	0.85	202.6	12370	85	1940	1000	3025	1340	1675	500	1940	26.10	20.12
30	224	HXR400LC8	742	0.75	16.5	2610	82	1120	710	1900	975	1185	355	1120	13.79	13.00
31	250	HXR400LE8	743	0.76	19.8	2870	82	1120	710	1900	975	1185	355	1120	13.81	14.50
32	280	HXR400LF8	743	0.76	21.5	3000	82	1120	710	1900	975	1185	355	1120	13.81	16.23
33	315	HXR400LH8	743	0.78	24.7	3250	82	1120	710	1900	975	1185	355	1120	13.81	18.27
34	355	HXR450LD8	746	0.75	30.4	3670	83	1300	800	2120	1025	1290	400	1300	15.62	13.91
35	400	HXR450LE8	746	0.75	33.1	3840	83	1300	800	2120	1025	1290	400	1300	15.62	15.68
36	450	HXR450LF8	745	0.78	35.8	3980	83	1300	800	2120	1025	1290	400	1300	15.60	17.64
37	500	HXR450LH8	746	0.79	41.2	4310	83	1570	800	2120	1025	1290	400	1570	15.62	16.23
38	560	HXR450LL8	746	0.8	49.4	4800	83	1570	800	2120	1025	1290	400	1570	15.62	18.18
39	630	HXR500LJ8	746	0.8	66.6	6110	84	1570	900	2455	1265	1530	450	1570	17.58	16.15
40	710	HXR500LK8	746	0.81	70.5	6310	84	1570	900	2455	1265	1530	450	1570	17.58	18.21
41	800	HXR500LL8	746	0.8	74.5	6510	84	1570	900	2455	1265	1530	450	1570	17.58	20.51
42	900	HXR500LM8	746	0.8	78.4	6710	84	1570	900	2455	1265	1530	450	1570	17.58	23.07
43	1000	HXR500LQ8	746	0.81	90.3	7310	84	1570	900	2455	1265	1530	450	1570	17.58	25.64
44	1120	HXR560LN8	747	0.8	143.2	10110	85	1940	1000	3025	1340	1675	500	1940	19.56	18.81
45	1250	HXR560LQ8	747	0.79	157.2	10550	85	1940	1000	3025	1340	1675	500	1940	19.56	20.98
46	1400	HXR560LT8	746	0.82	199.2	12090	85	1940	1000	3025	1340	1675	500	1940	19.53	23.51
47	630	AMA400L4A	1487	0.85	16.3	3150	82	1450	750	1865	1470	1710	375	1450	29.20	12.63
48	710	AMA400L4A	1485	0.88	17.3	3240	82	1450	750	1865	1470	1710	375	1450	29.16	14.26
49	800	AMA400L4A	1485	0.88	18.3	3350	82	1450	750	1865	1470	1710	375	1450	29.16	16.06
50	900	AMA400L4A	1486	0.88	19.3	3460	82	1450	750	1865	1470	1710	375	1450	29.18	18.06
51	1000	AMA400L4A	1484	0.89	20.3	3560	82	1450	750	1865	1470	1710	375	1450	29.14	20.10
52	1120	AMA450L4A	1487	0.86	29.5	4130	83	1600	850	2025	1570	1860	425	1600	33.09	15.84
53	1250	AMA450L4A	1488	0.87	32.9	4390	83	1600	850	2025	1570	1860	425	1600	33.11	17.67
54	1400	AMA450L4A	1489	0.87	36.4	4630	83	1600	850	2025	1570	1860	425	1600	33.13	19.78
55	1600	AMA500L4A	1490	0.89	55	5620	84	1800	950	2305	1785	2060	475	1800	37.06	16.07
56	1800	AMA500L4A	1489	0.9	57.9	5780	84	1800	950	2305	1785	2060	475	1800	37.03	18.10
57	2000	AMA500L4A	1491	0.91	66.7	6270	84	1800	950	2305	1785	2060	475	1800	37.08	20.08

58	2240	AMA500L4A	1490	0.91	66.7	6270	84	1800	950	2305	1785	2060	475	1800	37.06	22.51
59	450	AMA400L6A	987	0.84	16.8	2970	82	1450	750	1865	1470	1710	375	1450	19.38	13.60
60	500	AMA400L6A	987	0.84	18	3070	82	1450	750	1865	1470	1710	375	1450	19.38	15.10
61	560	AMA400L6A	987	0.85	20.4	3270	82	1450	750	1865	1470	1710	375	1450	19.38	16.91
62	630	AMA400L6A	988	0.84	21.6	3340	82	1450	750	1865	1470	1710	375	1450	19.40	19.00
63	710	AMA400L6A	989	0.84	24	3540	82	1450	750	1865	1470	1710	375	1450	19.42	21.39
64	800	AMA450L6A	989	0.87	34.6	4020	82	1600	850	2025	1570	1860	425	1600	22.01	17.02
65	900	AMA450L6A	989	0.87	36.8	4140	82	1600	850	2025	1570	1860	425	1600	22.01	19.14
66	1000	AMA450L6A	990	0.87	39	4270	82	1600	850	2025	1570	1860	425	1600	22.03	21.26
67	1120	AMA450L6A	990	0.87	43.3	4520	82	1600	850	2025	1570	1860	425	1600	22.03	23.80
68	1250	AMA500L6A	991	0.88	66.8	5600	82	1800	950	2305	1785	2060	475	1800	24.65	18.89
69	1400	AMA500L6A	992	0.87	70.4	5750	82	1800	950	2305	1785	2060	475	1800	24.67	21.14
70	1600	AMA500L6A	990	0.89	73.9	5900	82	1800	950	2305	1785	2060	475	1800	24.62	24.20
71	1800	AMA500L6A	991	0.88	80.9	6180	82	1800	950	2305	1785	2060	475	1800	24.65	27.20
72	280	AMA400L8A	739	0.8	18.5	2940	81	1450	750	1865	1470	1710	375	1450	14.51	11.30
73	315	AMA400L8A	738	0.81	19.8	3040	81	1450	750	1865	1470	1710	375	1450	14.49	12.72
74	355	AMA400L8A	740	0.8	22.6	3230	81	1450	750	1865	1470	1710	375	1450	14.53	14.31
75	400	AMA400L8A	740	0.8	23.9	3330	81	1450	750	1865	1470	1710	375	1450	14.53	16.11
76	450	AMA400L8A	739	0.81	25.3	3400	81	1450	750	1865	1470	1710	375	1450	14.51	18.15
77	500	AMA400L8A	740	0.81	28	3600	81	1450	750	1865	1470	1710	375	1450	14.53	20.15
78	560	AMA450L8A	741	0.83	39.3	3980	81	1600	850	2025	1570	1860	425	1600	16.49	15.89
79	630	AMA450L8A	742	0.82	41.9	4110	81	1600	850	2025	1570	1860	425	1600	16.51	17.86
80	710	AMA450L8A	741	0.83	44.5	4210	81	1600	850	2025	1570	1860	425	1600	16.49	20.14
81	800	AMA450L8A	742	0.83	49.7	4460	81	1600	850	2025	1570	1860	425	1600	16.51	22.68
82	900	AMA500L8A	743	0.84	74	5390	81	1800	950	2305	1785	2060	475	1800	18.48	18.13
83	1000	AMA500L8A	743	0.84	78.4	5550	81	1800	950	2305	1785	2060	475	1800	18.48	20.14
84	1120	AMA500L8A	743	0.83	82.7	5700	81	1800	950	2305	1785	2060	475	1800	18.48	22.56
85	1250	AMA500L8A	744	0.84	100.1	6320	81	1800	950	2305	1785	2060	475	1800	18.50	25.15
86	710	AMA400L4L	1486	0.86	14.8	2980	80	1450	750	1865	1470	1710	375	1450	29.18	14.25
87	800	AMA400L4L	1485	0.88	16.8	3180	80	1450	750	1865	1470	1710	375	1450	29.16	16.06
88	900	AMA400L4L	1486	0.88	17.8	3300	80	1450	750	1865	1470	1710	375	1450	29.18	18.06
89	1000	AMA400L4L	1484	0.89	18.8	3400	80	1450	750	1865	1470	1710	375	1450	29.14	20.10

90	1120	AMA450L4L	1487	0.86	27.4	3930	80	1600	850	2025	1570	1860	425	1600	33.09	15.84
91	1250	AMA450L4L	1488	0.87	30.8	4190	80	1600	850	2025	1570	1860	425	1600	33.11	17.67
92	1400	AMA450L4L	1489	0.87	34.3	4420	80	1600	850	2025	1570	1860	425	1600	33.13	19.77
93	1600	AMA500L4L	1491	0.89	52.6	5380	80	1800	950	2305	1785	2060	475	1800	37.08	16.07
94	1800	AMA500L4L	1489	0.9	55.5	5540	80	1800	950	2305	1785	2060	475	1800	37.03	18.10
95	2000	AMA500L4L	1491	0.91	64.3	6030	80	1800	950	2305	1785	2060	475	1800	37.08	20.08
96	2240	AMA500L4L	1490	0.91	64.3	6030	80	1800	950	2305	1785	2060	475	1800	37.06	22.51
97	500	AMA400L6L	987	0.84	16.5	2910	80	1450	750	1865	1470	1710	375	1450	19.38	15.10
98	560	AMA400L6L	987	0.85	18.9	3100	80	1450	750	1865	1470	1710	375	1450	19.38	16.91
99	630	AMA400L6L	988	0.84	20.1	3170	80	1450	750	1865	1470	1710	375	1450	19.40	19.00
100	710	AMA400L6L	988	0.85	21.3	3270	80	1450	750	1865	1470	1710	375	1450	19.40	21.42
101	800	AMA400L6L	988	0.84	22.5	3380	80	1450	750	1865	1470	1710	375	1450	19.40	24.14
102	900	AMA450L6L	989	0.87	34.6	3940	80	1600	850	2025	1570	1860	425	1600	22.01	19.14
103	1000	AMA450L6L	990	0.87	36.8	4070	80	1600	850	2025	1570	1860	425	1600	22.03	21.26
104	1250	AMA450L6L	988	0.88	43.3	4440	80	1600	850	2025	1570	1860	425	1600	21.99	26.62
105	1328	AMA450L6L	988	0.88	41.1	4310	80	1600	850	2025	1570	1860	425	1600	21.99	28.29
106	1400	AMA500L6L	989	0.88	63.4	5350	80	1800	950	2305	1785	2060	475	1800	24.60	21.18
107	1600	AMA500L6L	990	0.88	67	5500	80	1800	950	2305	1785	2060	475	1800	24.62	24.19
108	1800	AMA500L6L	991	0.88	77.5	5940	80	1800	950	2305	1785	2060	475	1800	24.65	27.20
109	2000	AMA500L6L	990	0.88	77.5	5940	80	1800	950	2305	1785	2060	475	1800	24.62	30.25
110	315	AMA400L8L	738	0.81	18.3	2870	80	1450	750	1865	1470	1710	375	1450	14.49	12.72
111	355	AMA400L8L	740	0.8	21.1	3070	80	1450	750	1865	1470	1710	375	1450	14.53	14.31
112	400	AMA400L8L	740	0.8	22.4	3160	80	1450	750	1865	1470	1710	375	1450	14.53	16.11
113	450	AMA400L8L	740	0.81	25.2	3330	80	1450	750	1865	1470	1710	375	1450	14.53	18.13
114	500	AMA400L8L	740	0.81	26.5	3430	80	1450	750	1865	1470	1710	375	1450	14.53	20.15
115	560	AMA450L8L	741	0.83	37.1	3780	80	1600	850	2025	1570	1860	425	1600	16.49	15.89
116	630	AMA450L8L	742	0.82	39.7	3910	80	1600	850	2025	1570	1860	425	1600	16.51	17.86
117	710	AMA450L8L	741	0.83	42.3	4000	80	1600	850	2025	1570	1860	425	1600	16.49	20.14
118	800	AMA450L8L	741	0.84	44.9	4130	80	1600	850	2025	1570	1860	425	1600	16.49	22.71
119	900	AMA450L8L	742	0.83	50.1	4380	80	1600	850	2025	1570	1860	425	1600	16.51	25.52
120	1000	AMA500L8L	742	0.84	70.6	5140	80	1800	950	2305	1785	2060	475	1800	18.45	20.16
121	1120	AMA500L8L	742	0.84	75	5300	80	1800	950	2305	1785	2060	475	1800	18.45	22.59

122	1250	AMA500L8L	742	0.84	79.3	5450	80	1800	950	2305	1785	2060	475	1800	18.45	25.20
123	1400	AMA500L8L	742	0.85	88	5760	80	1800	950	2305	1785	2060	475	1800	18.45	28.24

Table A.3 Electric Motor Database

A.4 Diesel Generator Set Database :

No	Manufacturer	Type	No.of Cyl	Power		alpha	Nominal Speed [rpm]	L	W	H	Db	Ls	Vs	p_me	cm
				kW	hp			[mm]	[mm]	[mm]	[mm]	[mm]	[m3]	[bar]	[m/s]
1	MAN	6L35/44 DF	6	3180	4261.2	0	750	10710	2958	4631	350	440	0.04	20.03	11.00
2	MAN	7L35/44 DF	7	3710	4971.4	0	750	11000	3108	4867	350	440	0.04	20.03	11.00
3	MAN	8L35/44 DF	8	4240	5681.6	0	750	11880	3108	4867	350	440	0.04	20.03	11.00
4	MAN	9L35/44 DF	9	4770	6391.8	0	750	12710	3108	4867	350	440	0.04	20.03	11.00
5	MAN	10L35/44 DF	10	5300	7102	0	750	13490	3108	4867	350	440	0.04	20.03	11.00
6	MAN	6L32/44 CR	6	3600	4824	0	750	10738	2490	4768	320	440	0.04	27.13	11.00
7	MAN	7L32/44 CR	7	3920	5252.8	0	750	11268	2490	4768	320	440	0.04	25.32	11.00
8	MAN	8L32/44 CR	8	4800	6432	0	750	11798	2573	4955	320	440	0.04	27.13	11.00
9	MAN	9L32/44 CR	9	5400	7236	0	750	12328	2573	4955	320	440	0.04	27.13	11.00
10	MAN	10L32/44 CR	10	6000	8040	0	750	12858	2573	4955	320	440	0.04	27.13	11.00
11	MAN	6L32/44	6	3498	4687.32	0	750	10460	2845	4701	320	440	0.04	26.36	11.00
12	MAN	8L32/44	8	4664	6249.76	0	750	11760	3054	4887	320	440	0.04	26.36	11.00
13	MAN	9L32/44	9	5247	7030.98	0	750	12590	3105	4887	320	440	0.04	26.36	11.00
14	MAN	10L32/44	10	5830	7812.2	0	750	13120	3105	4887	320	440	0.04	26.36	11.00
15	MAN	6L32/40	6	3000	4020	0	750	9755	2584	4622	320	400	0.03	24.87	10.00
16	MAN	7L32/40	7	3500	4690	0	750	10285	2584	4622	320	400	0.03	24.87	10.00
17	MAN	8L32/40	8	4000	5360	0	750	11035	2584	4840	320	400	0.03	24.87	10.00
18	MAN	9L32/40	9	4500	6030	0	750	11565	2584	4840	320	400	0.03	24.87	10.00
19	MAN	5L28/32H	5	1100	1474	0	750	6679	1800	3184	280	320	0.02	17.86	8.00
20	MAN	6L28/32H	6	1320	1768.8	0	750	7269	1800	3184	280	320	0.02	17.86	8.00
21	MAN	7L28/32H	7	1540	2063.6	0	750	8179	1800	3374	280	320	0.02	17.86	8.00
22	MAN	8L28/32H	8	1760	2358.4	0	750	8749	1800	3374	280	320	0.02	17.86	8.00

23	MAN	9L28/32H	9	1980	2653.2	0	750	8889	1800	3534	280	320	0.02	17.86	8.00
24	MAN	5L28/32DF	5	1000	1340	0	750	6721	1800	2835	280	320	0.02	16.24	8.00
25	MAN	6L28/32DF	6	1200	1608	0	750	7311	1800	3009	280	320	0.02	16.24	8.00
26	MAN	7L28/32DF	7	1400	1876	0	750	7961	1800	3009	280	320	0.02	16.24	8.00
27	MAN	8L28/32DF	8	1600	2144	0	750	8531	1800	3009	280	320	0.02	16.24	8.00
28	MAN	9L28/32DF	9	1800	2412	0	750	8931	1800	3009	280	320	0.02	16.24	8.00
29	MAN	5L27/38	5	1600	2144	0	750	6832	1770	3712	270	380	0.02	23.53	9.50
30	MAN	6L27/38	6	1980	2653.2	0	750	7557	1770	3712	270	380	0.02	24.27	9.50
31	MAN	7L27/38	7	2310	3095.4	0	750	8002	1770	3899	270	380	0.02	24.27	9.50
32	MAN	8L27/38	8	2640	3537.6	0	750	8667	1770	3899	270	380	0.02	24.27	9.50
33	MAN	9L27/38	9	2970	3979.8	0	750	9112	1770	3899	270	380	0.02	24.27	9.50
34	MAN	5L23/30DF	5	625	837.5	0	750	5671	1210	2749	225	300	0.01	16.77	7.50
35	MAN	6L23/30DF	6	750	1005	0	750	6091	1210	2749	225	300	0.01	16.77	7.50
36	MAN	7L23/30DF	7	875	1172.5	0	750	6511	1210	2749	225	300	0.01	16.77	7.50
37	MAN	8L23/30DF	8	1000	1340	0	750	6931	1210	2749	225	300	0.01	16.77	7.50
38	MAN	5L21/31	5	1000	1340	0	1000	5829	1400	3183	210	310	0.01	22.35	10.33
39	MAN	6L21/31	6	1320	1768.8	0	1000	6314	1400	3183	210	310	0.01	24.59	10.33
40	MAN	7L21/31	7	1540	2063.6	0	1000	6639	1400	3289	210	310	0.01	24.59	10.33
41	MAN	8L21/31	8	1760	2358.4	0	1000	7682	1400	3289	210	310	0.01	24.59	10.33
42	MAN	9L21/31	9	1980	2653.2	0	1000	8062	1400	3289	210	310	0.01	24.59	10.33
43	Wartsila	4L20	4	800	1072	0	1000	4910	1168	2338	200	280	0.01	27.28	9.33
44	Wartsila	6L20	6	1200	1608	0	1000	5325	1299	2373	200	280	0.01	27.28	9.33
45	Wartsila	8L20	8	1600	2144	0	1000	6030	1390	2524	200	280	0.01	27.28	9.33
46	Wartsila	9L20	9	1800	2412	0	1000	6535	1390	2574	200	280	0.01	27.28	9.33
47	Wartsila	6L26	6	1969	2638.46	0	1000	7500	2300	3043	260	320	0.02	23.18	10.67
48	Wartsila	8L26	8	2625	3517.5	0	1000	8000	2300	3068	260	320	0.02	23.18	10.67
49	Wartsila	9L26	9	2953	3957.02	0	1000	8500	2300	3168	260	320	0.02	23.17	10.67

50	Wartsila	6L32	6	3000	4020	0	750	8345	2290	3940	320	400	0.03	24.87	10.00
51	Wartsila	6L32	6	3480	4663.2	0	750	8345	2490	3745	320	400	0.03	28.85	10.00
52	Wartsila	7L32	7	3500	4690	0	750	9215	2690	4140	320	400	0.03	24.87	10.00
53	Wartsila	8L32	8	4000	5360	0	750	9755	2690	3925	320	400	0.03	24.87	10.00
54	Wartsila	8L32	8	4640	6217.6	0	750	10410	2690	4010	320	400	0.03	28.85	10.00
55	Wartsila	9L32	9	4500	6030	0	750	10475	2890	3925	320	400	0.03	24.87	10.00
56	Wartsila	9L32	9	5220	6994.8	0	750	10505	2890	4010	320	400	0.03	28.85	10.00
57	Wartsila	6L38	6	4350	5829	0	600		3000	4752	380	475	0.05	26.92	9.50
58	Wartsila	8L38	8	5800	7772	0	600		3350	4752	380	475	0.05	26.92	9.50
59	Wartsila	9L38	9	6525	8743.5	0	600		3350	4752	380	475	0.05	26.92	9.50
60	MAN	12V32/44 CR	12	7200	9648	45	750	11338	4260	5014	320	440	0.04	27.13	11.00
61	MAN	14V32/44 CR	14	7840	10505.6	45	750	11968	4260	5014	320	440	0.04	25.32	11.00
62	MAN	16V32/44 CR	16	9600	12864	45	750	12598	4260	5014	320	440	0.04	27.13	11.00
63	MAN	18V32/44 CR	18	10800	14472	45	750	13228	4260	5014	320	440	0.04	27.13	11.00
64	MAN	20V32/44 CR	20	12000	16080	45	750	13858	4260	5014	320	440	0.04	27.13	11.00
65	MAN	12V32/40	12	6000	8040	45	750	11045	3365	4850	320	400	0.03	24.87	10.00
66	MAN	14V32/40	14	7000	9380	45	750	11710	3365	4850	320	400	0.03	24.87	10.00
67	MAN	16V32/40	16	8000	10720	45	750	12555	3730	5245	320	400	0.03	24.87	10.00
68	MAN	18V32/40	18	9000	12060	45	750	13185	3730	5245	320	400	0.03	24.87	10.00
69	MAN	MAN175D	12	1440	1929.6	45	1500	5530	1641	2365	175	215	0.01	18.56	10.75
70	MAN	12V26	12	3937	5275.58	55	1000	8400	2700	3686	260	320	0.02	23.17	10.67
71	MAN	16V26	16	5250	7035	55	1000	9700	2700	3716	260	320	0.02	23.18	10.67
72	MAN	12V32	12	6000	8040	55	750	10075	3060	4365	320	400	0.03	24.87	10.00
73	MAN		12	6960	9326.4	55	750	10700	3060	4130	320	400	0.03	28.85	10.00
74	MAN	16V32	16	8000	10720	55	750	11175	3060	4280	320	400	0.03	24.87	10.00
75	MAN		16	9280	12435.2	55	750	11465	4245	4445	320	400	0.03	28.85	10.00
76	MAN	18V32	18	9000	12060	55	750	11825	3360	4280	320	400	0.03	24.87	10.00

77	MAN	12V38	12	8700	11658	50	600		3650	5012	380	475	0.05	26.92	9.50
78	MAN	16V38	16	11600	15544	50	600		3650	5212	380	475	0.05	26.92	9.50
79	MAN	8V31	8	4880	6539.2	50	750				310	430	0.03	30.07	10.75
80	MAN	10V31	10	6100	8174	50	750				310	430	0.03	30.07	10.75
81	MAN	12V31	12	7320	9808.8	50	750				310	430	0.03	30.07	10.75
82	MAN	14V31	14	8540	11443.6	50	750				310	430	0.03	30.07	10.75
83	MAN	16V31	16	9760	13078.4	50	750				310	430	0.03	30.07	10.75

Table A.4 Diesel Generator Set Database

B

Regression Analysis and Sensitivity Analysis of other Diesel Engine Types

There are three types of diesel engines being considered in this thesis. The four strokes engine with L-cylinder arrangement, the four strokes engine with V-cylinder arrangement and the two-strokes engine. In the chapter 3 and 5 only the results of the four strokes engine with L-cylinder arrangement are shown. This appendix will provide the rest of the diesel engine types.

B.1 Diesel Engine with V Cylinder Arrangement Regression Analysis

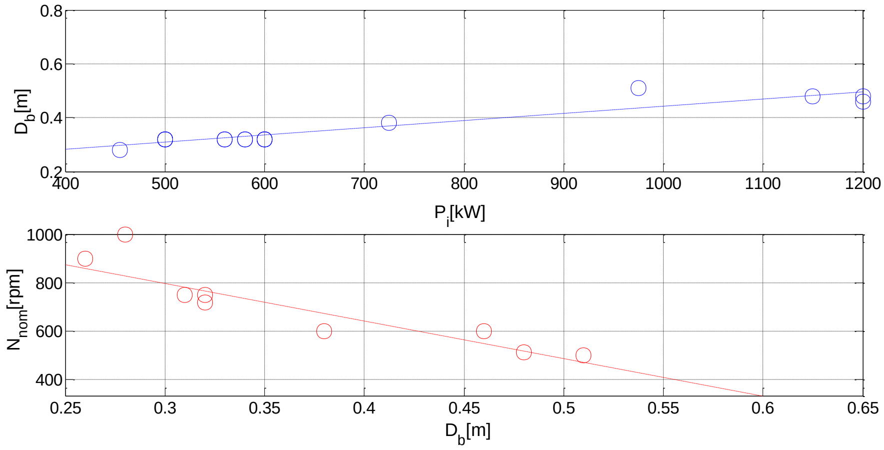

Figure B.1 Linear regression analysis of power per cylinder and bore diameter (above) and Linear regression analysis of bore diameter and nominal speed of the diesel engine (below)

B.2 Diesel Engine with V Cylinder Arrangement Sensitivity Analysis

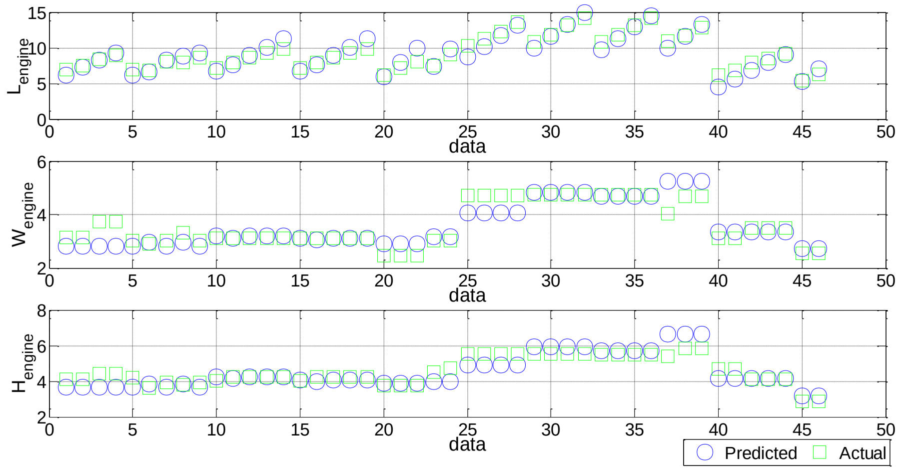

Figure B.2 Diesel Engine with V-cylinder arrangement Sizing Results and Comparisons

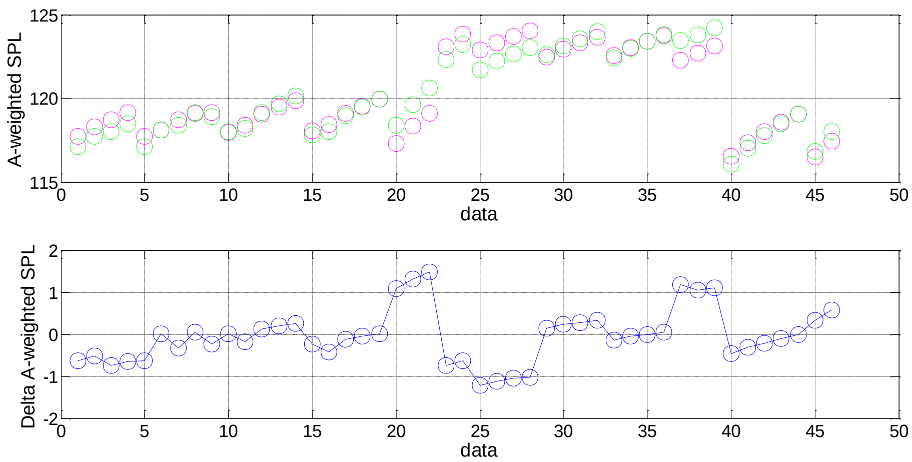

Figure B.3 Diesel Engine with V-cylinder arrangement SPL Calculation Results and Comparisons

B.3 Two Strokes Diesel Engine Regression Analysis

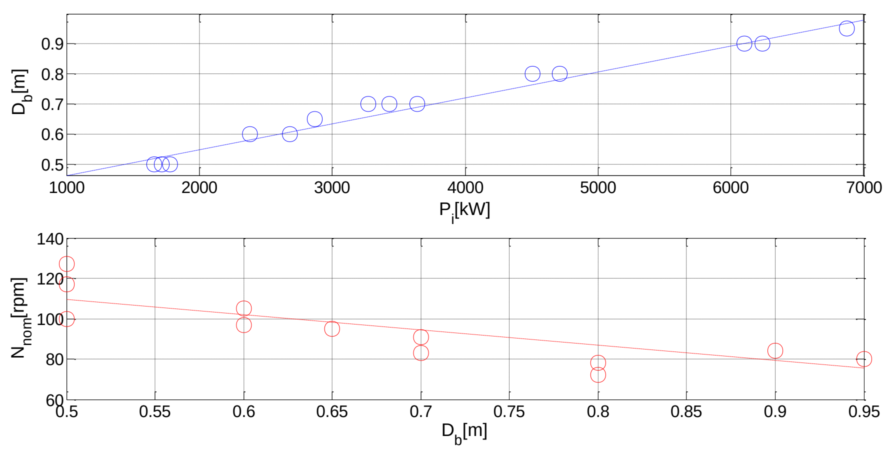

Figure B.4 Linear regression analysis of power per cylinder and bore diameter (above) and Linear regression analysis of bore diameter and nominal speed of the diesel engine (below)

B.4 Two Strokes Diesel Engine Sensitivity Analysis

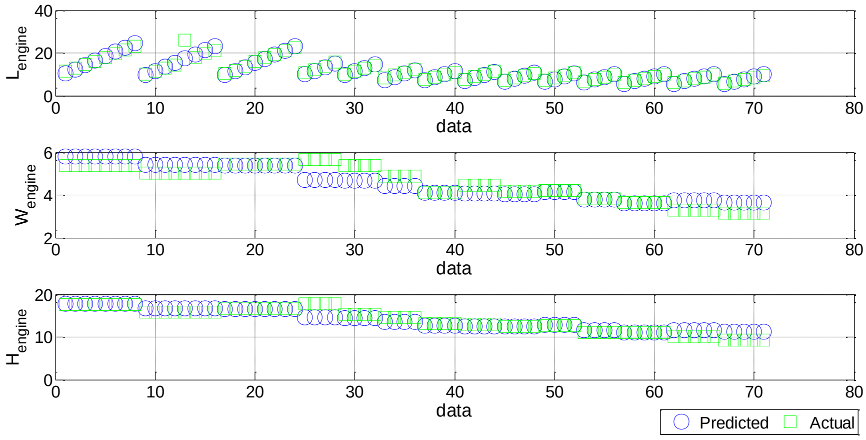

Figure B.5 Diesel Engine with 2-strokes arrangement Sizing Results and Comparisons

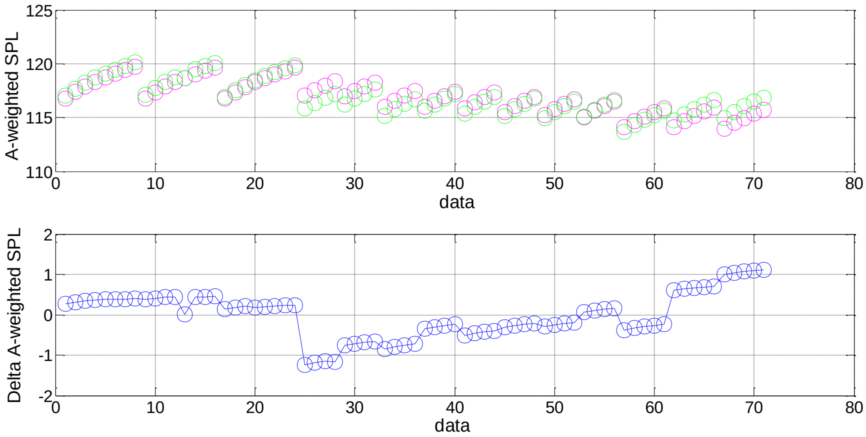

Figure B.6 Diesel Engine with 2-strokes arrangement SPL Calculation Results and Comparisons

C

Added SNAME Coefficients

C.1 Mounting Transfer Functions

The mounting transfer functions are presented here. Various mountings can be used to reduce the structure-borne noise level of the equipment. Every mounting type has a different transfer function value, depends on how efficient the mounting reduce the structure-borne noise level.

Foundation Type	Weight Class	31.5	63	125	250	500	1000	2000	4000	8000
Hard-mounted B	1	-13	10	8	6	6	6	6	6	6
	2	9	7	6	5	5	5	5	5	5
	3	5	4	3	3	3	3	3	3	3
Hard-mounted A	1	6	6	6	6	6	6	6	6	6
	2	5	4	4	4	4	4	4	4	4
High-Frequency mounting B	1	13	11	9	8	10	15	15	15	15
	2	9	7	7	6	8	8	9	10	10
	3	5	4	3	3	3	3	4	5	8
High-Frequency mounting A	1	6	6	6	7	8	9	10	10	10
	2	5	4	4	4	4	4	5	6	8
Low-Frequency mounting B	1	20	25	30	30	30	30	30	30	30
	2	12	16	20	23	25	25	25	25	25
	3	8	12	13	14	15	18	20	20	20
Low-Frequency mounting A	1	9	14	20	23	25	25	25	25	25
	2	4	8	12	14	17	20	20	20	20
Two-stage mounting B	1	25	33	40	45	50	50	50	50	50
	2	22	30	35	40	45	48	50	50	50
	3	20	25	30	35	40	45	50	50	50
Two-stage mounting A	1	20	25	30	35	40	45	45	45	45
	2	15	22	27	32	35	40	45	45	45

Table C.1 Mounting Transfer Functions Table

C.2 Hull Loss Factors

The hull loss factors represent the transmission loss by the type of hull structure. The SNAME Model categorizes it into four types, i.e., (1) Steel, Dry (2) Steel, Fluid Loaded (3) Alumunium, Dry and (4) Alumunium, Fluid Loaded. However, in this project the hull structure is assumed to be steel, dry.

f (Hz)	31.5	63	125	250	500	1000	2000	4000	8000
Steel, Dry	0.008	0.007	0.006	0.005	0.005	0.004	0.004	0.003	0.003
Steel, Fluid Loaded	0.04	0.035	0.03	0.025	0.016	0.013	0.09	0.007	0.006
Alumunium, Dry	0.013	0.012	0.011	0.010	0.009	0.008	0.007	0.007	0.006
Alumunium, Fluid Loaded	0.06	0.055	0.05	0.035	0.027	0.021	0.014	0.012	0.009

Table C.2 Hull Loss Factors Table

D

Operation Points for Various Loading Points

This appendix shows the operation point of the case study ship for different transmission types and dfifferent configurations. It is assumed that when more than one equipment (with the same type) is working the equal loading operation is implemented.

D.1 Diesel-Mechanical

	1-DM		2-DM			3-DM			4-DM				
gamma	De-1	Dg-1	De-1	De-2	Dg-1	De-1	De-2	Dg-1	De-1	De-2	De-3	De-4	Dg-1
10	400	100	400	0	100	400	0	100	400	0	0	0	100
20	800	200	800	0	200	800	0	200	800	0	0	0	200
30	1200	300	1200	0	300	1200	0	300	600	600	0	0	300
40	1600	400	1600	0	400	1600	0	400	800	800	0	0	400
50	2000	500	2000	0	500	2000	0	500	1000	1000	0	0	500
60	2400	600	1200	1200	600	1200	1200	600	600	600	600	600	600
70	2800	700	1400	1400	700	1400	1400	700	700	700	700	700	700
80	3200	800	1600	1600	800	1600	1600	800	800	800	800	800	800
90	3600	900	1800	1800	900	1800	1800	900	900	900	900	900	900
100	4000	1000	2000	2000	1000	2000	2000	1000	1000	1000	1000	1000	1000

Table D.1 Diesel-Mechanical Operation Points for Various Gamma

D.2 Diesel-Electrical

	1-DE			2-DE				4-DE					
gamma	Em-1	Em-2	Dg-1	Em-1	Em-2	Dg-1	Dg-2	Em-1	Em-2	Dg-1	Dg-2	Dg-3	Dg-4
10	400	0	500	400	0	500	0	400	0	500	0	0	0
20	800	0	1000	800	0	1000	0	800	0	1000	0	0	0
30	1200	0	1500	1200	0	1500	0	1200	0	750	750	0	0
40	1600	0	2000	1600	0	2000	0	1600	0	1000	1000	0	0
50	2000	0	2500	2000	0	2500	0	2000	0	1250	1250	0	0
60	1200	1200	3000	1200	1200	1500	1500	1200	1200	1000	1000	1000	0
70	1400	1400	3500	1400	1400	1750	1750	1400	1400	1166.667	1166.667	1166.667	0
80	1600	1600	4000	1600	1600	2000	2000	1600	1600	1000	1000	1000	1000
90	1800	1800	4500	1800	1800	2250	2250	1800	1800	1125	1125	1125	1125
100	2000	2000	5000	2000	2000	2500	2500	2000	2000	1250	1250	1250	1250

	6-DE							
gamma	Em-1	Em-2	Dg-1	Dg-2	Dg-3	Dg-4	Dg-5	Dg-6
10	400	0	500	0	0	0	0	0
20	800	0	500	500	0	0	0	0
30	1200	0	750	750	0	0	0	0
40	1600	0	666.6667	666.6667	666.6667	0	0	0
50	2000	0	833.3333	833.3333	833.3333	0	0	0
60	1200	1200	750	750	750	750	0	0
70	1400	1400	700	700	700	700	700	0
80	1600	1600	800	800	800	800	800	0
90	1800	1800	750	750	750	750	750	750
100	2000	2000	833.3333	833.3333	833.3333	833.3333	833.3333	833.3333

Table D.2 Diesel-Electrical Operation Points for Various Gamma

D.3 Diesel-Hybrid

	1-DH			2-DH					
gamma	Em-1	De-1	Dg-1	Em-1	Em-2	De-1	De-2	Dg-1	Dg-2
10	400	0	500	400	0	0	0	500	0
20	800	0	1000	400	400	0	0	1000	0
30	1200	0	1500	600	600	0	0	750	750
40	600	1000	1000	300	300	500	500	500	500
50	750	1250	1250	375	375	625	625	625	625
60	900	1500	1500	450	450	750	750	750	750
70	1050	1750	1750	525	525	875	875	875	875
80	1200	2000	2000	600	600	1000	1000	1000	1000
90	1350	2250	2250	675	675	1125	1125	1125	1125
100	1500	2500	2500	750	750	1250	1250	1250	1250

Table D.3 Diesel-Hybrid Operation Points for Various Gamma

E

Noise Comparisons between Equipment

This appendix shows the airborne noise and the structure-borne noise comparisons between the electric motor and generator versus the gearbox. The ship requirements data are the same as the case study. The number of engines for all equipment is one. The comparisons aim to show that the real propulsion system configurations with electric motor(s) can have lower overall noise level compared to the prediction by the SNAME model.

E.1 Electric Motor + Generator vs Gearbox

The figures below show the noise comparisons between the mentioned equipment. The airborne noise results indicate that one gearbox has higher airborne noise compared to electric motor and generator combined. From the verification results, it is evident that the SNAME model overestimates the airborne noise of the electric motor. The difference between the measurement value and the predicted value by SNAME is the highest compared to other equipment. The gearbox airborne noise is overestimated by approximately 0.5 dBA from the value given by TNO. On the other hand, the SNAME model has overestimated the electric motor airborne noise by approximately 10 – 15 dBA compared to the measurement value from ABB. In that sense, if the electric motor prediction can be made more accurate, it is expected the configurations with electric motor(s) can possibly have lower overall noise level, especially the battery plant. The diesel-electric plant can have lower overall airborne noise too, but it would not be significant because the diesel-electric plant still has diesel engine(s) in its configuration.

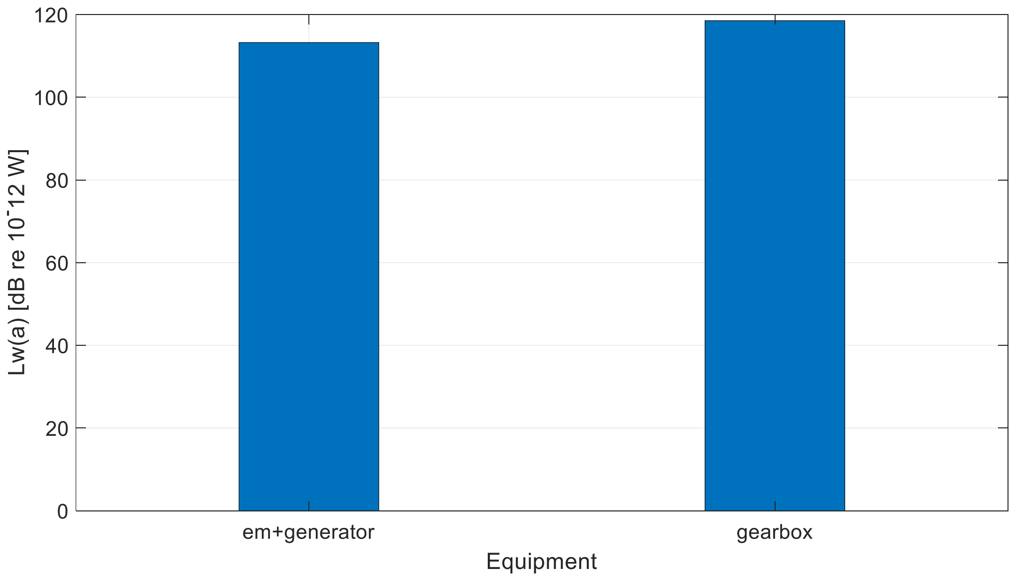

Figure E.1 EM + Generator vs Gearbox Airborne Noise Results

The structure-borne noise comparisons show that the combination of electric motor and generator has a higher structure-borne noise compared to the gearbox. However, it is expected the prediction of generator and electric motor structure-borne noise levels are higher than its real acceleration level. As explained in the subchapter 4.5.1, the SNAME model for generator structure-borne noise level is not very accurate since the contribution from the diesel engine contaminates its measurement. Furthermore, the generator formula is also used to calculate the electric motor structure-borne noise level since both of the equipment has similar characteristics.

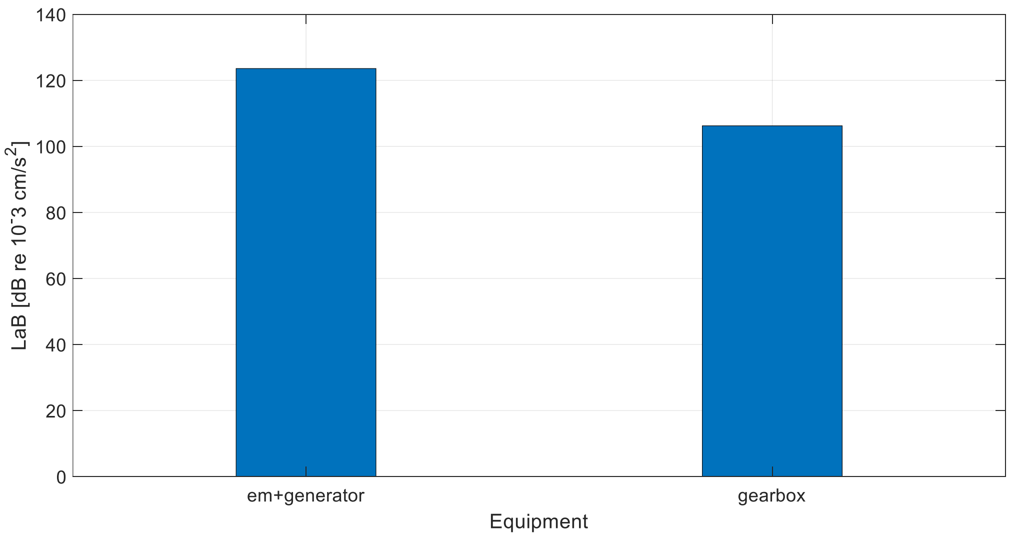

Figure E.2 EM + Generator vs Gearbox Structure-borne Noise Results

Symbol	Description	Unit
ρ	$density$	kg/m^3
η	$damping\ loss\ factor$	$-$
ε	$primary\ element\ regression\ coefficient$	$-$
γ	$loading\ factors$	$-$
β	$correction\ factor\ for\ non-boundary\ surfaces$	$-$
β	$overall\ dimensions\ regression\ coefficient$	$-$
α	$Sabine\ absorption\ coefficients$	$-$
z	$number\ of\ cylinders$	$-$
v	$velocity$	m/s
t	$time$	s
t	$thickness\ of\ the\ plate$	
r	$distance\ from\ the\ source\ to\ the\ receiver$	ft
p	$pressure$	Pa
m	$mass$	kg
k	$number\ of\ revolutions\ per\ cylce$	$-$
f	$frequency$	Hz
d	$displacement$	m
c	$speed\ of\ the\ sound$	m/s
b	$width\ of\ the\ foundation$	m
a	$acceleration$	m/s^2
a	$length\ of\ the\ foundation$	m
W	$sound\ power$	$Watt$
W	$equipment\ or\ room\ width$	m
TL_{inter}	$transmission\ losses\ by\ intersection$	$dB\ re\ a_{ref}$
S	$boundary\ surface\ area$	m^2

R	room constant	$-$
Q	sound directivity factor	$-$
P	power	$Watt$
L	equipment or room length	m
I	sound intensity	W/m^2
H	equipment or room height	m
E	vibrational energy	$Joule$
AV_a	$A-$ scale correction factor values	$-$
ΔL_d	transmission losses beyond effective source area	$dB\ re\ a_{ref}$
ΔL_a	transmission losses within the compartment	$dB\ re\ a_{ref}$
Δ	octave band adjustmenst values	dB
χ_{min}	minimum height between the equipment and the boundary surface	m
τ_{em}	mean shear stress	kN/m^2
λ_s	shape ratio	$-$
λ_R	rotor shape ratio	
Ω_{min}	minimum length between the equipment and the boundary surface	m
Ψ_{min}	minimum width between the equipment and the boundary surface	m
v_t	circumferential speed	m/s
v_{ref}	reference velocity	$10^{-9}\ m/s$
r_{fr}	minimum spacing between frame	m
p_{ref}	reference pressure	$2 * 10^{-5}\ Pa$
p_{me}	mean effective pressure	bar
p_0	maximum sound pressure	Pa
n_s	number of surfaces	$-$
f_s	shaft frequency	Hz
f_m	gear mesh frequency	Hz
d_{ref}	displacement reference	$10^{-11}\ m$

c_m	*mean piston speed*	m/s
a_{ref}	*reference acceleration*	$10^{-3}\ cm/s^2$
W_0	*reference sound power*	$10^{-12}\ Watt$
R_h	*radius of the gear housing*	m
P_{tot}	*total installed power*	kW
P_{prop}	*total propulsion power*	kW
P_{nom}	*nominal power*	kW
P_{in}	*input power*	kW
P_i	*power per cylinder*	kW/cyl
P_{comb}	*power combination*	kW
$P_{b,em}$	*electric motor brake power*	kW
$P_{b,dg}$	*diesel generator set brake power*	kW
$P_{b,de}$	*diesel engine brake power*	kW
P_{aux}	*total auxiliary power*	kW
N_{teeth}	*number of gear teeth*	$-$
N_{shaft}	*number of shafts*	$-$
N_{room}	*number of compartments/rooms*	$-$
$N_{propeller}$	*propeller speed*	rpm
N_{nom}	*nominal speed*	rpm
N_i	*number of subsystems*	$-$
N_{em}	*number of electric motors*	$-$
N_{dg}	*number of diesel generator sets*	$-$
N_{de}	*number of diesel engines*	$-$
N_{act}	*actual speed*	rpm
M_{gb}	*gearbox torque*	Nm
L_w	*sound power level*	$dB\ re\ 10$
L_v	*velocity level*	$dB\ re\ v_{ref}$

L_s	*stroke length*	m
L_p	*sound pressure level*	$dB\ re\ p_{ref}$
$L_{p,rev}$	*reverberation sound pressure level*	$dB\ re\ p_{ref}$
$L^*_{p,eq}$	*normalised A − weighted sound pressure level*	$-$
$L_{p,eq}$	*equivalent sound pressure level*	$dB\ re\ p_{ref}$
L_d	*displacement level*	$dB\ re\ d_{ref}$
L_{aB}	*acceleration level*	$dB\ re\ a_{ref}$
$L^*_{aB,eq}$	*normalised equivalent acceleration level*	$-$
$L_{P,dir}$	*direct sound pressure level*	$dB\ re\ p_{ref}$
$L_{A,eq}$	*a − weighted equivalent sound pressure level*	$dB(A)\ re\ p_{ref}$
I_{ref}	*reference sound intensity*	$10^{-12}\ W/m^2$
D_b	*bore diameter*	mm
Δ_{tol}	*gear manufacturing tolerance adjustments*	$-$

List of Figures

List of Tables