NTNU
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Science and Technology

Mechanical Design of Massive Genset-Modules

Elias Haugsbø

Marine Technology
Submission date: June 2017
Supervisor: Eilif Pedersen, IMT

Norwegian University of Science and Technology
Department of Marine Technology

NTNU Trondheim
Norwegian University of Science and Technology
Department of Maine Technology

PROJECT DESCRIPTION SHEET

Name of the candidate: Elias Haugsbø

Field of study: Marine Machinery Engineering

Thesis title (Norwegian): Mekanisk design av massiv generator modul.

Thesis title (English): Mechanical design of massive genset module system.

Background

The increased focus on cost efficient operation, and ever stricter emission regulations from the international maritime organization (IMO), has open for new thinking regarding propulsion system. A method that has been increasingly used since the early 80s is diesel electric propulsion system. Diesel electric system commonly consist of 2-10 gensets working together, however is diesel genset know to have low efficiency at low loads. For a vessel operation with large load variations such a configuration can lead to high operation costs.

The idea is to consider a diesel electric power system consisting of 10-50 gensets, hooked together in a common frame. A high number of gensets working together can give a flexible power production, and flatten the specific fuel consumption over a large span. This can lead to lower operation cost and reduced emission.

Work description

1. Discuss the use of diesel electric power system, and what technology that has been used over the last decades.
2. Make a system description, and discuss the advantages a massive genset module system can have compared with more conventional diesel electric systems.
3. Modeling of the oscillation created by the genset module by use of Lagrange equation and simulation in 20-Sim.

4. Mechanical design of frame and genset module. Analysis of how the frame structure is influenced by the oscillating motion of a high number of genset modules. (by use of SolidWorks)
5. Propose a solution on the connection method between the frame and genset module.

Guidelines

The report shall be written in English and edited as a research report including literature survey, description of mathematical models, description of control algorithms, simulations results, discussion and conclusion including a proposal for further work. Source code developed shall be provided on a CD or equivalent with code listing enclosed in appendix. The Department of Marine Technology, NTNU, can use the results freely in its research work by referring to the students work.

Start date: January 15, 2017 **Due date:** June 11, 2017

Supervisor: Eilif Pedersen
Co-advisor(s): None

Trondheim, 18.01.2017

Eilif Pedersen
Supervisor

SUMMARY

The research conducted in this thesis aims to develop a mechanical design of a large electric propulsion system. This electric propulsion system consisting of multiple genset operating together to reduce operation cost and emissions. The idea with this configuration is to design a stacking frame with room for 12 genset in the power range 200 to 400 kW. By adjusting the number of frames and generators, power output from this configuration can meet a large variety of vessel types and deliver a flexible solution.

The massive genset system is intended to consist of three main parts, a stacking frame that allow compact storage of the modules, the genset-module designed as a canopy and connection system for exhaust and auxiliary systems. Since the diesel engine produce dynamic loads from operation it is important to investigate if the frame can withstand the vibration excitation from multiple gensets. To analyze the load acting on the frame it is required to make a dynamic model of the engine mounting system, that give the excitation force from the reciprocating motion. The method used to develop the model is based on the Lagrange equation of motion with implementation into Bond graph. This allow simulation of a rigid body system that evaluate frequency response and excitation force in the mountings.

The frame module was modulated in SolidWorks where the dynamic loads was implemented into a multi-body simulation tool. The result is compared to DNV GL rules for machinery vibration in steel structures. The results show that the frame design is within the limitations from vibration excitations of machinery. How the genset can be connected together with a common supply system is discussed, a few possible configurations are introduced and visualized through sketches. Finally, a sketch of the machinery room layout presented. The work conducted in this thesis is not a complete design of the massive genset system, but can give a good foundation for further development of the concept.

SAMMENDRAG

Forskningen som gjennomføres i denne oppgaven tar sikte på å utvikle en mekanisk utforming av et stort diesel-elektrisk fremdriftssystem. Dette fremdriftssystemet skal består av flere generatorer som opererer sammen for å redusere driftskostnader og utslipp. Ideen med denne konfigurasjonen er å designe en stablingsramme med plass til 12 generatorer i effektområdet 200 - 400 kW. Ved å justere antall rammer og generatorer kan effekten fra denne konfigurasjonen møte et stort utvalg av fartøystyper og levere en fleksibel løsning.

Systemet skal hovedsakelig bestå av tre individuelle deler, den første er en ramme som tillater kompakt stabling av generator moduler, den andre er kombinert diesel motor og generator koblet sammen i en beholder. Den siste komponenten er tilkoblings metode for eksos, kjølevann, drivstoff og elektrisitet. Siden diesel motoren produserer stor dynamisk last som påvirkere rammen, er det viktig å undersøke om rammen tåler belastningene. For å analysere belastningen som virker på rammen, vil det være nødvendig å lage en dynamisk modell som gir eksitasjonskraften i motoropplagrene. Metoden som blir brukt for å lage modellen er basert på implementering av Lagrange-ligning in i bond graph. Dette tillater simulering av komplekse dynamiske systemer som gjør det mulig å evaluere frekvensrespons og opplagerkrefter. Videre er rammen modulert i SolidWorks, som inneholder et verktøy for simulering av dynamiske krefter i konstruksjonen. Dette tillater implementering av opplagerkreftene slik at resultatene kan bli sammenlignet med reglene for stålkonstruksjoner fra DNV GL. Resultatene viser at rammen er innenfor kriteriene på maks hastighet og frekvensspekteret er flyttet ut av eksitasjonsområdet til dieselmotoren. Til slutt er det presentert noen forslag til tilkoblingsmetoder mellom rammen og rørsystemet, dette er presentert gjennom skisser. Arbeidet som er gjennomført i denne oppgaven presenterer ikke et fullstendig design av systemet, men gir et godt grunnlag for videre arbeid og utvikling av konseptet.

PREFACE

This master thesis is submitted to the Norwegian University of Science and Technology (NTNU) and account for the total workload in the final semester before completing the degree Master of Science in Marine Technology. The research in this report is introducing a new concept for electric propulsion, where the aim is to reduce the emission produced from vessel operation.

This research is executed at the Department of Maine Technology and has been supervised by Associate Professor Eilif Pedersen. I would like to thank Eilif Pedersen for introducing me for the idea and good advices during the process.

Finally, I would like to give a thanks to my fellow student Åsmund Kyrkjeeide Karlsen and Jarl Bernhard Berg Kjølseth for proofreading, and to my office colleague for positive discussions and motivation through a long semester.

$Trondheim, June\ 9^{th}, 2017$

Elias Haugsbø

Elias Haugsbø

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ABBREVIATIONS

AC	Alternating Current
AES	All-Electric Ship
BDC	Bottom Dead Center
BESS	Battery Energy Storage System
CAD	Computer Aided Design
CAE	Computer-Aided Engineering
CI	Compression Ignition
DC	Direct Current
DF	Duel Fuel
DOF	Degree of Freedom
DP	Dynamic Positioning
ECA	Emission Control Area
EGR	Exhaust Gas Recirculation
FCCC	Framework Conventional Climate Change
FFT	Fast Fourier Transformation
FO	Fuel Oil
GHG	Greenhouse Gasses
HFO	Heavy Fuel Oil
HV	High Voltage
IMO	International Maritime Organization
ISO	International Standard Organization

LNG Liquefied Natural Gas

NOMENCLATURE

$\boldsymbol{m}$	$Mass\ [kg]$
$\ddot{\boldsymbol{x}}$	$Acceleration\ [m/s^2]$
$\dot{\boldsymbol{x}}$	$Velocity\ [m/s]$
$\boldsymbol{x}$	$Displacment\ [m]$
$\boldsymbol{F}$	$Force\ [N]$
$\boldsymbol{k}$	$Spring\ stiffnes\ [N/m]$
$\boldsymbol{c}$	$Actual\ dempening\ [Ns/m]$
$\boldsymbol{\omega_n}$	$Radian\ frequency\ [rad/s]$
$\boldsymbol{f_n}$	$Natrual\ frequency\ [Hz]$
$\boldsymbol{t}$	$Time\ [s]$
$\boldsymbol{c_c}$	$Dampening\ coefficient\ [Ns/m]$
$\boldsymbol{\xi}$	$Dampening\ ratio$
$\boldsymbol{\omega_d}$	$Dampened\ frequency\ [rad/s]$
$\boldsymbol{M}$	$Moment\ [Nm]$
$\dot{\boldsymbol{p}}$	$Effort$
$\dot{\boldsymbol{q}}$	$flow$
$\boldsymbol{q_j}$	$Generalized\ coordinates$
$\boldsymbol{Q_j}$	$Virtual\ work$
$\boldsymbol{\partial r_j}$	$Virtual\ displacment$
$\boldsymbol{V}$	$Potential\ energy$
$\boldsymbol{T}$	$Kinetic\ energy$
$\boldsymbol{\psi}$	$Euler\ angle\ z$
$\boldsymbol{\theta}$	$Euler\ angle\ y$
$\boldsymbol{\phi}$	$Euler\ angle\ x$

$\boldsymbol{\omega}$	*angular velocity* $[rad/s]$
$\boldsymbol{M}$	*Mass matrix*
$\boldsymbol{C}(\boldsymbol{v})$	*Coriolis centfugal force*
$\boldsymbol{\tau}$	*External force*
$\boldsymbol{z_g}$	*Center of gravity*
$\boldsymbol{P_c}$	*Cylindere pressure* $[Pa]$
$\boldsymbol{A_p}$	*Area piston* $[m^2]$
$\boldsymbol{l}$	*Connecting rod* $[m]$
$\boldsymbol{r}$	*Crank radius* $[m]$
$\boldsymbol{B}$	*Bore* $[m]$
$\boldsymbol{S}$	*Stroke* $[m]$
$\boldsymbol{g}$	*Gravity constant* $[]$
$\boldsymbol{J_i}$	*Moment of inertia*
$\boldsymbol{\delta\phi}$	*Phase angle* $[deg]$
$\boldsymbol{N_{cyl}}$	*Number of cylinders*
$\boldsymbol{C_r}$	*Resistance coefficient*
$\boldsymbol{\lambda}$	*Crank arm ratio*
$\boldsymbol{MT}$	*Translation matrix*
$\boldsymbol{P}$	*Power* $[W]$
$\boldsymbol{R}$	*Transformation matrix*

1 INTRODUCTION

1.1 BACKGROUND

The world is challenged by climate change, and the population is forced to reduce the environmental foot print on the earth. Marine traffic is a significant contributor to the emission of greenhouse gasses (GHG), and account for more than 2.2 % of the worlds carbon dioxide (CO_2) emission every year, according to International Maritime Organization (IMO, 2014). The largest contributor to marine traffic is by fare transportation and trade of goods. Approximately 95 % of the world trade is transported by sea, and with a growing economy and closer connection between countries the future trade is expected to increase. Other vessel-types operating within markets as fishing, offshore, Ro-Ro ferries and cruise are significant contributors to the marine emissions. These types of vessels are of special interest because they operate with larger load variations compared to transportation ships with long transient voyage.

Historically there has been few regulations regarding international shipping. This has allowed the vessels owners to neglect the environmental responsibility and focus on cheap operation. However, increased focus on climate change and clean operation has raised the last century. The framework conventional climate change (FCCC) conference in 1992, that culminated into the Kyoto protocol in 1997 was the initial step towards an international climate policy. (Michaelowa and Krause, 2000) This lead to the Tier 1 protocol that was developed by IMO. Tire 1 mainly contained measures to reduce the emission of nitrogen oxides (NO_x) and Sulphur oxides (SO_x) from marine traffic. This has been developed further into Tier 3 with even stricter emission regulations, that was put to act in 2016. Combined with the even stricter emission measures in the emission control area (ECA) that include the Baltic sea, North Sea, coast of America and Caribbean.

Earlier the marine traffic had mainly been choosing the fuel according to price and availability. Now with the new emission regulation, the vessel operators are forced to choose machinery installation and fuel that keep them within the regulation criteria. After an extended period with high fuel prices there has also been a focus on increasing efficiency of the machinery and hull construction to reduce the operation cost. With the fast development in technology the companies are forced to develop their fleet to stay competitive and keep the operation profitable.

The combination of emission regulation and pressure of reducing operation cost, has forced the marine sector to develop new methods and technology to increase vessel efficiency. Mix of low cost, good availability and high energy density has made diesel fuel the preferred energy source for marine trafficking. Almost all the fuel used in marine traffic is diesel oil. The most commonly used diesel type has been the low raffinate fuel called heavy fuel oil (HFO). Because of little processing HFO is cheap and have high availability. However, HFO can contain a concentration of Sulphur up to 4.5 % and other impurities that increase the emission. New regulations give limitations of Sulphur content of 1.5 % in international water and 0.5 % in ECA, this forces the vessel operators to look toward alternative fuels such as biofuel and liquefied natural gas (LNG) or more refined diesel oil such as marine diesel oil (MDO). The alternative is to use after treatment systems on the exhaust gas to reduce the emission. Scrubbers, Selective catalytic reduction (SCR) and exhaust gas recirculation (EGR) are all methods that can reduce the emission from the vessel. The disadvantages of these methods are that they are expensive to install and can reduce the fuel efficiency of the engine.

The diesel engine is known to have a significantly reduction of efficiency with reduced load. Low load operation of diesel engines is also known to increase production of soot and lead to wear and tear of the machinery. For vessels operating with large load variation it is preferred to install a high number of engines. A common system installation in such vessels is a diesel electric propulsion system. This enables control over how many engines that are running and can help optimize the fuel consumption. A diesel electric system commonly consist of 2-10 generator sets depending on various factors such as cost, redundancy demand and available power. This gives the possibility to switch on and off generators depending on power demand. High number of installed generators give good redundancy in case of failure, this is required in many vessel applications. Another advantage with an electrical system is that it requires less space and reduce vibrations since driveshafts are replaced by electric cables. Space is often a concern when designing a vessel and diesel electric system can give more flexibility since the location of the engines are not dependent on where the thrust units are placed (Ådnanes, 2003).

1.2 SCOPE AND LIMITATIONS

The research done in this thesis aims to propose a new concept where a high number of genset are operating together. It would be interesting to investigate if such a system can be implemented to reduce the emissions and operation cost for marine vessels. The idea is to construct genset-modules that can be stacked into a common frame. By implementing up to fifty genset in a vessel it can increase flexibility significantly during large load variations. The intended benefits are listed beneath.

- Constant specific fuel oil consumption (SFOC) over a large operation range.
- Reduced operation and maintenance cost.
- Increase availability of the vessel.

Before this can be realized there are many issues that must be studied. It can be summarized into three categories. Mechanical design, control and auxiliary optimization. Modern technology gives excellent control of closed electric loops, by implementing a state of the art management system it would be possible to optimize load sharing between the generators and reduce total power consumption. Another goal is to develop a common auxiliary system for the module, this can reduce pumping work. In general, there are many problems that can be considered for this configuration. However, this thesis will mainly focus on the mechanical design of the massive genset system and give a brief introduction to electric propulsion in general. The main objective is to propose a design of the frame and canopy and perform a dynamic analysis to evaluate the impact of vibration. Thereby this work will be limited to the following points.

- Propose a design of canopy and frame.
- Consider mounting of genset and analyze vibration excitation from reciprocating machinery.
- Dynamic stress analysis for frame module.
- Discuss the connection method between auxiliary system and frame module.
- Propose a layout for the mechanical design.
-

1.3 MOTIVATIONS

The increased focus on reducing cost and develop methods to reduce emission, has forced companies to consider new methods and technology to reduce fuel consumption and increase energy efficiency. This thesis aims to develop a mechanical design of a diesel electric propulsion system, with multiple genset connected in parallel. For vessels with large variation in power demand this can lead to reduced fuel consumption. In a pre-study for this thesis it was develop a comparison in SFOC between 4 gensets from Wartzila W9L2 and 40 Scania SG280. The result is shown in Figure 1. It shows that such a system has especially superior performance at low load compared to more conventional diesel electric systems. Because of the promising result, it is desirable to look further into this system, with focus on mechanical design and how such a system can be constructed.

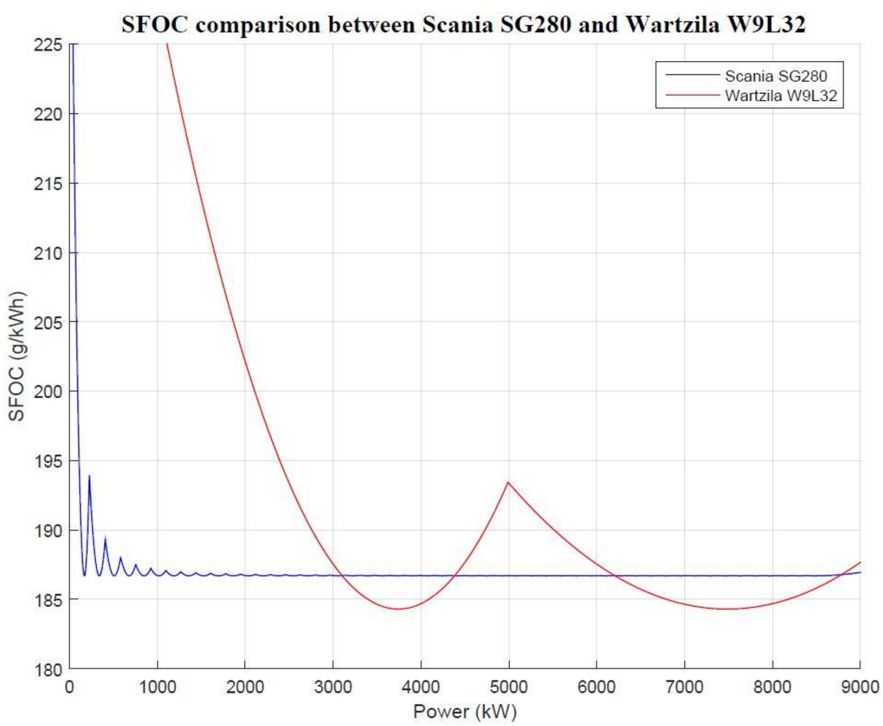


Figure 1 - SFOC comparison between Scania SG280 and Wartsila W9L32.

1.4 PREVIOUS WORK

Marine electric power systems have gone through a tremendous development during the last decades and become commonly used in many marine applications. Recently (Skjong et al., 2015) and (Skjong et al., 2016) presented the history and challenges for marine electric power systems through two papers.

There has been presented multiple simulation models of diesel engines and electric propulsion systems with different simulation tools. Among them are (Pedersen and Engja, 2000) that used bond graph method to model an diesel engine for transient performance. Later (Bruun, 2010) used a similar approach to model fuel cells and investigate fuel optimization. A introduction for components and functions of electric propulsion systems was given by (Ådnanes, 2003). (Radan, 2008) presented a model for control of marine electric power system and later (Pedersen, 2009) developed a bond graph library for marine electric systems.

For dynamic simulation of the rigid body of the diesel engine motion, the Lagrange method is of high relevance. (Ginsberg, 1995) gives a good understanding of rigid body dynamics, and include both applied theory and fundamental knowledge. In addition the book gives a good introduction to moving frame and relative motion. (Pedersen and Engja, 2014) present a method for linking the bond graph method and Lagrange mechanics through IC-field modeling. This gives a powerful tool for implementation and simulation of advanced dynamics.

A lot of work has been done within the topic of machinery foundation and mounting systems. An introduction to problems related to marine machinery foundation is presented by (R.M. Cashman, 1962). A more mathematical approach for mounting optimization and minimizing dynamic loads was performed by (Tao et al., 2000). The paper presents a four-stroke one cylinder engine modulated as a rigid body and analyze the transmission force and natural frequency of the engine. Another work that consider optimization of engine mounting is the doctoral thesis of (Alkhatib, 2013). By implementing different techniques, he evaluates load transmission between the engine and frame.

Even though the problem introduced in this thesis is a new concept, there is much to learn and use from the earlier work done within the topics of vibration, engine and electric modulation. The idea is to use existing knowledge to develop a new system configuration that can increase efficiency of marine vessel operation and reduce the global footprint form vessel machinery.

1.5 ORGANIZATION OF THESIS

The thesis structure and a brief introduction to each chapter is presented in the list below.

Chapter 2 introduces a short presentation of electric power system and how different techniques is used to improve efficiency of marine energy supply. Further it presents the initial sketch and idea behind the massive genset module. A brief discussion introduces some of the benefits and problems in the design process of such a system

Chapter 3 presents the background knowledge of dynamics, rules and regulation and engine operation. This knowledge gives the background for modulation of diesel engine and mounting system.

Chapter 4 apply the theory to construct a simulation model of a diesel engine mounting unit based on rigid body dynamics. Furthermore, two designs of the genset frame made in SolidWorks for simulation of vibration performances are shown.

Chapter 5 display results from the model simulation.

Chapter 6 discusses the result and compare performance to classification rules. A method for connection between exhaust pipe and auxiliary system is presented. Finally, a vessel engine room is sketched.

Chapter 7 conclusion and suggestions for further work that can be done on the subject.

2 SYSTEM PRESENTATION

2.1 INTRODUCTION TO DIESEL ELECTRIC PROPULSION TECHNOLOGY

The development of marine electric systems is not a new idea. The first commercial vessel that had installed electricity was SS Colombia back in 1880 (Skjong et al., 2015). The development of marine electric systems is propelled by the desire to have reliable supply of electric power and reduce the fuel consumption. However, it was not until the early 80s that the use of marine electric systems became a more common method. New technology with variable speed drive for electrical motors and development of complex control systems has made the marine electric power system more efficient (Ådnanes, 2003). Also, the introduction of dynamic positioning (DP) with high requirement for fast acting power and large load variation combined with the requirement for more efficient fuel consumption pushes the development toward marine electric systems.

2.1.1 USE OF DIESEL ELECTRIC SYSTEMS

Diesel electric power systems are used in different vessel types, with various applications. Vessels with large variation in power demand, requires good maneuverability, have large hotel load, strict requirement to noise, vibrations and reliability. Regarding all these factors it can be beneficial to install a diesel electric power system. Vessels with large variation in load, can reduce the fuel consumption with up to 30 – 40 % when installing diesel electric system compared to conventional shaft propulsion. (Ådnanes, 2003)

For cruise ships, RoRo-ferries and other passenger vessels, the diesel electric power system is frequently used. One of the reasons for this is the strict requirement regarding noise and vibration. By removing the propeller shaft and install an electric system, vibration and noise can be reduced significantly. Another noteworthy factor that has boosted use of electric propulsion systems is the increased focus on emissions, with stricter regulations and emission tax in ports. Electric system can reduce the fuel consumption and emission significantly for vessels operating at variating load. Electric propulsion can also increase maneuverability in ports when using podded propulsion. The latest development in electric propulsion is construction of All-electric ship (AES) wich only uses batteries. The first AES build is the car-ferry MF Ampere, however the power density in batteries is rather low, and the battery technology would need further development before it can be an alternative on large cruise ships and long distance voyage. (Skjong et al., 2016)

Oil and gas drilling units and offshore vessels most commonly use diesel electric propulsion systems. The high requirement of redundancy, maneuverability, DP and power consuming equipment are among the reasons for this. Diesel electric power systems give fast response, and combined with podded propulsion it allows god maneuverability in the harsh environments that offshore vessels operate in. Vessel operation close to offshore platforms also has strict requirement for available power, when a blackout can lead to catastrophic consequences, by installing diesel electric propulsion with redundancy the chances of total blackout can be minimized. Other vessels with applications within icebreaking, research, artic operation or navy, are of different reasons also most regularly built with diesel electric propulsion systems. (Ådnanes, 2003)

2.1.2 DESCRIPTION OF DIESEL ELECTRIC SYSTEMS

In the modern vessels, the requirements for available electricity is enormous. Electricity is used for accommodation, pumps, navigation, control system and main propulsion system, among other components. For vessels with a large variation of components that requires electric power and the load demand has large variations, electric propulsion system is the preferred machinery installation.

The source of power in the electric propulsion system is the prime mover that runs the generators. The most commonly used machine as prime mover is a diesel engine that operate on MDO or HFO. The reason diesel engines have been the preferred choice is because of the relatively high efficiency, robustness, maintainability and the large variation in power output according to application. However, the stricter regulations on emissions has pushed the development toward use of LNG. Duel fuel engines and gas turbines can reduce the production of NO_x and SO_x significantly.

Figure 2 shows a single line diagram of main components in a diesel electric propulsion system. The system consists of a prime mover, that produces rotational force, that is transferred to a synchronous generator through a short shaft. The synchronous generator produces power that is supplied to the switch board. Depending on the required redundancy, the switchboard can be divided into separate parts with circuit breakers in between. Because of the different voltage required from components, a transformer is used to set voltage for the distributed power. A vessel normally has two different voltage levels available and they are categorized as low voltage (LV) and high voltage (HV). LV is used by accommodation, HV is used by propulsion system, pumps, cranes and similar equipment.

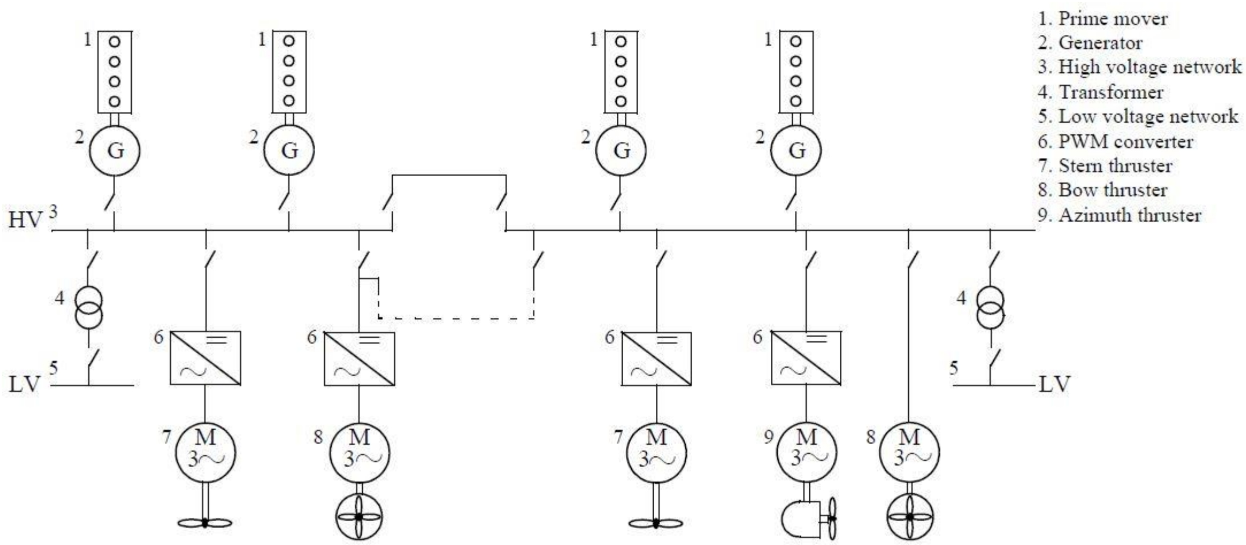


Figure 2 - Diesel electric propulsion system single line diagram. (Pedersen, 2009)

For electric propulsion system in ships it is necessary to have a control system to ensure safe and cost-efficient operation. Such a system can be a power management system (PMS). The PMS is the vessel's power coordinator that is governing the engine and generator. PMS has multiple functions that are put together in a common system to ensure available power, distribution of power, prevent system blackout and in case of failure the PMS shall restore power in a brief time. All these factors shall secure safe operation, decrease the maintenance requirement and optimize fuel consumption.(Foss, 2000)

Some of the disadvantages and advantages with an electric propulsion system are summarized in the list below, and are adopted from (Ådnanes, 2003) and (MAN B&W).

Advantages:

- With multiple gensets it is possible to optimize load on each diesel engine so that they are operating close to peak efficiency. This is done by switching off and on generators according to variations in power demand. By applying this it is possible to reduce the fuel consumption and emission.
- With a high number of gensets installed, the redundancy of the vessel is increased significantly. By breaking up into multiple switch boards with bus-breakers between the switch boards, the vessel can avoid blackout with single failure.

- When operating the diesel engine at optimal load the wear and tear is minimized. This leads to less maintenance, and low life cycle costs of the engine.
- Diesel electric propulsion systems supplies the motors through cables, and no shaft is needed between the engine and propeller, this reduces the noise and vibration.
- Position of propulsion units are not dependent on the location of machinery room, because of cables instead of shaft the machinery room does not need to be close to the propulsion units.
- Electric propulsion systems allow use of podded propulsion and thrusters, this increases the maneuverability of the vessel compared with propeller connected to shaft.

Disadvantages:

- Electric propulsion system consists of a high number of advanced components; this gives a higher investment cost compared with conventional shaft propulsion.
- The advanced control and complexity of the system demand more knowledge to operate and maintain. This puts higher demand to training and education of operation personnel.
- The high number of components between the prime mover and propeller shaft increase the loss of power compared with a propeller connected to a shaft. This loss can be up to 10 % higher. (Ådnanes, 2003)

2.1.3 ELECTRIC PROPULSION TECHNOLOGY

The most commonly used power system in modern vessels has been the AC-grid system. However, recent technology and development of power electronics allowed more efficient use of DC grid systems onboard vessels. Both *Siemens* (blue drive C) and *ABB* (DC-grid) have developed systems that use DC supply grid instead of AC-grid. One of the main advantages with a DC-grid system is that the frequency is not required to stay fixed at 60 Hz, this makes it possible to run the prime mover at variable speed. This means that it would be possible to optimize speed and allow significantly fuel savings. Optimal operation of prime mover can also reduce maintenance requirement. By implementing DC-grid, it would be possible to reduce the foot print of the electric system because of less components. (Siemens, 2016), (Skjong et al., 2016) and (Hansen et al., 2011)

Both Man Diesel & Turbo and Rolls Royce has developed diesel electric systems with a hybrid shaft solution. Figure 3 shows a single line diagram of the hybrid propulsion from MAN. The hybrid solution allows the propeller to be driven directly through a shaft from a diesel engine, or combine with power supplied from the auxiliary machinery as shown. A reduction gear that is connected between the shaft engine and main switchboard allows the diesel engine to operate with variable speed and still supply AC current at constant frequency. This configuration makes it possible to operate the system as a shaft propulsion system in transit with high efficiency. However, it is possible to supply the grid or shut down the main engine when the power requirement is reduced. The aim for this configuration is to operate the available diesel engines at optimal speed and reduce the fuel consumption. Shaft propulsion would also increase the efficiency during transit compared with a diesel electric propulsion system. (MAN Diesel & Turbo) and (Rolls-Royce, 2010)

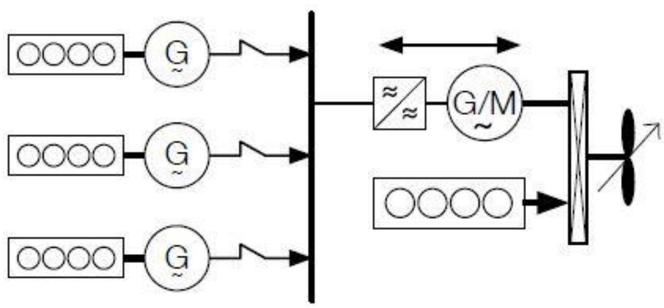


Figure 3 - Hybrid propulsion (MAN Diesel & Turbo)

Hybrid installation with combined battery and diesel electric power production, called battery energy storage system (BESS) has received much attention the last years. DNV GL made a guideline for installation of large battery systems and actively promoted battery installations. The idea behind the BESS system is to store electric power in a battery for use at sudden load changes. This will allow the generators to operate at constant load while the battery supply the “peak” power. Using batteries can reduce fuel consumption because it allows generators to operate at a optimal load. The battery can also be used as backup while new generators is synchronized to the grid. However, there are some drawback with the batteries. The power density is low compared with diesel fuel, and a battery system would require a large footprint. Batteries also generate a lot of heat, and the lifetime of the batteries can be short. (Skjong et al., 2016) and (DNV GL, 2013a)

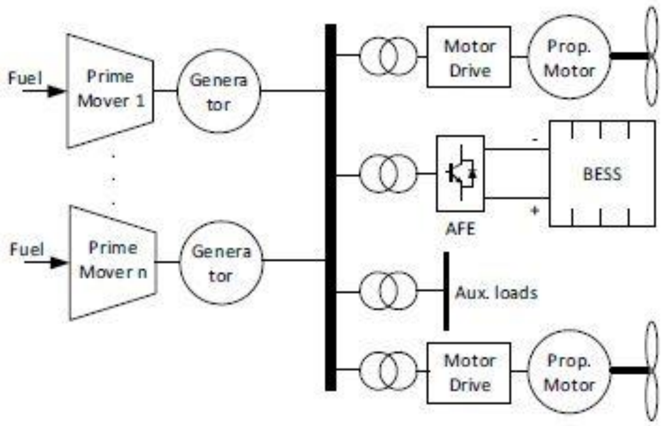


Figure 4 - Electric propulsion with BESS (Skjong et al., 2016)

Duel fuel (DF) engines and use of LNG as energy source has increased the last years. Implementation of Tier 3 and ECA have forced engine manufactures to focus on fuel types that contain less Sulphur and produce less NO_x during combustion, to comply with the new regulations. DF engines can run on a combination of LNG and FO or switch between the different fuel types. By running on pure LNG, emission of NO_x can be reduced from 40 to 85 % depending on application, and 100 % for SO_x emissions. (Adams, 2015)

2.2 SYSTEM IDEA

As mentioned in the introduction, this report should prospect a specific genset-module design that can consist of up to 50 gensets. This chapter is going to introduce the basic description of the system and how to proceed to solve some of the problems regarding installation and design.

The thought behind the massive genset-module system is to install a flexible power supply that can provide a vessel with sufficient energy at a wide specter of loads. Each generator should be in the range of 150 – 400 kW. This gives a maximum possible power output of 20 000 kW. If this is compared to data from Table 1 it shows that the genset-module can reach a broad specter of vessel types, by changing numbers and power outputs for the generators.

Table 1 - Power installed different vessels. (Skipsrevyen)

Vessel name	Vessel operation	DW[t]	Installed power[kW]
M/S Polarsyssel	PSV	3 700	6 566
M/S Whit Harvest	Fish food transport	3 300	3 570
M/S Bourbon Artic	AHTS	4 129	20 640
M/S Brage Supplier	PSV	4 800	6 408
M/F Korsfjord	Ferry	2 970	3 000
M/V Namsos	LFC	3 650	3 000

The installation would mainly consist of two parts, the genset-module shown in Figure 5, and a stationary stacking frame with multiple module slots shown in Figure 6. The genset-module shall be placed in a canopy that include connection possibilities for piping and cables at the rear. This configuration allows simple plug in plug out of each canopy, such that each genset can be removed for maintenance. A lifting hook at the top of each canopy, combined with a rolling ledge mechanism, makes it possible to slide each canopy out of the frame. This is intended to make lifting and relocation of each canopy fast and simple.

Figure 5 – Genset canopy

The stacking frame can be pictured as a frame with several racks where it is simple to add or remove units. Each module slot will be equipped with an integrated auxiliary supply system for connection to a common supply system. The idea is to have all the required connections available in a panel for each slot, this allows simple plug in for each genset-module.

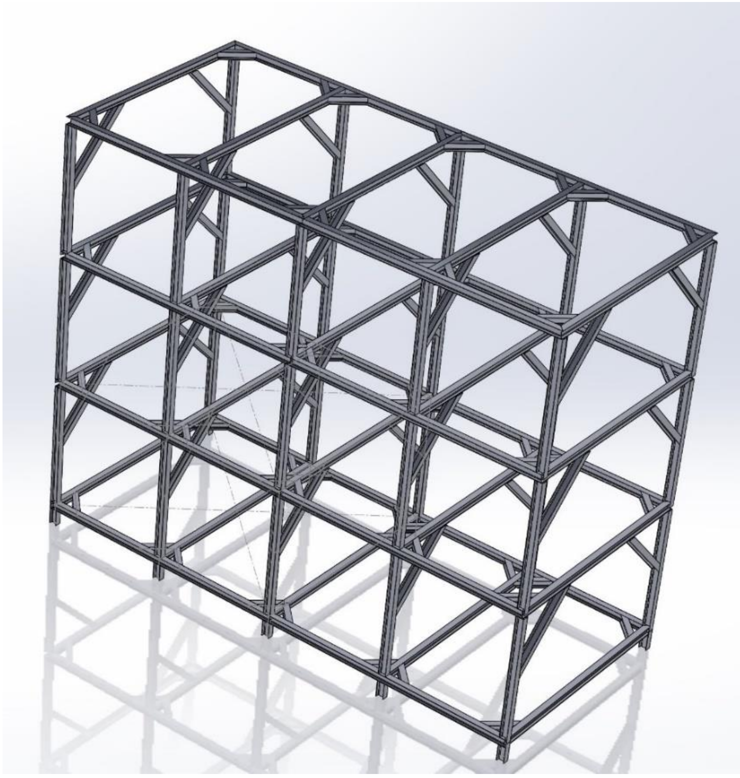


Figure 6 – Stationary frame with multiple geneset slots.

Figure 7 show a single line diagram of one plane of the frame. The figure illustrates how each genset is connected to the common grid for supply and control. By connecting the grid together, the total length of pipe can be reduced and pumping work for supply can be optimized. This can reduce both installation and operation cost. However, it would be important to secure the redundancy of the system in case of failure.

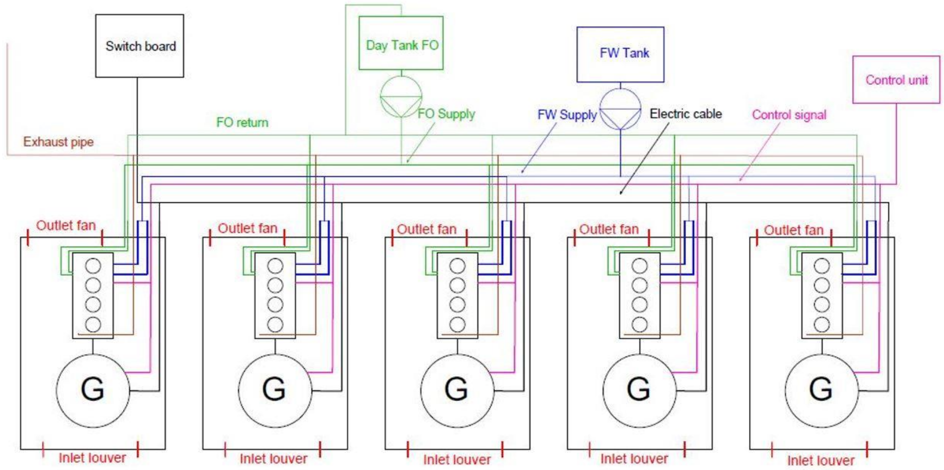


Figure 7 - Single line diagram for genset connection.

2.3 DESIGN ISSUES

The configuration of multiple gensets would require new thinking and offers some challenges that a conventional diesel electric system avoids. On the other hand, this arrangement can offer some benefits that makes it interesting to investigate how it is possible to construct and operate such a system. In designing a vessels machinery system there are mainly two categories that should be considered. The first part is regarding control and power supply, the second part is the mechanical installation including foundation and configuration of auxiliary systems.

2.3.1 GENSET CONTROL SYSTEM

Control and governing of a massive genset system where up to 50 generators are working together can offer some problems. The most commonly used method in vessels these days is to share load between the generators by droop control (Skjong et al., 2016). However, when 50 generators are working together, a droop mode control can be to complex and lead to large frequency variation in the system. This can become a large problem when new generators are synchronized and connected to the grid. Another method that is used is combined isochronous and droop mode operation. Isochronous operation keeps the frequency constant by having one generator taking all the load during sudden operation changes, then distribute the load on the generators operating in droop mode.

If the system should become a success it is critical to have a control system that allows efficient operation and reduces wear on the equipment. It is desirable to run as few gensets as possible at all time, on the other hand it is important to have available power in case of sudden load changes. New techniques that has been introduced, such as BESS can provide backup energy during spin up of genset or during small load variations. To realize the prospected cost savings, it would be required to perform simulations and analyze the performance of different configurations. Since this work would focus on the mechanical design, this problem will be dealt with in later projects.

2.3.2 MECHANICAL DESIGN

When proposing a mechanical design of a vessels machinery system there are many factors to consider. Some of the main concerns are engine mounting, vibration, engine room layout and pipe configuration. Class societies propose many rules and guidelines regarding these problems and are a good place to find information. The international maritime organization (IMO) are together with International standard organization (ISO) responsible for making rules and standards for navigation and ship building. However, can the contractor specify certain class rules that the vessel should be built after.

Concerning the design of the massive genset-module system, the structural construction of the frame should be analyzed in such a way that it can withstand the stress produced from the genset operation. It would also be important to find a satisfactory solution for auxiliary supply and a connection method that secure efficient operation and allow simple disconnection for maintenance.

3 THEORY

This chapter presents background knowledge required to perform the structure analysis of the frame system, there are mainly three tools that are going to be used. These are Lagrange equation of motion, bond graph and multi-body simulation with SolidWorks.

3.1 DIESEL ENGINE

Diesel engines have been the first-choice workhorse in marine propulsion system for decades and is believed to hold this position for many years to come. Abilities such as high efficiency, toughness, minimal maintenance cost, large operation area and relative compact design made the diesel engine the preferred prime mover in marine vessels.

3.1.1 WORKING PRINCIPLE

Diesel engines is also called compression ignition engines (CI), this is because air is compressed before fuel is injected into the hot compressed air. This lead to auto ignition of the fuel air mixture. The main principle of a diesel engine is to convert thermal energy released by combustion of fuel to mechanical work. Diesel engines are mainly distinguished between two-stroke or four-stroke engines. Two-stroke operation cycle is mainly used for large low speed engines used in vessels within transit operation. The advantage of two-stroke engines is higher efficiency that can come up to 55 % with a heat recovery system. (Tschoke et al., 2010) A two-stroke engine can theoretical produce twice the power output compared with the four-stroke engine because it produces a power stroke on every revolution. In practices this is not possible because the gas exchange process is less efficient for a two-stroke engine.

When choosing engine for power production it is most common to choose four-stroke medium speed diesel engines. The main advantages compared with large two-stroke engine is the low weight-to-power ratio, the operation speed is also more suited for generator operation and four stroke-engines allow better emission control.

Figure 8 show the 4 different steps in a four-stroke operating cycle, intake, compression, power and exhaust stroke respectively. A presentation of the 4 steps follows (Heywood, 1988):

- The intake stroke start at TDC where fresh air is drawn into the cylinder while the piston moves toward BDC. The inlet valve is opened a bit before TDC and is open through the complete intake stroke, this is to maximize the fresh air drawn into the cylinder.

- The compression stroke starts when the inlet valve is closed and the piston start to move toward TDC. The fresh air charged into the cylinder is compressed, typical compression ratio in diesel engines are between 12 and 24. At the end of the compression stroke, fuel is injected and ignited due to the high pressure and temperatures above the auto-ignition properties of the fuel.
- The power stroke is produced by the high-temperature and high-pressure gas in the cylinder, which forces the piston toward BDC. This produces a rotational force on the crank shaft that gives output work from the engine. When the piston is close to BDC the exhaust valve opens and the pressure in the cylinder decreases.
- The exhaust stroke starts when the exhaust valve opens and the piston moves toward the TDC, the exhaust gas is forced out of the exhaust valve due to the pressure difference over the valve and the piston motion toward TDC.

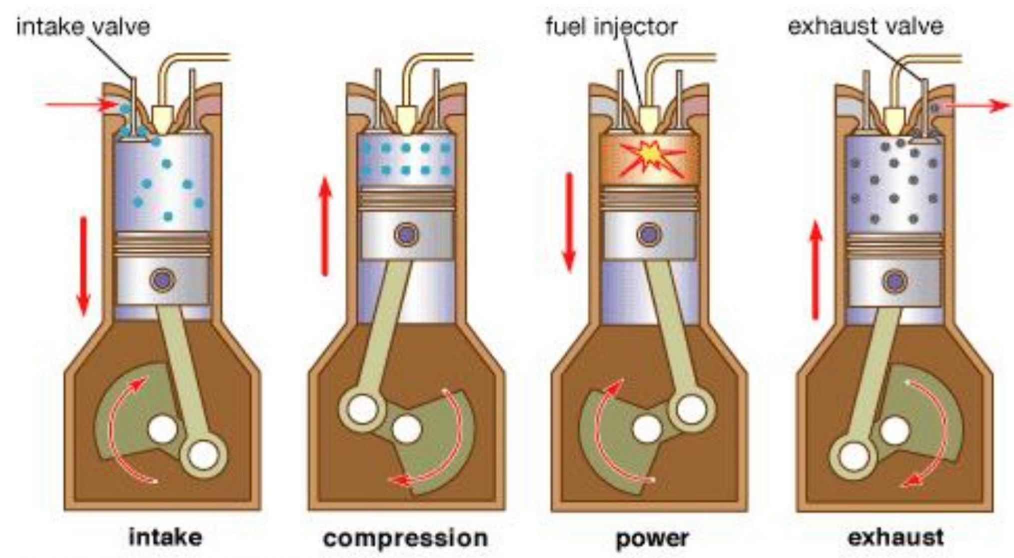


Figure 8 – Operation cycle for four-stroke diesel engine.

3.1.2 COMBUSTION

Combustion in CI-engines occurs when high pressure fuel is injected in the compressed air. The most commonly used injection method in modern CI-engines is common rail injection system with injection pressure up to 2000 bar. The most used method to present energy produced in combustion is with rate of heat release (ROHR) this presents energy released for each crank angle based on change in cylinder pressure and volume. The combustion process can be defined by 4 different steps shown in Figure 9. The first step is called ignition delay period. When injection starts, the high pressure injected fuel starts to evaporate and mixes with the compressed air inside the cylinder. When the mixture temperature is high enough the mixture auto ignites, supercharged engines commonly delay between 0.3 and 0.8 ms. Because of the large amount of fuel air mixture that burn simultaneously this leads to a rapid increase in the ROHR curve as shown in Figure 9. This is called premixed combustion phase. Since the fuel is continued to be injected the combustion is stabilized as the evaporation and fuel-air mixing process continue until the injection stop and most of the fuel is burned. This is called the mixing control combustion phase. After the injection stops and most of the fuel is burned there are some particulates of soot and fuel-rich material that are burned during the expansion stroke, this is called late combustion phase. (Tschoke et al., 2010)

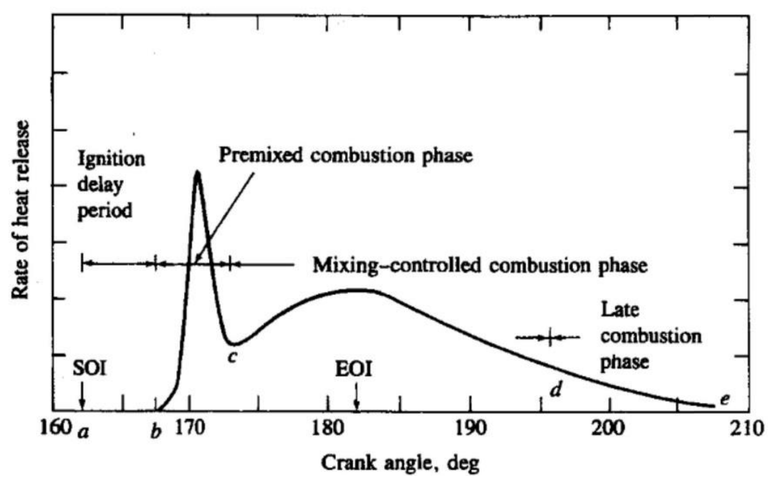


Figure 9 – ROHR curve for combustion in CI-engine. (Heywood, 1988)

3.1.3 ENGINE CRANKSHAFT ASSEMBLY

The engine reciprocating components consists of a piston, piston pin, connection rod, crankshaft and flywheel, this is also called the crank assembly of the engine. The main attribute of the crank assembly is to convert thermal energy produced when burning fuel to rotational mechanical force. Design of the crank assembly is a complex task, this is because it need to withstanding both the large gas forces acting on the piston and the tremendous mass acceleration forces from the reciprocating motion. On the other hand, it is desirable to keep the mass as low as possible and have high stiffness. (Tschoke et al., 2010)

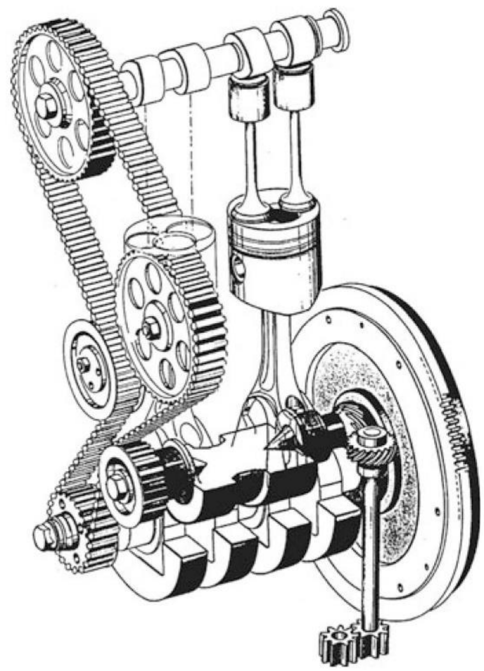


Figure 10 – Engine crank-mechanism (Tschoke et al., 2010)

The moving masses in the crank mechanism, that is operating at high speed, produce a vibration and acoustic problems that should be minimized. High gas pressure and temperature combined with the high operation speed put the crank mechanism under tremendous stress. Because of this, the requirement to material and design is a critical factor. To be able to reduce engine vibration, the mass of the piston should be as low as possible. Nevertheless, the material and construction method should secure that the engine does not suffer a breakdown.

The main parameters that influence the vibration from the diesel engine are: number of cylinders, engine design and cylinder configuration, cylinder pressure, firing sequence, distance between cylinders and kinematic parameters of crank shaft assembly.

3.2 FRAME STRUCTURE AND MATERIAL

When designing a frame construction, it is important to have knowledge of how vibration, material and design influence the structure. To analyze a beam construction, it is normal to perform a static or dynamic analysis to determine if the construction is according to standards and rules. This chapter will give a brief introduction to material, stress and beams.

3.2.1 BEAM SELECTION

Beams come in a large variety. To achieve a satisfactory design, selection of correct beam type is critical. The key factor when choosing beams is to consider the cross-sectional area and how this influences stress in the construction. For constructions that suffers large static or dynamic loads, I-beams are often used. This is because I-beams have suitable properties to withstand shear and bending stress. High stiffness and reduced mass is also a key factor to prevent extensive vibration problems in the structure. I-beams come in different standardized sizes and are most commonly produced in structural steel or aluminum. Data for I-beams and steel properties can be found in the appendix C.

3.2.2 STRUCTUAL STRESS

When making a design of a steel construction it is important to choose the correct construction material to limit stress and displacement in the structure. Different material gives different properties, some of the most crucial factors are the yield strength, density and elasticity modulus. Yield strength is defined as the stress where the material starts to become plasticly deformed. When plastic deformation occurs, the material would stay deformed permanently once the load is removed. By analyzing the dynamic and static load of the construction it is possible to determine if the structure would withstand the stress produced from the load. Table 2 shows DNV GL rules for steel constructions that is excited by vibration and state the maximum structure velocity to reduce risk of fatigue or deformation.

Table 2 – Maximum values for velocity in steel constructions (DNV GL, 2009)

Steel Constructions
Velocity 4 – 200 Hz 45 mm/s

3.3 VIBRATION

Vibration is a wide phenomenon that applies for all mechanical machinery. The main sources come from variation in the dynamics of the machine. This is influenced by mass, stiffness, dampening and degree of freedom of the system. The two most important parts are mass and stiffness that can be expressed by kinetic and potential energy, respectively. It is distinguished between periodic and non-periodic motion of the machinery, when the vibration comes from an unbalanced mass in a reciprocating machinery operating at constant speed, the vibration can be assumed to be periodic. With excessive uncontrolled vibration, machinery can suffer a failure or fatigue in bearings and structure. Another problem related to vibration is noise, because of these factors it is desirable to control and reduce vibration. (Varadan et al., 2006)

3.3.1 VIBRATION FUNDATMETAL

It is mainly distinguished between three different types of vibrations when talking about mechanical vibrations. Those are free vibrations, forced vibrations and self-excited vibrations. The two most common types of vibrations are free vibrations and forced vibrations.

Both free and forced vibrations occurs from motion in a mass-spring-damper system. It can be observed as periodic response that variates with time and can be expressed mathematically by the equation of motion. The main difference between forced and free vibrations is that the free vibrations comes from some initial condition or velocity that starts a motion around the equilibrium position of the mass, while the forced vibrations is excited by an external force that causes a motion.

$$m\ddot{x} + c\dot{x} + kx = \sum F_x \qquad (3\text{-}1)$$

A simple example of a free vibration system is a mass-spring system, as shown in Figure 11. By analyzing the free body diagram of the system, it is possible to recognize the variables from the equation of motion (3-1). The spring force is dependent of the position of the cart and inertia force variates with acceleration. Since there is no external force or dampening in the system this part disappears from the equation.

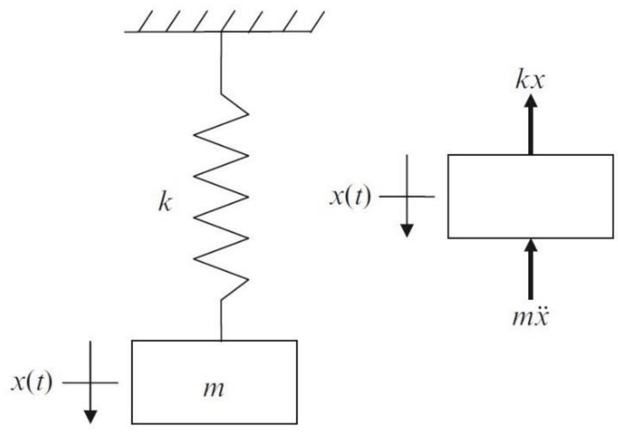


Figure 11 – Mass spring system.

By considering an undamped free vibration system it is possible to decide the natural frequency of the system. The natural frequency describes the expected oscillating motion of the mass when it is excited by an initial velocity or displacement. Since the system only is influenced by a spring and a mass, the radian frequency ω_n can be found by the following equation.

$$\omega_n = \left(\frac{k}{m}\right)^{\frac{1}{2}} \tag{3-2}$$

To find the natural frequency f_n measured in Hertz, the equation can simply be divided by 2π.

$$f_n = \frac{1}{2\pi}\left(\frac{k}{m}\right)^{\frac{1}{2}} \tag{3-3}$$

To decide the behavior of the system a function dependent on time $x(t)$ is described. If it is assumed that the motion is undamped and harmonic the position can be found through the following formula.

$$x(t) = X_1 e^{i\omega_n t} + X_2 e^{-i\omega_n t} \tag{3-4}$$

To describe the two variables the intial conditon of position $x(0)$ and intial velocity $\dot{x}(0)$ can be used. This give the following equation that describes motion of free vibration.

$$x(t) = 2Acos(\omega_n t + \beta) \tag{3-5}$$

Where $A = \sqrt{\left(\frac{x_0}{2}\right)^2 + \left(-\frac{\dot{x}_0}{2\omega_n}\right)^2}$ and $\beta = \tan^{-1}(\frac{\dot{x}_0}{\omega_n x_0})$. For a more thorough explanation on how the equations are developed, chapter 2 of the book Mechanical vibration: modeling and mesurment (Schmitz et al., 2012) can be consulted. Figure 12 shows the respon when plotting equation 3-5 with different initial conditons for position and velocity.

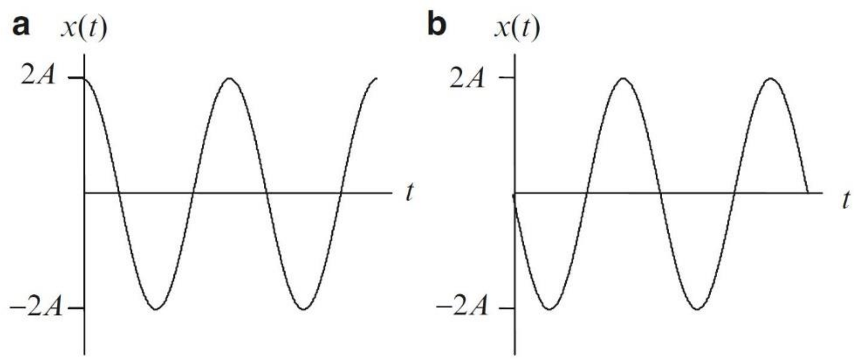


Figure 12 – Response of free vibration, a) show $x(0) \neq 0$ and $\dot{x}_0 = 0$; b) show $x(0) = 0$ and $\dot{x}_0 \neq 0$. (Schmitz et al., 2012)

An important parameter when working with dynamic systems is the dampening factor. The damper gives resistance for the relative velocity acting on it, and works as a shock absorber. The dampening coefficient c_c can be found by analyzing the behavior of the system and is dependent of mass and spring.

$$c_c = 2(km)^{\frac{1}{2}} \tag{3-6}$$

When analyzing a dampened system the dampening ratio gives a good indication of the expected behavior of the system. The dampening ratio can be found by dividing the actual dampening by the critical dampening.

$$\xi = \frac{c}{c_c} \tag{3-7}$$

The three possible outcomes of the dampening ratio are critical damped system, overdamped system or underdamped system.

- For a critical damped system $\xi = 1$

- For an overdamped system $\xi > 1$
- For an underdamped system $\xi < 1$

Since oscillating motion only occurs for underdamped systems, all vibrating systems are underdamped. When the system is dampened, the oscillating frequency changes from pure natural frequency to the dampened frequency ω_d. This can be expressed with the following equation.

$$\omega_d = \omega_n(1 - \xi^2)^{\frac{1}{2}} \tag{3-8}$$

3.3.2 VIBRATION IN RECIPROCATING MACHINERY

Engines are rotating machinery that is transferring oscillating piston motion to rotational motion by a sliding crank mechanism. The motion of the crank mechanism combined with the cylinder pressure produces inertia forces that can be transferred through the engine mounting to the surroundings. The inertia forces that are produced are based on the acceleration of masses in the crank assembly. The characteristics of the inertia forces are decided by the number of cylinders, assembly of crank shaft masses and the kinematic properties, cylinder pressure and firing sequence. (Tschoke et al., 2010) The inertia forces can cause undesirable fatigue damages to the bearings and create noise that is transferred to the hull. Because of these problems, it is desirable to minimize the inertia forces that is produced from the engine.

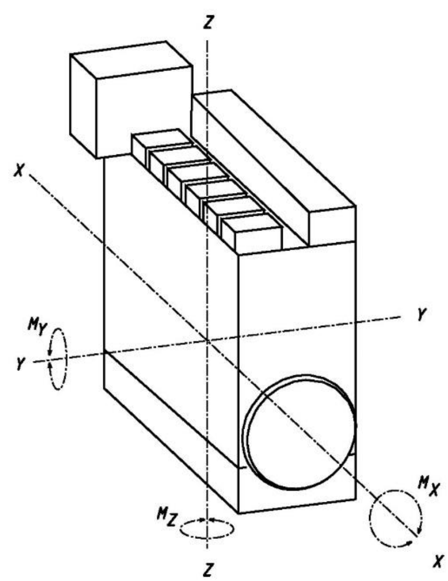


Figure 13 – Engine block coordinate system with moment of inertia. (Wartsila, 2005)

Figure 13 shows an engine block with associated axis. The inertia forces can be divided into two main categories, oscillating and rotational. The rotating inertia forces are only dependent on the rotation of the crank shaft and therefore appear only in the first order. The oscillating inertia forces are influenced by both the first and higher orders. The inertia forces for a 6 DOF engine model can be divided as follows:

$$\vec{F} = \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} \quad \vec{M} = \begin{bmatrix} M_x \\ M_y \\ M_z \end{bmatrix} \tag{3-9}$$

The oscillating and rotational forces can be found by using the equation of motion on the crank mechanism, in this thesis a combination of Lagrange and bond graph simulation will be used to develop a simulation for the forces that is produced. Each method would be described further later in this chapter. Another method that can be used is introduced in the course TMR4280 internal combustion engines compendium. (Valland, 2007) This method is based on the equation of motion and gives good results for operation at steady state. Based on this method the inertia forces for a Scania DI09 engine operating on 1800 RPM are given in Table 3.

Table 3 – Calculated inertia force and moment for Scania DI09. (source me)

Fw	Fg	Fy	Fz	Mx	My	Mz
kN	kN	kN	kN	kNm	kNm	kNm
11.28	203	0	0	-0.09	6.48	2.23

There are different methods to reduce the inertia forces produced in the engine. One method is by mass balancing of the crank mechanism. Then counterweights are used to balance the forces produced by the rotational motion. However, the balancing is optimized for operation at constant speed. All engines that are produced have to meet the balancing criteria in the ISO 21940-11 for mechanical vibration. (ISO, 2016) Another method that is used to reduce the inertia forces is to adjust the firing sequence of each cylinder, this allow to balance out the inertia forces when the acting force for the cylinders cancel each other out.

3.3.3 FREQUENCY RESPONSE

A response spectrum can be used to measure vibration in linear systems. One method is to make Fast Fourier transformation (FFT) on an input signal of displacement, velocity or acceleration during constant operation. The response spectrum gives an indication on excitation frequency from the engine operation and can be used to analyze engine vibration. According to class rules it is normal to measure the response in Root mean square (RMS) values in an area from 0 to 200 Hz. (DNV GL, 2009)

Figure 14 shows a typical response spectrum from a diesel engine. The spectrum shows multiple peak amplitudes that indicate different harmonic orders of vibration. The first order can be found by operation speed in hertz. In this case it is a diesel engine operating at 600 RPM, this gives a frequency of 10 Hz. A four-stroke engine gives response for both complete and half orders. (Tienhaara, 2004)

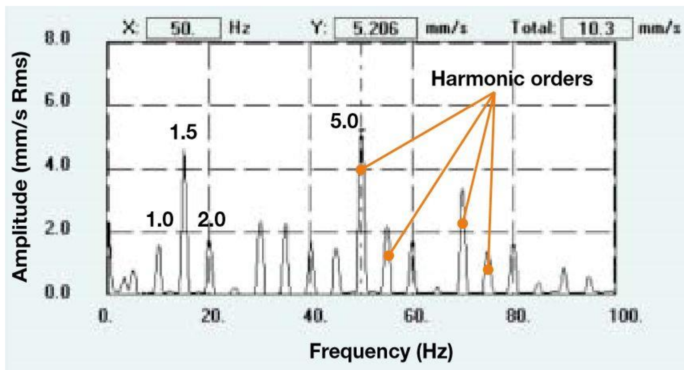


Figure 14 – Typical frequency response spectrum for 4-stroke diesel engine.(Tienhaara, 2004)

3.3.4 FUNDATION AND BEARINGS

Foundation is a crucial factor when installing a marine diesel generator. The foundation of the genset is exposed to a considerable load from both ship movement and internal motion. It is commonly distinguished between two different load classes, dynamic and static. Dynamic load mainly considers forces from the reciprocating motion, gas forces and vibrations from the hull. Static load is dependent on weight of the genset and acting motion from the seaway. By isolating the genset from the hull with a mounting system, the transmitted forces between the hull and genset can be minimized. To avoid fatigue in the bearing it is important to design the mounting system in such a way that it is capable of withstanding and that it reduces the forces. Marin diesel generators can either be rigid or resiliently mounted. Rigid mounted means that the genset is directly connected to the ship's hull. When using rigid mounts, large forces can be transferred from the genset block to the hull. The two most commonly used methods for rigid mounting are steel chocks or resin chocks. Figure 15 show an example of both a steel and resin chock. Steel chocks have been the traditional method. Nevertheless, the resin chocks have become more common over the latest years, because they are cheaper, easier to fit and give good strength characteristics. (GL, 2015, Jaroszewicz, 2004)

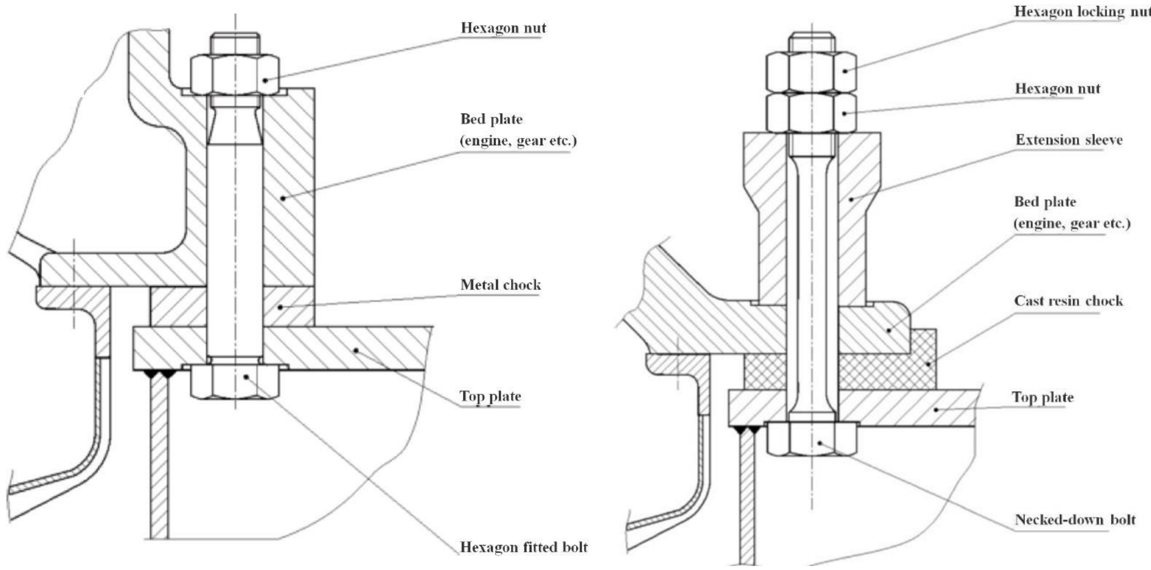


Figure 15 – Drawing of metal and resin chock mounting. (GL, 2015)

Resilient mounting is when the genset is connected to the foundation by elastic bearings. This type of mounting can be used to reduce the transfer of both noise and vibration from the machinery to the hull, and from the hull to the machinery. It is mainly distinguished between active and passive mountings. The most common resilient mounting is rubber mounts with elastic stiffness in all directions. A rubber mounting can be illustrated by a mass-spring-damper system as shown in Figure 16. It can be difficult to decide how stiff the mounting system should be designed, high stiffness can create problems at high frequencies. On the other hand the use of low stiffness can produce large vibration motion.

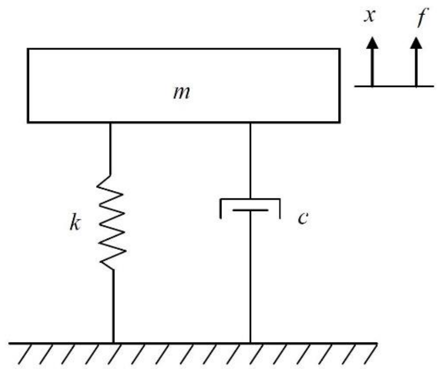


Figure 16 –Mechanical illustration of resilient mounting with elastic rubber

When mounting machinery in a marine vessel it should be according to class requirement. The vibration levels for diesel engines are most commonly judged according to the following ISO rules:

- Reciprocating machinery: ISO 10816-1 / ISO 10816-6
- Generator sets: ISO 8528-9 and ISO 10916-3

In addition, there can be specific requirements from different classification societies or engine manufactures. An example is shown in Table 4 that state the requirements given by DNV GL. This shows that the restriction on maximum velocity in the bearing of a diesel driven generator is set to 18 mm/s. (GL, 2015, Wartsila, 2005)

Table 4 - Vibration criteria for diesel driven generators (DNV GL, 2009)

Diesel driven generators and electrical motors on thrusters
Velocity 4 – 200 Hz 18 mm/s
To be measured in any direction on the bearings. Applies to both fixed and resilient mounted 1 order vibration above 7 mm/s should be investigated.

3.4 BOND GRAPH MODELING

3.4.1 INTRODUCTION TO BOND GRAPH MODELING

Bond graph was invented by professor Henry Paynter at MIT in 1959, but it was not until his students, professor Dean Karnopp and professor Ronald Rosenberg, continued development that the concept became more used for modeling.

Bond graph is a graphical method for simulation and modeling of multi-domain dynamic systems. The fact that it can be used for multi-domain systems makes bond graph a handy method to develop advanced models that can consist of system with different types of energy. The literature regarding bond graph that is used in this chapter is mainly brought from the book; Bond Graph Modelling of Engineering Systems (2011), (Karnopp et al., 2012) and (Pedersen and Engja, 2014)

3.4.2 FUNDAMETAL CONSEPT

Bond graph is a graphical modelling method that allows for interaction between multiple systems with different energy domains. Each sub system interacts and exchanges energy with each other through power ports and signal ports. The sub systems are built up by different basic elements that either storage, dissipate or supply energy. To transfer energy between the sub system they are connected with power bonds, which allows for interaction between the systems. Since power interaction between two elements cannot be described with only one variable, the power bond is divided into two variables called power variables. These two variables are defined as effort $e(t)$ and flow $f(t)$ and are expressed by the following relation (Pedersen and Engja, 2014):

$$P(t) = e(t) * f(t) \tag{3-10}$$

Figure 17 show how two sub systems are connected with power bonds, where flow and effort are transferred between the two sub systems. If the system is mechanical the effort represents force or torque and the flow represents angular or linear velocity.

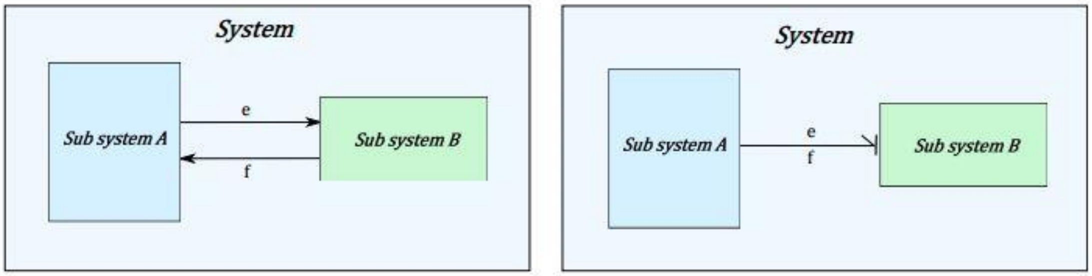


Figure 17 – Power bond interaction between two sub systems (Rokseth, 2014)

Besides effort and flow two extra variables are defined that are useful when describing the interaction between two systems. Those are momentum $p(t)$ and displacement $q(t)$, these variables are expressed as the integral of effort and flow.

$$
\begin{aligned}
p(t) &= \int_0^t e(t)dt + p(0) \\
q(t) &= \int_0^t f(t)dt + q(0)
\end{aligned}
\tag{3-11}
$$

In a mechanical system, momentum has the unit as linear momentum or angular momentum, while displacement is in distance or angle as shown in Table 5. Bond graphs contain nine different elements. These elements have either the ability of storing, dissipating, suppling or transferring energy. Effort source, S_e and flow source, S_f are the two elements that are a source of energy in the system, they supply external energy into the system. There are two elements that are considered as ideal and have the ability to store or deliver energy. Mechanical domain energy, can mainly be stored as either potential or kinetic energy. The two elements are called capacitor element, C and inertia element, I. The final element in bond graph is used to simulate dissipation of energy. This is characterized by a static relationship between effort and flow and is expressed as a resistor, R.

Table 5 – Variable identification in mechanical domain. (Pedersen and Engja, 2014)

Energy domain	Effort (e)	Flow (f)
Mechanical Translation	Force (N)	Velocity [m/s]
Mechanical rotation	Torque [Nm]	Angular velocity [rad/s]
Energy domain	**Momentum (p)**	**Displacement (q)**
Mechanical Translation	Linear momentum [kgm/s]	Distance [m]
Mechanical rotation	Angular momentum [Nms]	Angle [rad]

3.5 LAGRANGE MECHANICS

This section will present the Lagrange equation of motion. The Lagrange equation can be used for simulations of multi body rigid systems. It is based on the use of internal energy of the system and utilizes the kinetic and potential energy of the system. The kinetic energy is described by the generalized coordinate q, and the time derivative of the generalized coordinate $\dot{q}$. While the potential energy can be described solely by the generalized coordinate. All external forces acting on the system are described by the generalized forces. Since the understanding of generalized displacement and forces is important to describe Lagrange equation of motion, they will be presented before developing the equation of motion. The knowledge of Lagrange mechanics is adapted from (Ginsberg, 1995) and MIT web course (Vandiver, 2011).

3.5.1 GENERALIZED COORDINATES

Generalized coordinates are a set of variables that uniquely describe the system position at all times, according to a reference frame. By implementing values for the generalized coordinates, combined with the fixed values, it is possible to calculate the position of the system. The number of generalized coordinates required to describe a system is depending on the degree of freedom of the system. If the system is holonomic and the generalized coordinates is independent, it is possible to describe the system with equal numbers of generalized coordinates as degrees of freedom. If the number of degrees of freedom is two, it is possible to describe the system by two generalized coordinates. If the system is not holonomic, and the number of generalized coordinates is larger than number of degree of freedom of the system, is the number of constrain-sets equal to the difference between generalized coordinates and degrees of freedom have to be established.

The choice of generalized coordinates is not unique and there can be a distinct set of generalized coordinates that can describe the same system. An example can be a point that is moving in circle around an origin, all positions can be described by the polar coordinates, $q = [r, \theta]$. However, the same system can be described in the cartesian system by $q = [x, y]$. The velocity of the system can be found by taking the time derivative of the generalized coordinates $\dot{q}$.

3.5.2 GENERALIZED FORCE

Generalized forces describe the external forces that are acting on the system. To describe all the forces acting on the system the expression virtual work is used. Virtual work is a summation of all the forces acting on the system and the virtual displacement for each force δr. Virtual displacement is the arm from the center of the reference frame to the point where the force is acting. Virtual work can be described by the following equation (Ginsberg, 1995).

$$Q_j = \sum \vec{F}_j * \frac{\partial \vec{r}_j}{\partial q_j} \tag{3-12}$$

3.5.3 DEVELOPING LAGRANGE EQUATION OF MOTION

How the Lagrange is developed is described in (Ginsberg, 1995) and a short description is written in this section. Change of potential energy in the system can be described solely on the general coordinates of each mass q_j and can be expressed as.

$$\delta V = \sum_{j=1}^{M} \frac{\partial V}{\partial q_j} \delta q_j \tag{3-13}$$

The kinetic energy can be described by the velocity and generalized coordinates of each mass. Velocity for each mass can be found by taking the time derivative of the general coordinates $\dot{q}_j$. Kinetic energy can then be expressed by the following equation.

$$\delta T = \sum_{j=1}^{M} \left(\frac{\partial T}{\partial q_j}\, \delta q_j + \frac{\partial T}{\partial \dot{q}_j} \delta \dot{q}_j\right) \tag{3-14}$$

External forces acting on the system can be introduced through virtual work that is dependent on the general coordinates.

$$\delta W = \sum_{j=1}^{M} Q_j\, \delta q_j \tag{3-15}$$

By coupling the kinetic and potential energy together with Lagrange, $L = T - V$, if additionally the external energy acting on the system is includes the following equation can be established.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right) - \frac{\partial T}{\partial q_j} + \frac{\partial V}{\partial q_j} = Q_j \tag{3-16}$$

3.5.4 LAGRANGIAN IC-FIELD MODELLING

When working with bond graph modeling the system can become large and complex when multiple systems are interacting with each other. When the system grows, the number of algebraic loops and differential causalities grow which makes the simulation slow. For modeling of mechanical systems, the Lagrange equation can be implemented into a IC-field to reduce the complexity and the problem with differential causalities are eliminated (Pedersen and Engja, 2014).

3.5.5 IMPLEMENTATION METHOD

This section will show how Lagrange can be implemented into bond graph system.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right) - \frac{\partial T}{\partial q_j} + \frac{\partial V}{\partial q_j} = Q_j \tag{3-17}$$

If kinetic $T(q, \dot{q})$ and potential $V(q)$ energy are expressed by the generalized displacement and generalized momentum it is possible to rewrite the Lagrange expression. The generalized momentum can be expressed as the following equation:

$$p = \frac{\partial T}{\partial \dot{q}_j} = B\dot{q} \tag{3-18}$$

If the equation is rearranged it gives an expression for flow $\dot{q}$.

$$\dot{q} = B^{-1}\, p \tag{3-19}$$

By combining Lagrange equation and expression for generalized momentum, the expression for effort $\dot{p}$ is found.

$$\dot{p} = \frac{\partial T}{\partial q_j} - \frac{\partial V}{\partial q_j} + Q_j \tag{3-20}$$

These leaves two equations for flow and effort respectively, that can be implemented into a IC-field when using the bond graph method.

3.6 TRANSLATION AND ROTATION OF RIGID BODY

When working with the Lagrange equation the aim is to describe the motion, velocity and acceleration of all linkages and mass centers as a function of the generalized coordinates. A useful tool for this is translation and rotational transformation. The method is based on moving reference frames where a point is observed in a moving frame, $x, y, z,$ according to a fixed reference frame, X, Y, Z. By describing the motion of the point according to the moving frame it is possible to transform the motion back to the fixed reference frame, making the position known at all times.

It will be discussed how to perform rotational and translation transformation for a body and further look into how it can be used in bond graph simulation. The literature is based on chapter 3 in (Ginsberg, 1995) and the lecture book from TMR4275 (Pedersen and Engja, 2000).

3.6.1 ROTATIONAL TRANSFORMATION

When working with rigid bodies moving in space it is important to keep control over the shifting position of the body. A powerful tool to keep track of the motion is to describe unit vectors that give the position of a point according to a fixed reference frame. When considering two coordinate systems where the rotating coordinate system x', y', z' with unit vectors i', j', k' rotate relative to a fixed coordinate system x_o, y_0, z_0 with unit vectors i_o, j_0, k_0 about the z axis. It is possible to describe the rotation (ψ) about z_0 based on the angle offset for each unit vector and get the following expression.

$$\begin{aligned} i_0 &= i' \cos(\psi) - j' \sin(\psi) \\ j_0 &= i' \sin(\psi) + j' \cos(\psi) \\ k_0 &= k' \end{aligned} \tag{3-21}$$

Since the rotation is fixed about the z axis, the unit vector about the axis is equal for the fixed and rotating frame, while the unit vector for the rotating frame change in x and y direction according to the frame rotation.

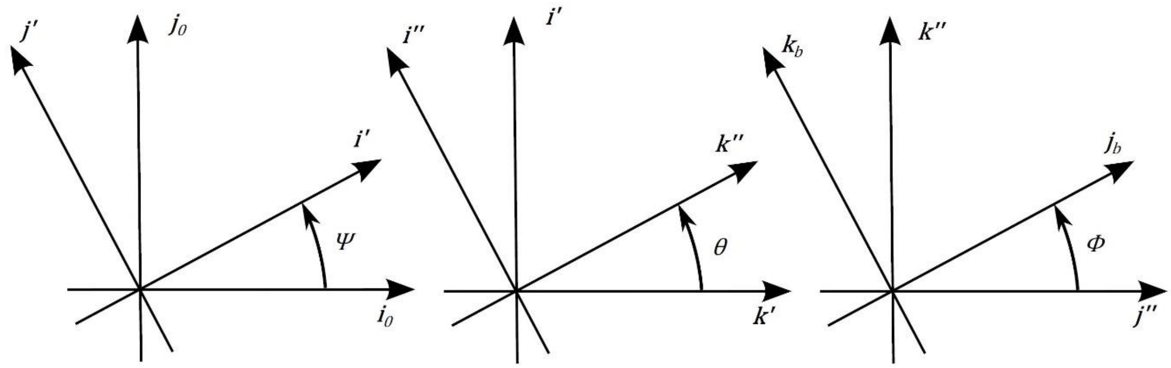


Figure 18 – frame rotation about separate axis (Rokseth, 2014)

When describing a body in three-dimensional spatial motion with rotation about two or more axes, the simplest method is to study each rotation separately. If the rotation about x_o, y_0, z_0 is expressed as it is in Figure 18, the rotating matrix according to the reference frame is expressed based on the angles $[\phi, \theta, \psi]$ rotating about a separated axis, this is called the Euler angle orientation matrix. To describe the transformation in all directions according to the reference frame it is possible to sum up each matrix about each axis according to a common reference frame, this gives the overall transformation matrix R, with the following rotational orientations. Where $c_x = \cos(x)$ and $s_x = \sin(x)$.

$$
\begin{aligned}
R_z(\psi) &= \begin{bmatrix} c_\psi & -s_\psi & 0 \\ s_\psi & c_\psi & 0 \\ 0 & 0 & 1 \end{bmatrix} \\
R_y(\theta) &= \begin{bmatrix} c_\theta & 0 & s_\theta \\ 0 & 1 & 0 \\ -s_\theta & 0 & c_\theta \end{bmatrix} \\
R_x(\phi) &= \begin{bmatrix} 1 & 0 & 0 \\ 0 & c_\phi & -s_\phi \\ 0 & s_\phi & c_\phi \end{bmatrix}
\end{aligned}
\tag{3-22}
$$

By summing up the individual rotation about each axis, as shown in Figure 18, it is possible to describe the motion in all directions according to the reference frame. First the rotation ψ is expressed according to the frame x_0, y_0, z_0 , further the rotation in angle θ is expressed based on the orientation of the x', y', z' frame and finally the last rotation ϕ about the x'', y'', z'' frame. By summing the individual rotation matrixes, it is possible to express the rotation about all axis with the following expression called Euler angles.

$$\begin{Bmatrix} x_o \\ y_o \\ z_0 \end{Bmatrix} = R \begin{Bmatrix} X \\ Y \\ Z \end{Bmatrix} \tag{3-23}$$

Another important variable that is described when using transformation according to a fixed reference frame is the translation of the position. Since the deflection in separate point can variate based on the motion of the rigid body it can be interesting to consider how the motion influences other points of the body. This can be done by introducing a skew matrix based on the offset in the separate directions a_i. For a coordinate system with a reference frame, x, y, z, the following vector can be used.

$$a = \begin{bmatrix} 0 & -a_z & a_y \\ a_z & 0 & -a_x \\ -a_y & a_x & 0 \end{bmatrix} \tag{3-24}$$

3.6.2 MULTIDIMESIONAL RIGID BODY IN BOND GRAPH

To simulate a rigid body with motion in 6 directions it is possible to put up a bond graph model. The procedure of making a bond graph model of a three-dimensional rigid body is explained in chapter 8 of the book (Pedersen and Engja, 2014). The basics of this procedure will be explained, but a more thorough explanation can be found in the book.

The aim of the bond graph model is to describe the motion of a mass moving in space because of external forces acting on the body. To describe the motion, it is necessary to find the velocities $v = [u, v, w]$ and angular velocities $\omega = [\omega_x, \omega_y, \omega_z]$ according to a fixed reference frame. To describe the velocities there are three factors that should be expressed, these are the forces/moments vector τ, mass matrix M and the Coriolis-centrifugal forces C(v).

$$M\dot{v} + C(v)v = \tau \tag{3-25}$$

The mass matrix is based on the mass of the body and center of gravity according to the refence frame, in addition to the inertia about center of gravity.

$$M = \begin{bmatrix} m & mz_g \\ -mz_g & I \end{bmatrix} \tag{3-26}$$

Where m is the mass of the body, z_g is the center of gravity in, $x, y, z,$ direction and I is the inertia matrix of the body. The Coriolis-centrifugal force describes the inertial force acting on the body because of the motion relative to the fixed coordinate system. The force vector is based on the external forces acting on the system and can be expressed as follows.

$$\tau = \left[F_x, F_y. F_z, M_x, M_y, M_z\right]^T \tag{3-27}$$

The bond graph implementation of the coordinate transformation can be done by the same method introduced in the previous chapter. By implementing the following expression of force and velocity into a MTF, it is possible to transform the velocity and force from reference frame f to the moving frame F.

$$\begin{gathered} \begin{bmatrix} v_f \\ \omega_f \end{bmatrix} = R \begin{bmatrix} v_F \\ \omega_F \end{bmatrix} \\ \begin{bmatrix} F_f \\ M_f \end{bmatrix} = R \begin{bmatrix} F_F \\ M_F \end{bmatrix} \end{gathered} \tag{3-28}$$

By combining the Euler transformation and the Lagrange method it is possible to model motion in a complex three-dimensional rigid body system according to a reference frame in a fairly simple way.

3.7 NUMERICAL SIMULATION TOOLS

When working with complex simulation and calculation it is important to use the correct computer software to achieve good results and perform an efficient analysis. There are three main softwares that have been used in this thesis. Matlab for calculation and deriving equations, 20-Sim for simulation of diesel engine and corresponding forces produced in the suspensions and SolidWorks for design sketches and dynamic load analysis of the base frame.

3.7.1 MATLAB

Matlab is a numerical computing program developed by MathWorks. It is specialized on matrix calculations and functions and work well for developing equations, analyze large data and visualize by plotting. Especially the sym function is used in this thesis, the function allow to describe symbol as input arguments. (MathWorks, 2017)

3.7.2 20-SIM

20-sim is a modeling and simulation software for simulation of multi-domain dynamic systems. The software allow simulation by implementation of equation, block diagram, physical components or bond graphs. It is possible to simulate and analyze the performance by plotting or 3-dimensional visualization. (20-Sim, 2017)

3.7.3 SOLIDWORKS

SolidWorks is a computer aided design (CAD) software for 3-dimensional design modelling developed by Dassault Systems. The program also includes a computer-aided engineering (CAE) function that allows for simulation and stress analysis of components. (Dassault-Systems, 2017)

4 SYSTEM MODULATION

The aim of this chapter is to develop a model for vibration analysis of the system. The main contributor of oscillating forces and momentums is the diesel engine. A simulation model must be developed to be able to analyze how the forces produced from the diesel engine influences the frame construction and pipe installation. The required mathematical description and modeling method for each system component will be presented.

To simplify the modulation process some assumptions must be made. Vibration produced from the generator is not included in the simulation. It is assumed that these forces are very small compared to vibration excitations from the diesel engine. The canopy is modulated as a homogenous body where the forces produced from reciprocating motion acts in the center.

4.1 MODELING OF A DIESEL ENGINE

The engine is going to be modulated as a 5-cylinder 4-stroke high speed engine that is produced by Scania. The engine is connected to a generator and produce a power output of 221 kW with frequency at 60 Hz. Figure 19 shows how the diesel engine and generator is coupled together in a common frame.

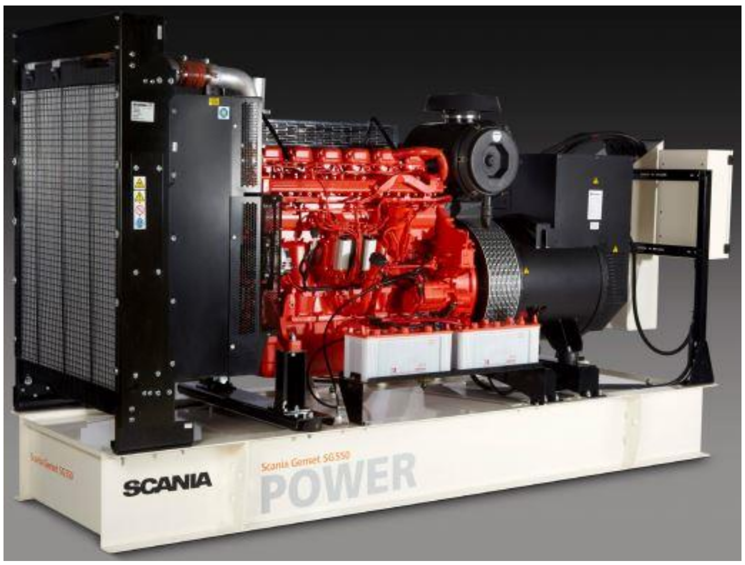


Figure 19 – Scania SG generator set. (Scania Engines, 2013)

4.1.1 INERTIA FROM ENGINE OPERATION

To simplify the task, the forces produced by the generator is neglected and the focus is only on the inertia forces produced by the Scania engine. The required data to calculate the inertia forces for the Scania DI09 engine is collected in Table 6. However, it was not possible to find mass of piston head and connection rod, so it is assumed to be 5 and 3 kg respectively.

Table 6 - Data Scania DI09 (Scania Engines, 2013)

Bore	Stroke	Connecting rod	Crank radius	Speed	Weight	Moment of inertia	Firing order
mm	mm	mm	mm	RPM	kg	kgm^2	
130	140	255	70	1800	1150	2.63	1-2-4-5-3

4.1.2 COMBUSTION

The force acting on the piston head is produced by the combustion in the cylinder. The most accurate method to simulate these forces is by producing a simulation of the ROHR based on fuel flow, air flow and heating value of the working medium. An example on a simulation that used this method in engine modeling is done by (Pedersen and Engja, 2000). This method allows for the simulation of thermodynamic reactions in the combustion process. However, since this thesis only considers the dynamic behavior of the engine, it is possible to use a simpler method by implementing the cylinder pressure and convert this to the force that is acting on the piston head. Data on cylinder pressure at different operation points are available from earlier experiments that are carried out on the Scania engine installed at the machinery laboratory for department of marine technology, and is presented in Figure 20.

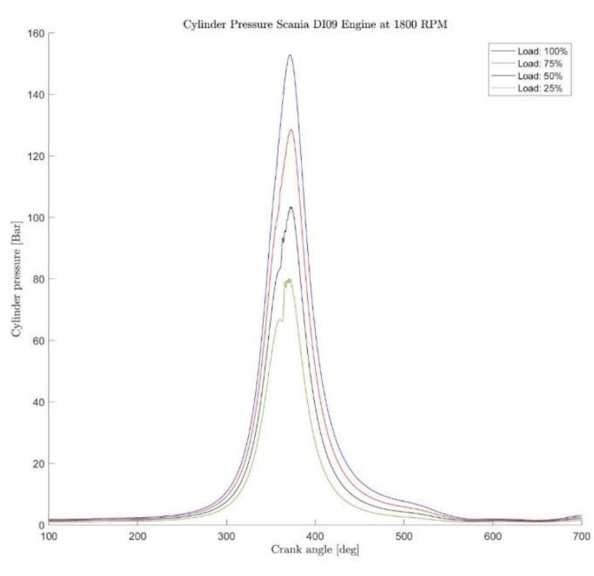


Figure 20 – Cylinder pressure data from engine test on a Scania DI09 at 1800 RPM

When using the cylinder pressure as input data, it would be required to convert the pressure to piston force and further to a twisting torque at the crank shaft. This is because simulation of the crank shaft is a function of the crank angle in the mechanical rotational domain. The pressure can be converted to a force by simply using the correlation of cylinder pressure, P_c and the piston area, A_p as shown in the following equation.

$$F = P_c * A_p \tag{4-1}$$

When introducing an external force into the system it can be implemented as a generalized force in the Lagrange equation. The twisting torque, T_t is then dependent on the force acting and the virtual displacement, δr_c that gives a variation in how the piston movement influences the system. The generalized force, Q_j is presented as a twisting torque and can be expressed with the following equation.

$$Q_j = T_t = \sum F \frac{\delta r_c}{\delta q_j} \tag{4-2}$$

The virtual displacement for a sliding crank mechanism can be found by differentiating the distance from the crank shaft to the piston head with respect to the generalized coordinate theta. Virtual displacement can be expressed in the following manner.

$$\delta r_c = -r \sin\theta \ [1 + \frac{r\cos\theta}{(l^2 - r^2 \sin^2\theta)^{1/2}}] \tag{4-3}$$

4.1.3 SLIDIER - CRANK MECHANISM

When analyzing the vibration forces produced in a diesel engine, the most important engine part is the crank mechanism. The rotational motion of the crank produces inertia forces that are transferred to the engine foundation. The function of engine sliding crank mechanism is more closely explained in chapter 3.1. To be able to simulate the engine crank mechanism, the kinematic motion for each component is described mathematically. By employing Lagrange equation of motion on the sliding crank mechanism and describing the position $[x_i, y_i]$ for each mass based on a set of generalized coordinates, q_c, it is possible to describe the kinetic and potential energy of the system. Equation (4.4) shows the Lagrange equation, the development of this equation is more closely covered in chapter 3.5.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right) - \frac{\partial T}{\partial q_j} + \frac{\partial V}{\partial q_j} = Q_j \qquad (4\text{-}4)$$

To be able to analyze the crank mechanism it is important to understand the geometric behaviors of the system. Figure 21 shows an illustration of a sliding crank mechanism including mass, position of crank, connection rod and piston. The first step in the process is to decide how many generalized coordinates that are required to describe all the positions, this is equal to the degree of freedom (DOF) for the system. A method to decide this is to study the number of components and how they are connected. Sliding crank mechanism consists of three components, this gives a total number of 9 DOF. However, there are three pined joints and one sliding body, these constraints reduces DOF for the system to one. By choosing generalized coordinates it is possible to describe the position of each component and how it varies with time. For this system, it is possible to express all positions by choosing generalized coordinates in the polar coordinate system, $q_c = [\theta]$.

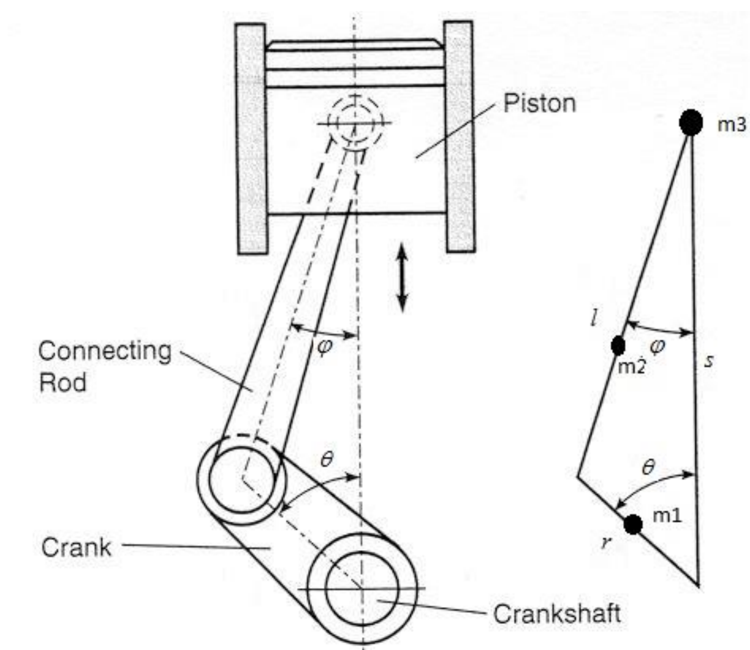


Figure 21 – Illustration of sliding crank mechanism.

The system consists of two masses and two momentums of inertia. The connection rod has both an inertia and a mass $[m_1, J_1]$, the piston head consists of only a mass, m_2 and the crank shaft consists of only an inertia, J_0. By studying the geometry of the crank mechanism, the following relation can describe the distance between piston and crank shaft.

$$S = r * \cos\theta + l\cos\varphi \tag{4-5}$$

Equation 4.5 describes the distance between crank shaft and piston based on theta and phi. However, it is desirable to describe the distance solely based on the generalized coordinate theta. When describing phi based on theta the following expression can be developed.

$$S = r * \cos\theta + \sqrt{l^2 - r^2\sin^2\theta} \tag{4-6}$$

Further the position of the mass center for the connection rod should be expressed. This can be done by taking the position of the crank rod and add the contribution from the pin connection to the mass center. Table 7 shows the final expression for all the positions based on the generalized coordinate theta.

Table 7 – Position center of mass on connection rod and piston head in x and y direction.

m	x	y
1	$\frac{l_a(s - r\cos\theta)}{l}$	$\frac{l_a * r\sin\theta}{l}$
2	$r\cos\theta + (l^2 - r^2\sin^2\theta)^{\frac{1}{2}}$	0

By differentiating the expression for position, it is possible to express the velocity for each mass.

$$\frac{dx}{dt} = \dot{x} \qquad \frac{dy}{dt} = \dot{y} \qquad \frac{d\theta}{dt} = \omega_c \tag{4-7}$$

Since the potential, V and kinetic, T energy is dependent on the position and velocity respectively, we now have all the required data to develop an expression for each of them.

$$\begin{gathered} T = \sum_i \frac{1}{2} m_i\left(\dot{x}_i^{\,2} + \dot{y}_i^{\,2}\right) + \sum_i \frac{1}{2} J_i\omega_i^2 \\ V = \sum_i m_i g y_i \end{gathered} \tag{4-8}$$

By using the expressions that are established for the sliding crank mechanism, it is possible to implement all the data into equation 4.8 and solve the Lagrange equation of motion for this system.

4.1.4 MULTIPLE CYLINDERS SYSTEM

In the previous chapter, it was explained how to mathematically simulate the motion of one sliding crank mechanism, but since the engine that is going to be simulated consists of 5-cylinders it is necessary to develop the model to give the correct output. As mentioned in chapter 3.3.2 the firing sequence and phase angle, $\delta\phi$ of each cylinder play an essential role for the intensity of the engine vibration.

For the Scania engine, operating in a four-stroke cycle with equal firing interval, the phase angle can be decided by the following expression, the firing sequence can be found in Table 6.

$$\delta\phi = \frac{720^{\circ}}{N_{cyl}} \tag{4-9}$$

To develop the Lagrange equation of motion for the combined 5-cylinders it is possible to use the same approach as in previous chapter, but the potential and kinetic energy must be summed for each cylinder with the correlated phase angle.

4.1.5 IMPLIMENTING ENGINE MODEL INTO 20-SIM

The bond graph method shall be used to model the interaction between the different systems in the diesel engine. By implementing the bond graph into the numerical computer program (20-sim) it is possible to make a dynamic simulation of the diesel engine. Figure 22 shows the finished engine model with interaction between the cylinder pressure, dynamic model of crank mechanism, with power and rotational speed as an output signal. A PID controller is used to adjust the cylinder pressure to achieve the desirable engine speed.

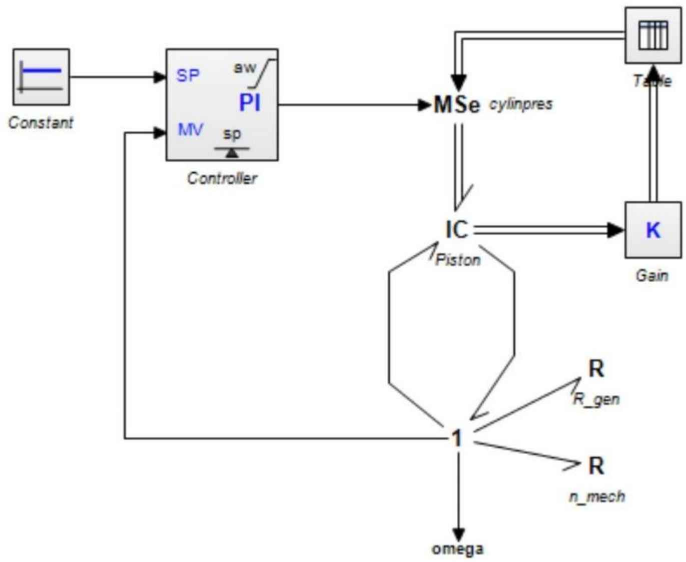


Figure 22 – Bond graph representation of Scania diesel engine

To simulate the motion and torque produced by the pistons, the Lagrange IC-field method is used. This method is more closely described in chapter 3.5.4. The IC-field called "piston" in the model is summing up the total potential and kinetic energy produced by the motion of each piston. To calculate the output effort, the Lagrange equation is used.

$$\dot{p} = \sum_i F_i * dr_i + \sum_i \frac{1}{2} m_i\left(\dot{x}_i{}^2 + \dot{y}_i{}^2\right)\frac{\partial}{\partial\theta} + \sum_i \frac{1}{2} J_i \omega_i^2 \frac{\partial}{\partial\theta} - \sum_i m_i g y_i \frac{\partial}{\partial\theta} \qquad \text{(4-10)}$$

Further the velocity of the system can be decided by summing up the inverse matrix of the differentiated kinetic energy based on the generalized velocity and the generalized momentum.

$$\dot{q} = inv\left(\sum_i \frac{1}{2} m_i\left(\dot{x}_i{}^2 + \dot{y}_i{}^2\right) \frac{\partial}{\partial\dot{\theta}} + \sum_i \frac{1}{2} J_i \omega_i^2 \frac{\partial}{\partial\dot{\theta}}\right) * int(\dot{p}) \qquad \text{(4-11)}$$

The expression for flow and effort becomes quite complex and will not be shown in their full length. To simulate the combustion, data from engine testing is used. The effort source "cylinpress" imports the cylinder pressure data based on the crank position for each cylinder. This is converted into a force that is acting on the piston head that can be recognized as F in equation (4.10). A controller that is made by C. Kleijn in September 1999 is used to scale the cylinder pressure in such a way that the speed is kept constant. The two R-elements at the crank represent the mechanical efficiency of the engine and a generator load that is attached to shaft. To calculate the generator load, the resistance coefficient, Cr is calculated based on the torque the engine must deliver to produce a given power output.

$$Cr = \frac{P}{\omega^2} \tag{4-12}$$

The speed of the crank is delivered as an output for use in the calculation of engine force and momentum from oscillating motions. How each bond graph element is modulated can be found in the digital file in appendix D.

4.1.6 FORCES FROM OSSCILATING MOTION

To modulate how the crank mechanism motion, influences the engine block, all the inertia forces and momentums that are produced must be calculated. All the cylinders are equal and produce the same inertia force and momentums, the only difference is the position at the crank shaft and the individual crank angle. The total inertia force and momentums is found by summing the inertia for each cylinder. Figure 23 shows how the engine and engine-block is connected by an effort source that calculates the inertia forces and momentums based on the engine speed.

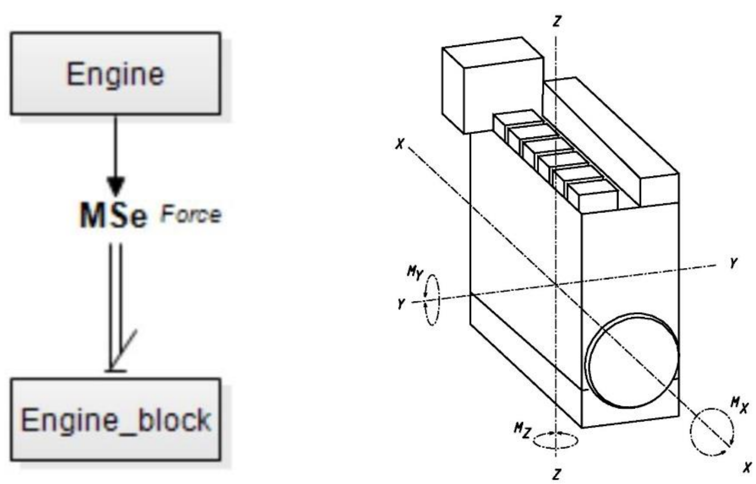


Figure 23 – Bond graph model of combined engine and engine block, engine block with corresponding coordinate system.

The output from the effort source is given as a 6x1 matrix with variation of inertia forces and momentums.

$$\tau = \left[F_x \; F_y \; F_z \; M_x \; M_y \; M_z\right]^T \tag{4-13}$$

According to the coordinate system of the engine-block, showed in Figure 23, the inertia forces can be calculated based on the rotational speed ω, crank radius r, crank angle φ, rotating mass m_r and oscillating mass m_o.

$$F_z = \sum_i \omega^2 r[(m_r + m_o) * cos\varphi_i + m_0(A_2 cos2\varphi_i - A_4 cos4\varphi_i)]$$

$$F_y = \sum_i \omega^2 r\, m_r\, sin\varphi_i \tag{4-14}$$

$$F_x = 0$$

Since the rotational motion of the crank shaft is moving purely in the z, y frame there are no forces produced in the x direction. However, production of a rotational momentum about the x-axis is made, which depends on the reduced inertia J'_{2T}, rotational speed and crank arm ratio λ. The vertical and transversal momentum is calculated based on the momentum arm which is the distance between each piston and vertical and horizontal force respectively.

$$M_x = \omega\lambda\, J'_{2T}(C_1 \sum_i sin\varphi_i - C_3 \sum_i sin3\varphi_i)\, F_y = \sum_i \omega^2 r\, m_r\, sin\varphi_i$$

$$M_z = \sum_i -h_i F_{y_i} \tag{4-15}$$

$$M_y = \sum_i h_i F_{x_i}$$

4.1.7 ENGINE-BLOCK MOTION

To be able to simulate the motion of an engine-block that is effected by dynamic loads it is necessary to construct a rigid body model with 6 DOF. By combining Lagrange equation, from chapter 3.5, with translational and rotational transformation, from chapter 3.6, it is possible to construct a rigid body moving according to a given reference frame.

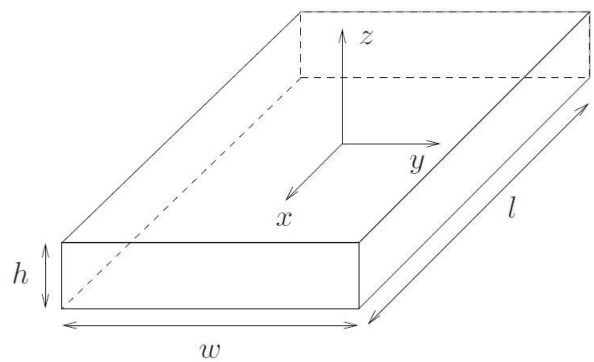


Figure 24 - Homogeneous block

If the genset module is assumed to be a homogenous block with a reference frame in the bottom of the block and the center of gravity is above the reference frame in the vertical direction. The behavior of the block can be expressed by employing the mass matrix M as expressed in chapter 3.6.2, this allows us to describe how the forces and momentum τ influence the velocity $\dot{q}$ of the body.

$$M = \begin{bmatrix} m & 0 & 0 & 0 & mz_g & -my_g \\ 0 & m & 0 & -mz_g & 0 & mx_g \\ 0 & 0 & m & my_g & -mx_g & 0 \\ 0 & -mz_g & my_g & I_x & -I_{xy} & -I_{xz} \\ mz_g & 0 & -mx_g & -I_{xy} & I_y & -I_{yz} \\ -my_g & mx_g & 0 & -I_{xz} & -I_{yz} & I_z \end{bmatrix} \tag{4-16}$$

From IC-field implementation the velocity can be expressed with the following equation.

$$\dot{q} = M^{-1} * int(\tau) \tag{4-17}$$

Further can the change in force because of the motion about the reference frame be expressed through the Coriolis-centrifugal force matrix $c(v)$

$$\dot{p} = c(v) * \dot{q} \tag{4-18}$$

This concludes the modulation of the rigid body for the genset module that is shown as an IC-field in Figure 25. Since the rigid body is modulated as a frame moving according to a reference system we can use Euler transformation to transform the motion in the moving frame into the reference system. Since the velocity and force in the moving frame is known it is possible to use the transformation matrix that was expressed in chapter 3.6. This gives the following expression for transformation of velocity and force.

$$\begin{bmatrix} u \\ v \\ w \\ \omega_x \\ \omega_y \\ \omega_z \end{bmatrix} = R \begin{bmatrix} U \\ V \\ W \\ \omega_X \\ \omega_Y \\ \omega_Z \end{bmatrix}, \quad \begin{bmatrix} F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} = R^T \begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_Y \\ M_Z \end{bmatrix} \tag{4-19}$$

Finally, we have to account for the correlation between rotational velocity of the body $[\omega_x, \omega_y, \omega_z]^T$ and the change of Euler angles $[\dot{\phi}, \dot{\theta}, \dot{\psi}]^T$ as shown below.

$$\begin{bmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{bmatrix} = \begin{bmatrix} 1 & s(\phi)t(\theta) & c(\phi)t(\theta) \\ 0 & c(\phi) & -s(\phi) \\ 0 & s(\phi)/c(\theta) & C(\phi)/s(\theta) \end{bmatrix} \begin{bmatrix} \omega_x \\ \omega_y \\ \omega_z \end{bmatrix} \tag{4-20}$$

The reference frame transformation is modulated in a MTF-field and is shown in the bond graph in Figure 25 as the “EulerTransf”. The reference frame gives an output of the rotation about the frame.

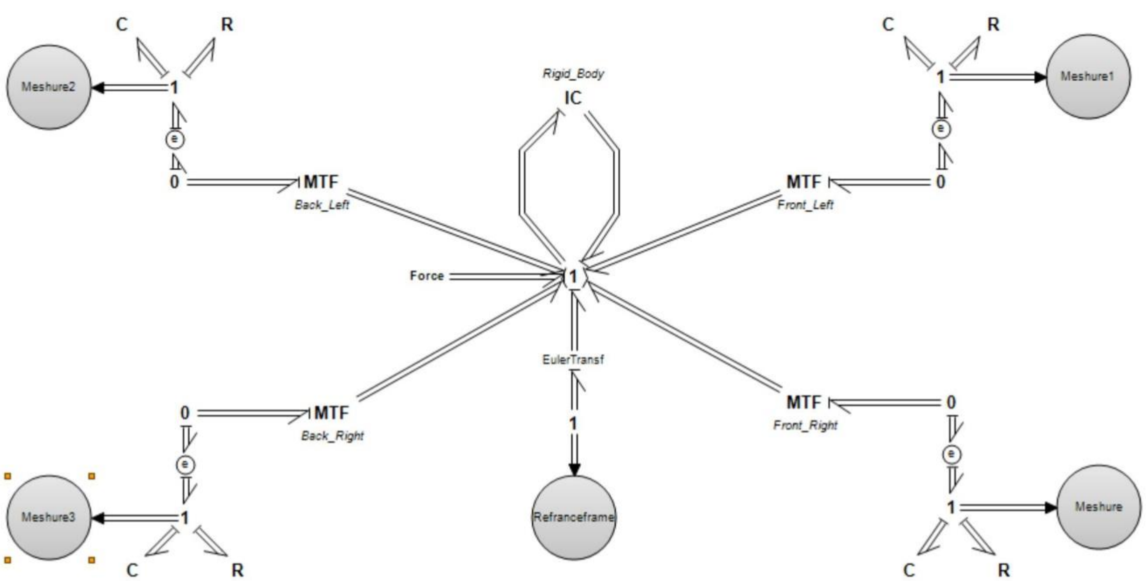


Figure 25 – Bond graph model of engine block with suspensions.

4.1.8 SUSPENSION FORCES

The goal of the model is to identify the suspension forces in the foundation. Since the motion in the suspension can be related to the motion of the rigid body it is possible to use a similar technique as for the transformation of the moving frame to the reference frame. The only difference is that we in this case should account for both rotational and translational transformation. To describe the rotational transformation, it is possible to use the same approach as in the previous chapter. For the translation, it is necessary to describe the position of the new frame according to the reference frame in all directions. This can be done by using a skew-symmetric matrix where the translation is in x, y, z direction.

$$MT = \begin{bmatrix} 1 & 0 & 0 & 0 & -a_z & a_y \\ 0 & 1 & 0 & a_z & 0 & -a_x \\ 0 & 0 & 1 & -a_y & a_x & 0 \end{bmatrix} \tag{4-21}$$

By combining the rotational and translational transformation it is possible to describe the motion in each of the four bearings caused by the motion of the rigid body. This allows for calculation of forces acting in each bearing separately. This is implemented into a MTF-field, and gives the following expressions for effort and flow.

$$\begin{aligned} \dot{q}_1 &= (R * MT)\, \dot{q}_2 \\ \dot{p}_2 &= (R * MT)^T\, \dot{p}_1 \end{aligned} \tag{4-22}$$

The last thing that needs to be included to finish the modulation of the engine frame with suspensions, is the stiffness in each bearing. As explained in chapter 3.3.4 the mounting can be expressed as a mass-spring-damper system. In bond graph this can be modulated simply by using a R- and C-element where the dampening coefficient and spring stiffness are implemented. The complete bond graph model of the rigid body including bearings are shown in Figure 25. A commonly used method for genset foundation is to connect the engine and generator to a common base frame and then fix it to the floor with rigid mounts. This allows for dampening of the force transferred from the frame to the foundation. To avoid that the engine operation speed match the natural frequency of the combined block, DNV GL states that the natural frequency of the mounted unit should be minimum 10 % larger than maximum operation speed. (DNV GL, 2009) Since the mass and frequency of the unit is known it is possible to calculate minimum spring stiffness by the formula for natural frequency.

$$k = ((\omega_n * 2\pi)^2 * m) * \%TOL \tag{4-23}$$

To express the losses from the foundation of the rigid body model a R-element is used to dissipate energy from the system. Since the dissipated energy is from the damper it can be expressed with the following expression.

$$F_d = C\dot{x} \tag{4-24}$$

By developing the expression for dampening coefficient, C by the dampening ratio, ξ it is possible to express the dissipated energy as follow.

$$\xi = \frac{C}{2m\omega_n} \tag{4-25}$$

This gives the final expression that is implemented into the R-element in each engine mounting

$$F_d = (2m\xi\sqrt{\frac{k}{m}})\dot{x} \tag{4-26}$$

Finally, the energy is stored from the spring modulated as a C-element. The amount of stored energy can be expressed by displacement and spring stiffness, where k is spring stiffness and x is the displacement.

4.2 FRAME DESIGN

4.2.1 INITIAL FRAME DESIGN

Since the frame is placed in a machinery room with limited space it is important to minimize the size. On the other hand, it is critical to design a frame that is such that it can withstand the dynamic and static loads from the gensets. To secure a safe construction simulation that visualize the stresses applied from genset operation will be made, the tool used is SolidWorks which allows for dynamic and static analysis of constructions.

As mention in chapter 2 each frame is intended to have between 9-15 canopy slots. With three columns and three to four rows, this can be variated depending on requirement and available space.

For simplicity, the model is constructed with possibilities to change material, construction beams and beam size. When the analysis is carried out it is easy to go back and change dimensions, material or fixed geometry to reinforce the construction. To make the analysis less time consuming and decrease the complexity the model is used for simulation of a frame with only four slots. SolidWorks includes a function with structural steel that can be used during the design process. This toolbox includes several types of beams and the construction can be given several types of material properties.

Figure 26 – Initial base frame design with canopy mounted.

For the initial design the beam used is IPE 80 constructed with material ASTM 36A steel. Data for material and beam can be found in appendix C, together with dimension drawings. Table 8 shows mass-center and weight for the construction with the chosen beam and material.

Table 8 - Properties of the initial frame design

Center of mass x	Center of mass y	Center of mass z	Mass frame
mm	mm	mm	kg
0	0	1518.4	650

4.2.2 REINFORCED FRAME DESIGN

Because the initial frame design did not meet the criteria from DNV GL according to velocity in steel constructions, a new iteration is done to reinforce the construction. The final design is shown in Figure 27. To increase the rigidity of the construction, an extra row of feet is implemented in the center of the construction. This reduces the free length of the beams at the first floor. Further, cross beams implemented in the horizontal plan to increase the stiffness and reduce the deflections.

Another parameter that is of interest is natural frequency of the frame. By increasing the rigidity of the frame, it is possible to manipulate the frequency.

Figure 27 – Reinforced frame design

The cross section of the beams is increased from IPE80 to IPE100. While the material that is used is the same as for the initial sketch. It is desirable to keep the weight of the construction low, but still be within the class requirements. By increasing the cross section and add extra beams the mass is increased by approximately 30 %. Mass center and weight of the finished design are shown in Table 9.

Table 9 – Properties of the second frame design

Center of mass x	Center of mass y	Center of mass z	Mass frame
mm	mm	mm	kg
0	0	1518.4	924

5 SIMULATION RESULTS

The simulation can be divided into two parts, the first would be to simulate the behavior of the engine modeled in 20-sim. The second simulation is done by implementing vibration response in the engine mountings of the frame, which is modeled in SolidWorks. For the first simulation, the desirable outcome is to verify that the engine suspension is within the requirement given from DNV GL. Engine simulation model in 20-sim will give data on the dynamic forces that are produced in each bearing. This data is implemented as dynamic response for analysis of the stacked frame drawn in SolidWorks. Another important result that can be found from the engine model is the frequency response specters, presented through a Fourier transformation (FFT). This can be compared to the natural frequency of the stacked frame to secure that there is no interference between them.

The simulation in SolidWorks is divided into two parts. First, a static analysis is carried out to estimate tension from gravity load. Then a dynamic analysis is made to evaluate combined static and dynamic response. From the dynamic study it is possible to decide natural frequency and velocities in the construction. This can be compared to response spectrum from genset module and class rules. To reduce the complexity of the simulation, only four slots in the frame are accounted for in the simulation.

Results from engine operation are presented as maximum values for displacement, velocity and force, during acceleration and at synchronous speed. This is because it is the peak values that are of interest and it is difficult to read data from the plots when the frequency is that high. Some additional plots can be found in appendix A.

5.1 ENGINE SIMULATION

This part presents results from simulation of the diesel engine and considers how the engine mounting is influenced by the inertia forces produced from the reciprocating motion of the crank mechanism. Since the generator are producing alternating current at a frequency of 60 Hz, the engine is required to operate at a constant speed of 1800 RPM before the generator is synchronized to the grid. In the simulation, the engine is started and accelerates up to generator speed, with low load before the load is gradually increased until it reaches maximum power. The engine should then stabilize at synchronous speed. The simulation is ended after 30 seconds.

Table 10 shows some variables that are used in the simulation, combined with parameters that are stated in chapter 4.1 are all required parameters to execute the simulation given. The 20-sim simulation tool makes it possible to choose between different integration methods to solve the numerical simulation. Both Runge-kutta and Vode Adams are possible to use for this calculation, but Vode Adams is especially suitable for stiff systems with low or high frequent vibration.

Table 10 – Parameters implemented in simulation model

Speed (N)	Power (P)	Spring stiffness (K)	Dampening ratio (ξ)	Dampening coefficient (D)	Mass of module (m)	Canopy size (L x H x W)
RPM	kW	kN/m		kNs/m	kg	m
1800	221	2231.7	0.7	4463.4	2284	2.6 x 1.4 x 1

Figure 28 shows how the engine accelerates during the first 5 seconds with very low load at the shaft. After approximately 10 seconds the load at the shaft has reached the engines maximum rated power and the controller stabilizes the engine speed to synchronous speed.

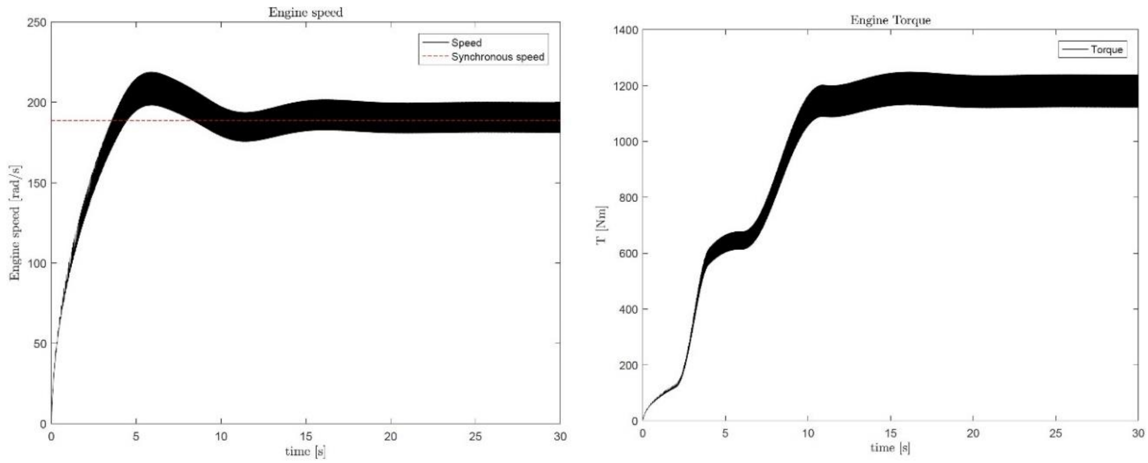


Figure 28 – Engine operation speed and Torque output based on generator load.

Simulation of the diesel engine in 20-sim gives output of inertia forces produced from the reciprocating motion of the crank-shaft. This gives the basis for calculation of velocity, displacement and forces in the engine bearings.

Table 11 shows the inertia forces and momentums produced from the oscillating and rotating mass for the 5-inline 4-stroke diesel engine from Scania. From the simulation, it was found that the inertia forces are so small that they can be neglected. The moment of inertia in rotational direction is balanced out by the firing sequence of the cylinders. On the other hand, there is significant contribution from the inertia momentums in vertical and transversal direction.

Table 11 – Inertia force and moments produced by reciprocating motion from engine simulation.

	Fx	Fy	Fz	Mx	My	Mz
Speed	kN	kN	kN	kNm	kNm	kNm
Max	0	0	0	0	24.32	0.67
Synchronous	0	0	0	0	20.55	0.56

According to regulations there are certain maximum displacement and velocity in the engine mounts because of dynamic loads. Table 12 shows maximum deflection and velocity in all three directions operating at synchronous speed and during engine spin up.

Table 12 – Displacements and velocity in engine mounting units

	x	y	z	$\dot{x}$	$\dot{y}$	$\dot{z}$
Speed	mm	mm	mm	mm/s	mm/s	mm/s
Max	0.0007	0.0017	0.038	0.13	0.32	16.12
Synchronous	0.0005	0.0014	0.034	0.10	0.24	12.79

Since the aim of this simulation is to study how the engine excitation influences the frame construction, the forces transferred through the engine mountings are of special interest. Table 13 shows the forces in three directions. While the horizontal and axial force are quite small and can almost be neglected, the vertical force gives a significant contribution of 4.26 kN at synchronous speed.

Table 13 – Forces produced in engine mounting units

	Fx	Fy	Fz
Speed	kN	kN	kN
Max	0.05	0.13	5.08
Synchronous	0.04	0.10	4.26

To analyze the vibration response in the flexible mountings it is common to measure the frequency response amplitude. This can be done by preforming a FFT of the RMS values of the velocity in the mounting units. This gives the vibration amplitude as a function of the frequency. Frequency response spectrum gives a good indication of which natural frequencies to avoid for constructions that are connected to the engine mounting units at different harmonic orders. To make the frequency response spectrum, the FFT function in Matlab has been used, by analyzing the RMS value of velocity from the 20-sim engine simulation. This is repeated for velocity in all three directions at synchronous speed. The first order harmonic response is given by the operational speed of the engine. This can be expressed by the following formula.

$$Hz = n * \frac{N}{60} \tag{5-1}$$

Where n is the harmonic order, Hz is the frequency and N is engien speed. From the formula it is possible to see that the first order response at operation speed of 1800 RPM is equal to 30 Hz. Figure 29 shows Frequency response spectrum for rms velocity in axial direction. This gives a peak amplitude for first order harmonic at 30 Hz. Because of the low velocity in transversal direction the reponse becomes quite small.

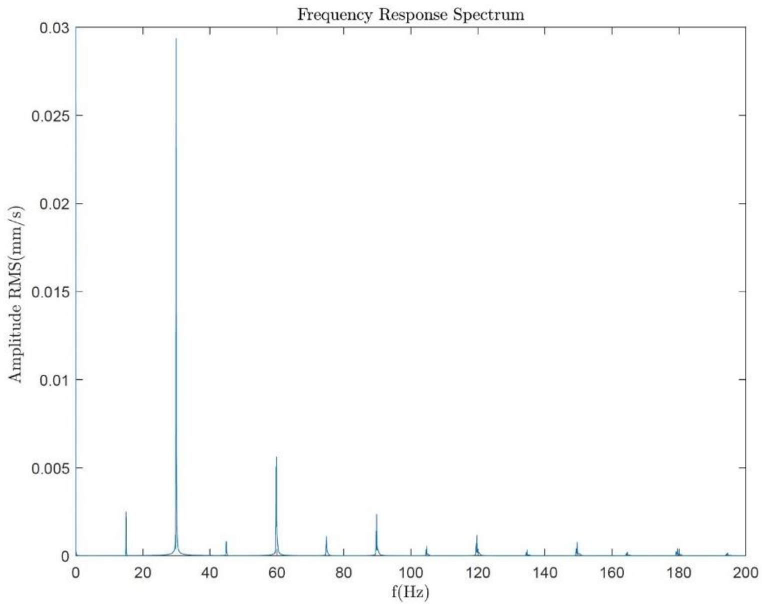


Figure 29 - Frequency response spectrum at synchronous speed. based on RMS velocity in x direction

Figure 30 Shows frequency response spectrum based on velocity in transversal direction. As the figure shows, the distribution of response is equal to the axial direction. However, the amplitude in transversal direction is larger.

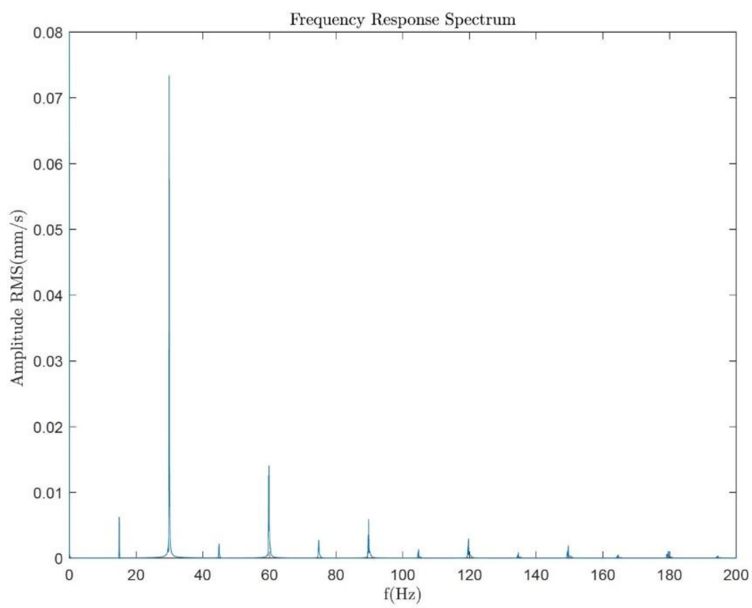


Figure 30 - Frequency response spectrum at synchronous speed. based on RMS velocity in y direction

Frequency response spectrum for the velocity in the mounting unit in vertical direction is presented in Figure 31. The result shows that second order harmonic frequency at 60 Hz gives a very large response while there is a small peak for half order and fourth order harmonic response.

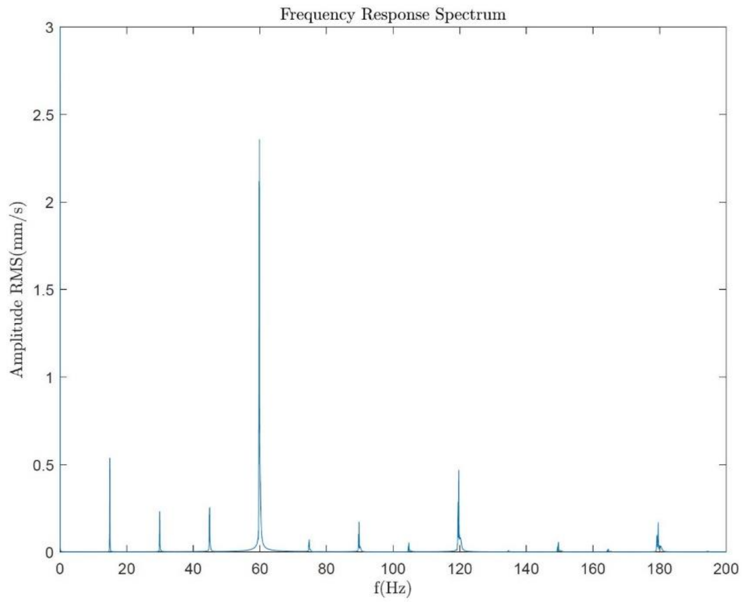


Figure 31 - Frequency response spectrum at synchronous speed. based on RMS velocity in z direction

5.2 PIPE CONNECTION

When connecting pipes to the vibrating canopy it is important to use flexible connections to avoid large stress and fatigue in the connection. When choosing mounting method and stiffness of the flexible pipe connection, the motion in the connection point is of interest. The 20-sim model of the rigid body includes a measuring of velocity and displacement in the connection point of the exhaust pipe. The exhaust pipe is located at the rear side of the canopy (-1.25 x 0.40 x -0.30) m according to mass-center of the canopy.

Table 14 shows measured maximum values in velocity and displacement at the connection point at synchronous speed. Like the result in the mounting units, the displacement and velocity are largest in vertical direction. However, the deflection and velocity in axial direction are larger for the pipe connection.

Table 14 – Displacements and velocity in pipe connections at synchronous speed

	x	y	z	$\dot{x}$	$\dot{y}$	$\dot{z}$
Speed	mm	mm	mm	mm/s	mm/s	mm/s
Synchronous	0.011	0.0014	0.033	3.97	0.25	12.32

5.3 DYNAMIC AND STATIC SIMULATION OF FRAME DESIGN

An initial design of the genset stacking frame was introduced in chapter 4.2. This will be the basis for the first simulation in SolidWorks. The simulation is divided into two parts, the first part looks at the static analysis of the frame and identify stress and displacement from the mass-force produced by each canopy. Furthermore, the frequency response and influence of dynamic load are analyzed. Since the dynamic load in axial and horizontal direction is very small, these are neglected in the dynamic simulation. Another important dynamic load that is excluded in the simulation are forces produced by sea motions. Because of this there should be a considerable safety margin in displacement and stress limits.

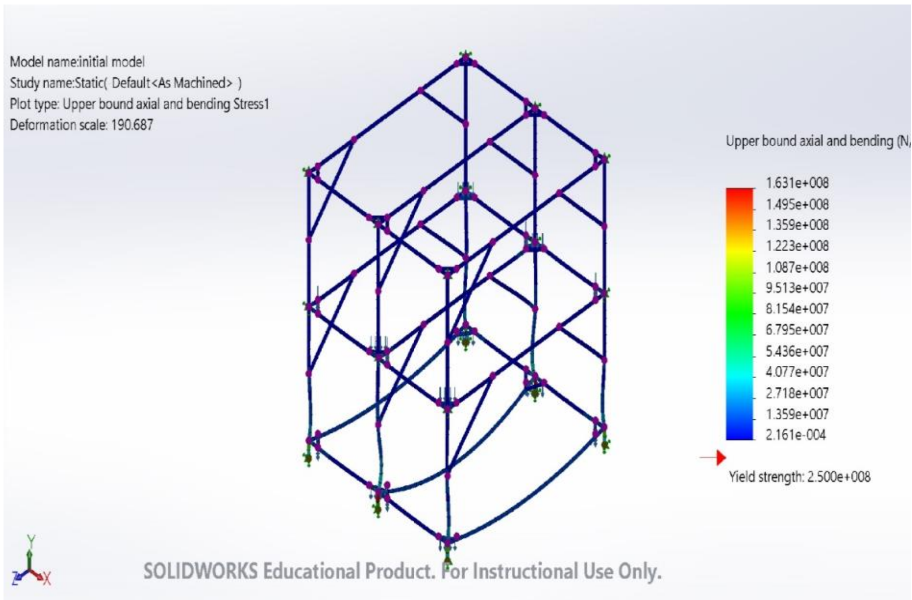
5.3.1 STATIC ANALYSIS

There are three factors that should be defined before the static analysis can be made. Since the mass-center of the canopy is placed in the middle of the frame, the force is equally distributed at each bearing. The excitation force is dependent on the mass of the canopy that is 2284 kg. This gives a force of approximately 5601 N in each bearing. To decide the fixed geometry of the frame several, simulations was performed to check how this influenced tension and frequency of the construction. By having a rigid fixed frame, the natural frequency and stress increased while the displacement was decreasing. Since it is desirable to increase the natural frequency of the frame to avoid the frequency response specter of the engine, a rigid fixed frame was chosen as the best solution.

The static simulation results of the initial frame design are presented in Table 15. The results present maximum displacement and stress in the frame.

Table 15 – Static analysis of initial frame design

Name	Type	Min	Max
Stress	Shear in Y Dir. on YZ Plane	0 MPa	163.1 MPa Element: 641
Displacement	Resultant Displacement	0 mm	1.168 mm Node: 306

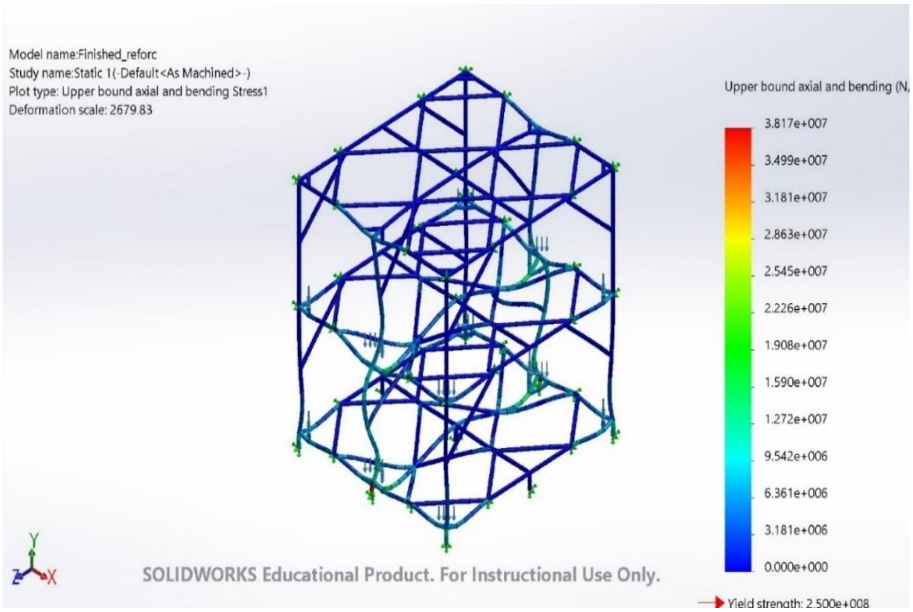


Initial frame design static stress analysis

The same simulation is made for the reinforced frame design. The simulation results from the improved frame are presented in Table 16. It shows that both stress and displacement are reduced significantly compared with the initial frame design. This is done by inserting extra supports in the center to reduce length between the fixed geometry and add brace beams to increase the stiffness of the frame. The maximum stress is 38.16 MPa and is located in the support between the two gensets, this point is placed close to two engine mounting units and support a heavier load. The largest measured deflection is 0.119 mm. This shows that the design measures have improved the model drastically.

Table 16 - Static analysis of base frame improved design

Name	Type	Min	Max
Stress	Shear in Y Dir. on YZ Plane	0 MPa	38.16 MPa Element: 550
Displacement	Resultant Displacement	0 mm	0.119 mm Node: 953



Improved frame design static stress and displacement analysis

5.3.2 COMBINED STATIC AND DYNAMIC ANALYSIS

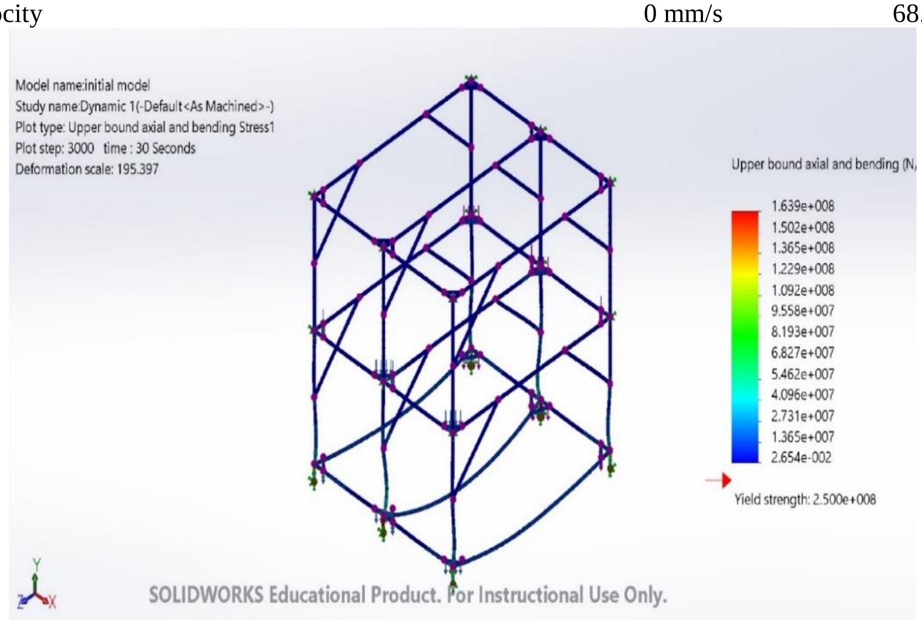
The dynamic analysis considers both the static loads from gravity and dynamic load from genset vibration. SolidWorks has a function that allows to implement the static result and run a combined analysis that studies influence from all forces simultaneously. As mention the axial and horizontal bearing forces are very small and are neglected in this simulation. SolidWorks includes a function that makes it easy to implement harmonic loads variating with either a sinusoidal or cosine function. Since the dynamic load at constant speed can be expressed by a sinusoidal function it is easy to implement the dynamic load into the simulation by the following equation.

$$x(t) = A * \sin(\theta + \Delta\theta) \tag{5-2}$$

Where $x(t)$ is the force with time, A is the force amplitude and $\Delta\theta$ the phase offset. The dynamic load is equal at the front and back support of the canopy but with 180° phase difference. Amplitude for the load is imported directly from the dynamic loads shown in Table 13. By implementing the dynamic load and including the static load from the first simulation it is possible to analyze the steel structure. The result from the initial design simulation is shown in Table 17. The table shows that the maximum stresses in the frame construction does not changing much when influenced by the dynamic loads. The displacement has, on the other hand, increase with approximately 30 % to a maximum of 1.677 mm. In the dynamic analysis it is possible to measure maximum velocity, for the initial design this become 68.60 mm/s.

Table 17 – Dynamic stress analysis of initial frame design.

Name	Type	Min	Max
Stress	Shear in Y Dir. on YZ Plane	0 MPa	163.9 MPa Element: 594
Displacement	Resultant Displacement	0 mm	1.677 mm Node: 42
Velocity		0 mm/s	68.60 mm/s

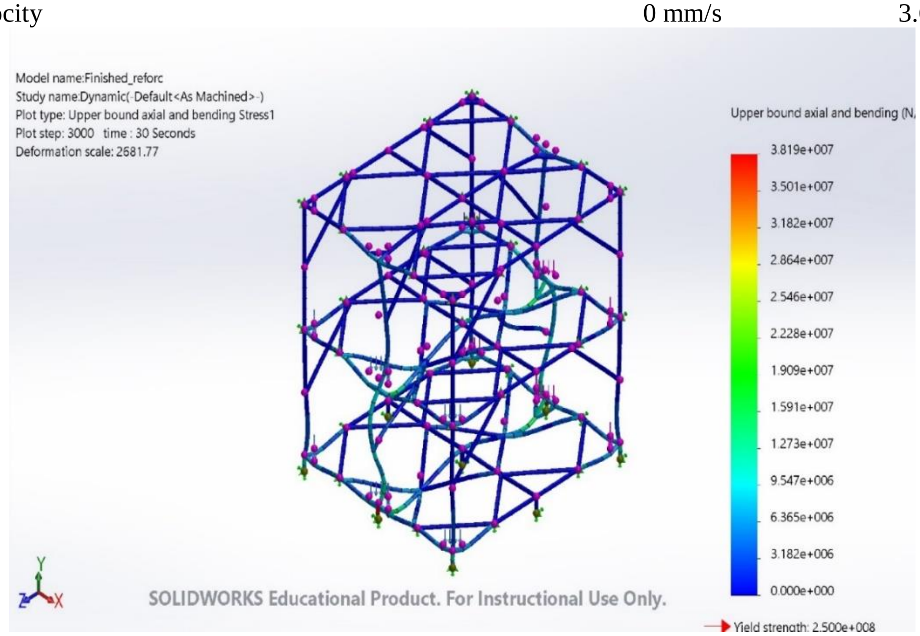


Initial frame design dynamic stress analysis

For the reinforced frame design is the same dynamic analysis carried out. The result is presented in Table 18, and show the same trend as for the static analysis. There are a drastically reduction in shear stresses and both displacement and velocity is decreased significantly compared to the initial design.

Table 18 – Combined dynamic and static analysis reinforced frame

Name	Type	Min	Max
Stress	Shear in Y Dir. on YZ Plane	0 MPa	38.19 MPa Element: 550
Displacement	Resultant Displacement	0 mm	0.119 mm Node: 550
Velocity		0 mm/s	3.63 mm/s



Reinforced frame design dynamic stress and displacement analysis

5.3.3 FREQUENCY RESPONSE

When steel structures are excited by dynamic loads it is important to secure that the natural frequency of the structure is outside the operational area of the periodic load. The natural frequency is dependent of stiffness and mass of the structure. SolidWorks can calculate different harmonic orders of natural frequency for the structure. From the dynamic analysis of each of the frames, the natural frequency of the first four orders is calculated. The increased stiffness and rigid mounting of the beam structure give a significant increase of natural frequency as shown in Table 19.

Table 19 – Natural frequency of initial frame construction.

Order	1st	2nd	3rd	4^{th}
	Hz	Hz	Hz	Hz
Initial frame	17.62	20.94	23.16	23.17
Reinforced frame	139.09	140.90	141.95	142.79

6 DISCUSSION

6.1 GENSET

6.1.1 GENSET MOUNTING SYSTEM

When installing a genset in the machinery room it is important to optimize mounting to reduce vibration transfer to the surroundings. It is normal to distinguished between rigid and resilient supports, in this case the engine is mounted with resilient rubber mounts. DNV GL rules for vibration in machinery equipment state maximum values regarding frequency and velocity in the mountings. Acording to these rules, should vibrations with a frequency response in the area 4 – 200 Hz be analyzed. Another important measure is the velocity in the mounting unit. For a diesel engine operating above 200 RPM, the maximum requirement for a resiliently mounted unit is 24 mm/s while for a diesel driven generator set the maximum velocity is 18 mm/s. For simplicity only the oscillating motion in the diesel engine is modeled, but since the installation consist of a genset module, the rules for diesel driven generators can be used.

From the simulation of the mounting system, a maximum velocity of 12.79 mm/s was found in the bearing, when operating at a constant speed of 1800 RPM. This is within the rules that is given from DNV GL and indicate that rubber mounting used as foundation is acceptable. However, the performance of this configuration is not optimized. As mention in chapter 4.1.8 the stiffness of the spring is decided by the rule that requires natural frequency of minimum 10 % above operational speed. By changing the spring stiffness and dampening factor it is possible to manipulate the force and velocity produced in each bearing. An optimization analysis of the rubber mounting should be performed before the unit is installed to reduce the dynamic loads transferred to the stacked frame.

6.1.2 INERTIA FORCES AND MOMENTS

When simulating the inertia forces and moments produced from the reciprocating motion of the engine there is only response in vertical and transversal moments. By comparing this to the result in the pre-project that is presented in chapter 3.3.2 with the result in rigid body simulation, it can be concluded that this behavior is as expected. This also correspond with the expectations stated in Wärtsilä project guide which states that the inertia forces of reciprocating motion is zero for an 5 cylinder engine. (Wartsila, 2005). Because mass and inertia of crank-shaft, connection rod and piston is unknown, the values needed to be assumed, which creates a certain margin of error in the result that should be considered.

6.2 FRAME

In a design process the designer is required to perform multiple iterations, to secure that the construction satisfy the requirements. During this process, there was first made a initial frame design that was analyzed, when the simulation result for the first design did not satisfy the criteria, a new model where made to eliminate the problems. The initial model had quit large shear stresses, displacement and natural frequencies where low. The dynamic simulation only considers vibrating load from reciprocating machinery. Which means that influence from seaway motion, shock from moving of canopies and frame installation is excluded. It is important to include a margin of safety from material yield strength to account for the shortcomings in the simulation. Axial stress in the intial design was 163 MPa, which is quite high. Because of this it was desirable to make a second design that decreased the stresses from gravity and dynamic loads.

In the second design a new row of frame feet was included between the existing feet. This is done to reduce tha space between the fixed supports. Another improvement that was performed on the second design was to include cross beams to increase frame stiffness. Finally, the rigidity was increased by fix the construction in multiple points.

6.2.1 FRAME STRESS

When considering the stress in the frame construction, the simulation analyzes the maximum stress that is present in the construction. This can be either bending or shear stress. This result is compared to the maximum yield strength for the material that is used in the construction beams. For the frame design is the structural steel ASTM 36A used, this steel is commonly used in beams and for steel construction. The maximum yield strength for this material is 250 MPa, before it suffers permanent deformations. As mention earlier, the simulation only consider static and dynamic load from reciprocating machinery, thus maximum stress should not be too close to the yield strength.

For the initial frame design the maximum stress in the construction is 163 MPa, when comparing the result from the static analysis and the combined static and dynamic analysis, the difference between the stress is very small. It was expected that the dynamic load should influence the frame tension more. However, the implemented dynamic load act as a moment about center of the canopy that is shifting with very high frequency. This moment influences the bending stress, but since the shear stress is most dominant in the construction, the maximum stress is not increasing appreciably. The same trend is observed for the reinforced frame design. In static analysis, the maximum stress was measured to 38 MPa, which is a significant reduction from the initial design. As for the initial design, it is observed that the dynamic load gives a small contribution to maximum stress in the construction.

6.2.2 STRUCTURE VELOCITY DO TO DYNAMIC LOADS

When simulating the dynamic and static load acting at the steel frame construction, the velocity in the frame is measured. Since large velocity in steel structures can lead to fatigue damage and cracks in the structure it is important to limit the velocity. As mention in chapter 3.2, DNV GL states limits regarding maximum structural vibration velocity. For steel constructions, it is specified that structural vibration velocity is not to exceed 45 mm/s.

The simulation result from the dynamic analysis is shown in chapter 5.3.2, the initial design maximum velocity was 68.30 mm/s while the reinforced frame designs maximum velocity was 3.63 mm/s. This show a significant improvement of frame motion with the redesigned frame. With the initial frame design the vibration velocity would exceed the requirement from DNV GL, this give substantial risk for fatigue damages and the design would not be approved. However, with the reinforced frame design is the vibration velocity decreased significantly and perform beneath the acceptance criteria with a large margin.

6.2.3 FREQUENCY COMPARED TO APLIED PERIODIC LOAD

An important design parameter is the natural frequency of the frame. If the natural frequency is in the same area as the excitation frequency, from the genset, there can be problems with large resonance in the frame. This can give fatigue damages in the frame because of large deflections or stress in the construction. The initial frame had a natural frequency between 17 – 23 Hz for the first four harmonic orders. When comparing natural frequency for the frame to the response spectrum for the genset can we see that the response comes in the same area. Peak amplitude for the genset is for 2 order harmonics at 60 Hz and a significant amplitude for both half order and first order at 15 Hz and 30 Hz respectively. For response in axial and transversal direction is there a maximum amplitude at 30 Hz. Since most of the excitation energy from the genset is placed in the area between 15 – 60 Hz it is desirable to move the natural frequency of the frame out of this area.

When redesigning the frame, the focus was to increase the stiffness of the construction such that the natural frequency would increase. By employing extra feet in the center that decrease free length of beams, use cross beams to stiffen up the construction in transversal and axial direction and increase amount of fixed geometry, the stiffness of the construction was improved. By implementing these measures, the natural frequency was increased to 139 – 143 Hz as shown in Table 19. It was desirable to increase the natural frequency of the frame to above 200 Hz to get it out of the genset response area. Different measures were implemented into the design to increase the natural frequency using extra beams in vertical direction, increase beam dimension and cross beams at the frame side. But all these actions made the natural frequency of the frame construction decrease. When studying the frequency response spectrum for vertical velocity in Figure 31, the excitation energy after the fourth order harmonic at 120 Hz is very low. Since the design give natural frequency above every large excitation frequency and dose not correspond to fifth, fourth and one-half order harmonics, the possibility is low for any resonance because of low frequency interference.

6.2.4 COMPARISON

When comparing all factors between the initial and the reinforced frame design we can see a significant improvement between the two designs. Table 20 show how the two-designs performed compared to each other and some limitations. The initial design is close to excitation frequency and yield strength which can lead to large oscillations or permanent deformation, the maximum velocity is also above the criteria from DNV GL on vibration in steel construction. For the reinforced design the result is improved significantly. Frame stress is kept beneath yield strength with good margin and the frequency is removed from the area with the largest excitation energy. It is also within the criteria for vibration velocity. This shows that with relatively small design changes it is possible to improve the performance of the structure significantly.

Table 20 – comparison between initial frame and second frame design.

	Displacement	Velocity	Stress	Frequency
	mm	mm/s	MPa	Hz
Initial frame	1.677	68.60	163.9	17.62 - 23.17
Reinforced frame	0.119	3.63	39.19	139.09 - 142.79
Limits		45	250	

The first design would not be possible to use because it exceeds the rules from DNV GL for vibration in steel structures close to machinery. The fact that the maximum stress in the construction is close to the limits for yield strength is not good when considering the fact motion of seaway and moving of canopy is not included in the calculation. On the other hand, the reinforced design gives good results and should be possible to use as a frame construction.

6.2.5 SPACE REQUEMENTS

Since there is limited space in a machinery room it is an interesting parameter to compare the space requirement for a massive genest module to a conventional machinery system. A possible configuration of the frame can be that each rack has three levels with five slots at each level. If three identical racks are installed in a machinery room, it will give 45 available slots for the gensets. This give a system that can produce 9.95 MW. If each slot is of a size of 2.6-meter-long, 1.1-meter-wide and 1.5-meter-high. Such a frame would require a total volume of 193m^3 with a foot print of 42.9 m^2. In comparison, two Wartsila W9L32 that give a total power output of approximately 9 MW, would require a total volume of 235 m^3 and a foot print of 60m^2(Wartsila, 2005). This show that with a compact design of the massive gensent system, it can be possible to realize without similar space requirements than a conventional machinery system. The fact that it is intended to remove the canopy to a maintenance slot or to a onshore workshop would make it possible to decrease the space between the genset because the requirement for service space between the machines would not be required.

6.3 PIPE CONNECTION METHOD

One of the critical design parameters for the frame is how to connect the auxiliary pipes. From the genset-module simulation the velocities and displacement in the connection point are measured, this would be evaluated according to class rules. This chapter intend to discuss some design concept for auxiliary connection and present some sketches on feasible solutions. It is assumed as reasonable to separate auxiliary supply and exhaust system in two separated arrangements.

6.3.1 AUXILLERY CONNECTIONS

It is important to develop an application that allow simple plug in and plug out of each canopy when it is removed for service. The intention is to construct a panel that include all the connections for auxiliary systems for fast assembly. For fast connection the best solution would be to install quick couplings. However, it could be challenging to find couplings that fits for all dimensions and can tolerate the high pressure and temperatures. Since the aim is to increase efficiency it is an essential detail to install the auxiliary system as a common supply unit. This can reduce the pumping work for operation compared to use of separate auxiliary supply for each genset. Figure

32 show a proposed design of the auxiliary supply system. It is connected to the rear side of the frame and is connected to each canopy through a connection panel. It can be possible to design the panel such that it provides a quick coupling method and secure a flexible connection between the canopy and external pipes.

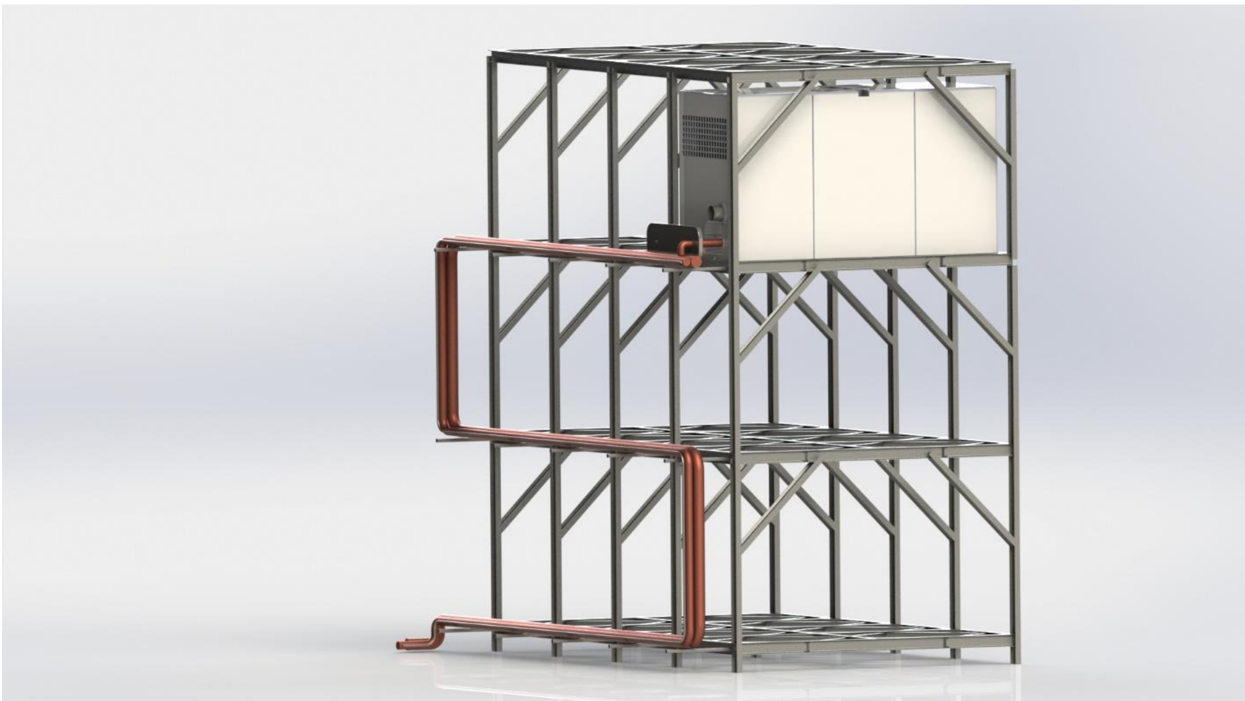


Figure 32 - Auxiliary pipe connection

Since the dynamic load from the genset produce stresses in the connection point it is important to mount the pipe with flexible couplings. This reduce the stress transmission and protect against fatigue. Since this analysis does not include stress in the pipes the standard for maximum vibration in steel constructions connected to machinery is used instead. The velocity is measured at the rear side of the genset module in the area where the pipes is connected. Maximum velocity in this area is 12.32 mm/s this is beneath the requirements from DNV GL at 45 mm/s. However, a stress analysis should be executed before realizing the design.

6.3.2 EXHAUST PIPE

For installation of the exhaust system, the intention was to have a common exhaust pipe for all the engines. This would reduce the amount of pipe required compared with the traditional method where each engine has an exhaust pipe directly to the atmosphere. However, there are some concerns with such a system. According to machinery installation standards (DNV GL, 2013b) it is required to prevent backflow into the engine when multiple machines is connected to a common exhaust pipe. A method used to solve this can be by implementing a non-return valve on the exhaust pipe for each engine. This can prevent the exhaust to flow back into the engines that is not running. One problem that can be of concern regarding this method is that a non-return valve can increase the back pressure in the piping. This can lead to reduction of the engine efficiency which should be prevented. Another problem that can be of concern when connecting the exhaust from multiple engines together, is that the exhaust flow from the engine is pulsating and has different frequency for each engine. This can lead to excessive vibrations in the piping or undesirable wear to the machinery. There is one method that can be used to circumvent this problem, that is to install a surge tank that can reduce the pressure and smoothen the oscillations in the flow. A surge tank would work as a damper that would smooth out the pulsating flow into a linear flow. (Engja) By implementing these two methods it should be possible to design a common exhaust pipe system.

During the design process, two different ideas has been reviewed. The first was to install a large stack at the engine room floor and branch out from the canopies at each floor into a horizontal pulse damper, then these pipes would be connected into the common exhaust stack. This would give one connection for each frame floor. One large advantage with this configuration is that the stack is mounted at the floor in the machinery room, this could reduce the stress in the pipe and reduce amount of fixed point for the construction. However, this design can require large space and it can be difficult to place the stack in a efficient location. The other design that has been considered is shown in Figure 33. This layout can be more space saving and give more room behind the frame construction. As shown in the figure, each pipe branches out from a separate canopy and is connected to a pulse dampener over the construction. One concern with this design is the it would require to be supported in many locations to avoid large stresses.

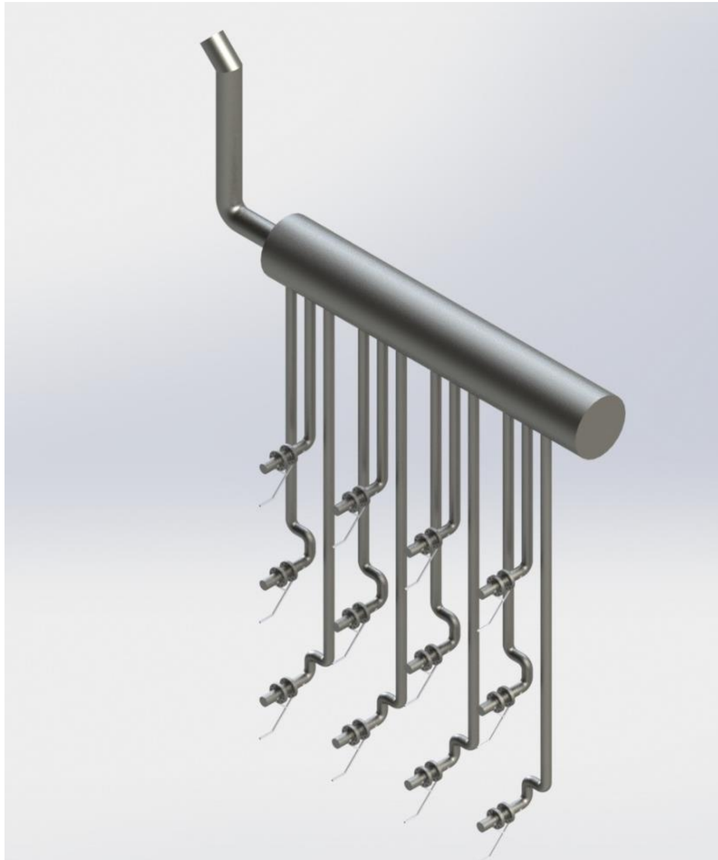


Figure 33 – Proposed configuration for exhaust pipe connection

6.4 MACHINERY ROOM LAYOUT

When assembling the system to a common module each frame can be considered as a self-supplied unit. This configuration makes it easy to implement multiple frames into a machinery room according to customer demand. Figure 34 show a machinery room layout where two frames are installed. This arrangement can provide redundancy and a flexible power production. By implementing a crain in the front of the frame it should be possible to extract each canopy from the frame for removal or maintenance. It is intended to construct a maintenance spot inside the machinery room. Then the genset can simply be disconnected from the grid and dismounted for maintenance work.

Figure 34 – Machinery room layout

7 CONCLUSION AND FURTHER WORK

7.1 CONCLUSION

This thesis has focused on developing a mechanical design for a massive genset system through dynamic simulation. The work was divided into several parts where the initial focus was to develop a basic knowledge on technology for diesel-electric propulsion systems today. Furthermore, the concept of the massive genset system is presented, early in the process a decision was made to design the construction in three parts. First a stacking frame for each genset module, then a canopy that contained the genset and a connection system with exhaust and auxiliary pipes. To evaluate how the system cope with vibration a dynamic analysis was established. The method used to execute the simulation of the diesel engine mountings is based on Lagrange mechanism and bond graph modeling. Data retrieved from this simulation is inserted into a 3D-modeling program with numerical simulation package to analyze how dynamic and static forces effects the frame structure. Finally, a connection method for auxiliary and exhaust system is discussed.

The motivation behind this work was to present a concept that could increase efficiency in marine vessels operation. From the preliminary study a simulation was made to compare fuel consumption for conventional diesel-electric propulsion as opposed to the massive genset system. It shows that vessels operation with large load variation can reduce the consumption, especially at low load operation. The idea of removable canopies gives the configuration flexibility and allow for onboard maintenance. Another concern was related to available space in a machinery room. Since multiple genesets could be space consuming compared to a conventional system. However, comparisons show that area required, for the genset modules, is similar because of the compact design.

The aim of this thesis was to establish a frame design that could be implemented as a base structure for a massive genset system. A crucial design criteria was to secure structural integrity and protect the frame from fatigue because of the vibration excitation of genset operation. According to DNV GL standards of vibration in machinery units, the two central design criteria that should be analyzed, is structural velocity and frequency. For the initial design the natural frequency where low and velocity in the construction was exceeding the limitations. Structure stress where also close to the yield strength of the construction. Therefore, a second design was made, where the aim was to increase the natural frequency and reduce stress and velocity in the construction. By increasing the stiffness, rigidity and increase beam cross-section these criterias were met. The design in this thesis is does not include dynamic loads from seaway and the optimization process is limited. This means that there is room for expanding the analysis and there can be performed multiple iterations to improve the construction.

One of the presupposed benefits of the massive genset system is that is can be possible to extract canopies for maintenance or replacement. To achieve this is it necessary to produce a system that allow fast quick plug in plug out of the modules. Because of limited time, a final solution for this problem is not proposed. However, an initial design idea is made, with an accompanying discussion of benefits and desirable outcomes. When assembling the frame, auxiliary supply and exhaust pipe it is possible to see a compact solution as an initial design. Because of assumptions and limitations in the work, is this design not complete but give a good basis for further work and development of the idea.

7.2 FURTHER WORK

The result show that the massive genset-module is possible to realize and can give some advantages compared to conventional systems. This thesis has introduced the concept and discussed many practical solutions. However, there are still much work to be done before such a system could be realized. How to implement control and synchronization of generators is key factors to optimize fuel consumption that is not analyzed in this work. The connection method for auxiliary and exhaust pipe is presented and discussed as an idea, but extensive analysis should be executed before it is possible to conclude with a feasible solution. Furthermore, the gain of installing a common supply system is an interesting parameter to investigate closer. One of the intentional outcome was to design a shelf construction that made it possible to remove canopies from the frame. How it is possible to extract each module in a safe manner should be investigated closer.

Although, it would require much effort before the system could be realized, this thesis established a good basis for further work on the concept and this topic can be expanded with interesting challenges related to control, structural design and machinery installation.

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APPENDIX

A. PLOT FROM DIESEL ENGINE SIMULATION IN 20-SIM

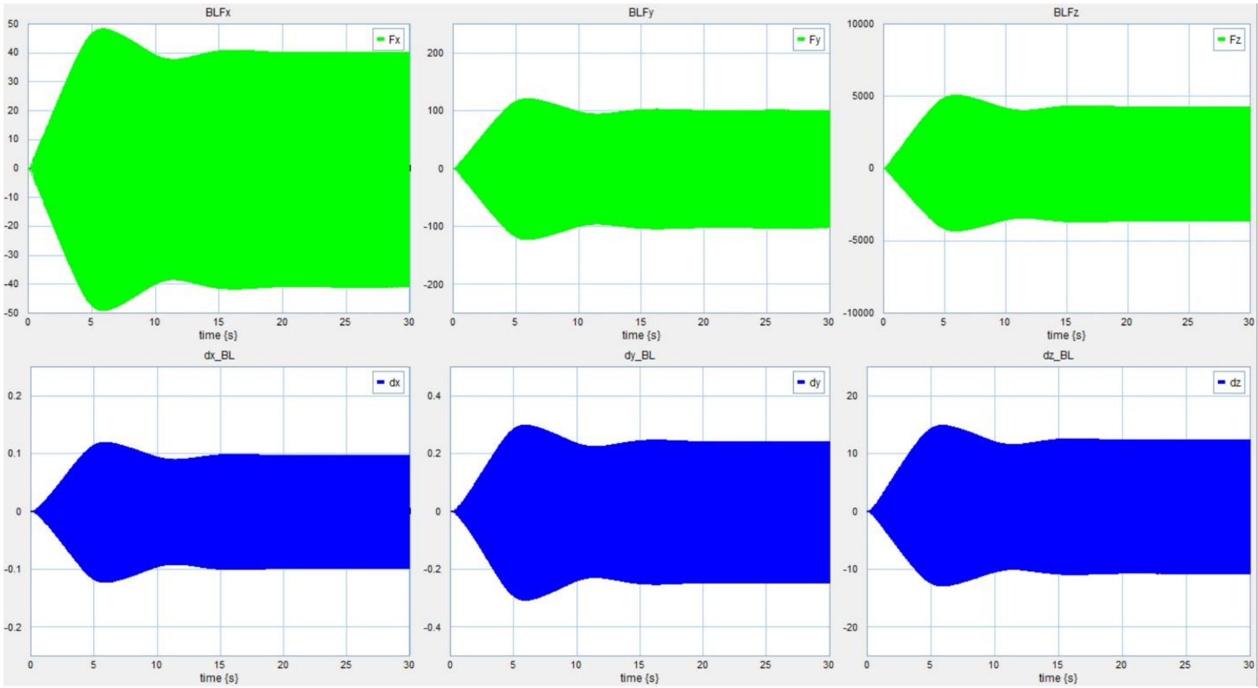


Figure A.1 – Force and velocity in genset mounting

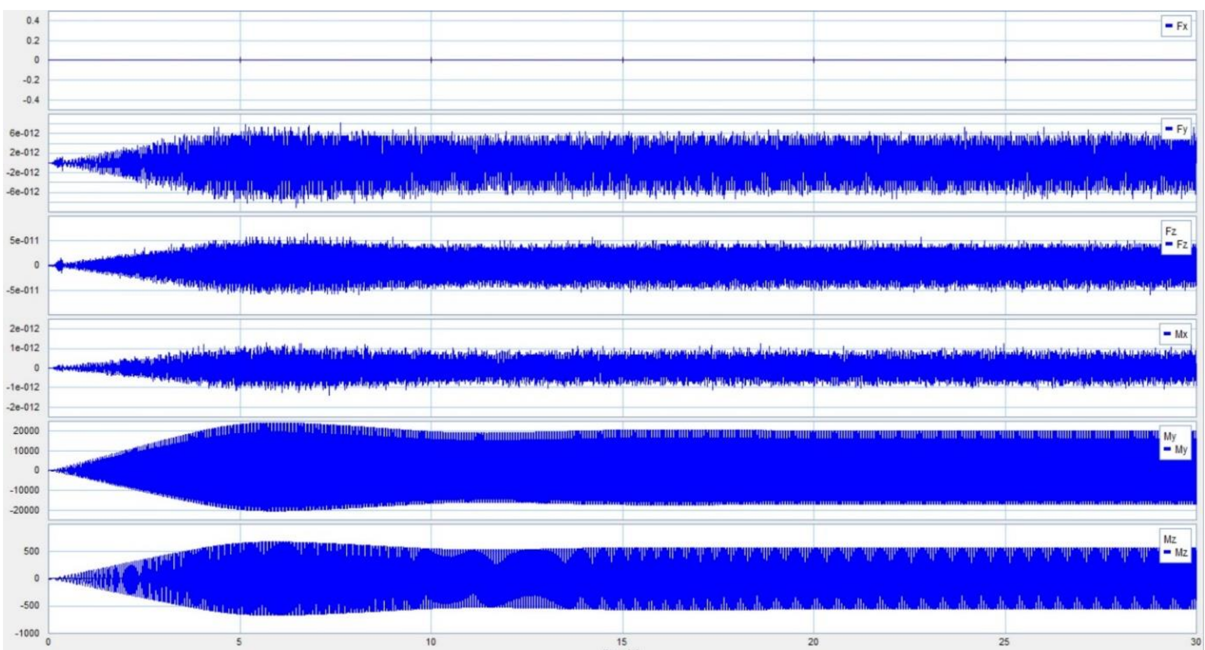


Figure A.2 – Inertia force and moment produced by reciprocating motion in diesel engine.

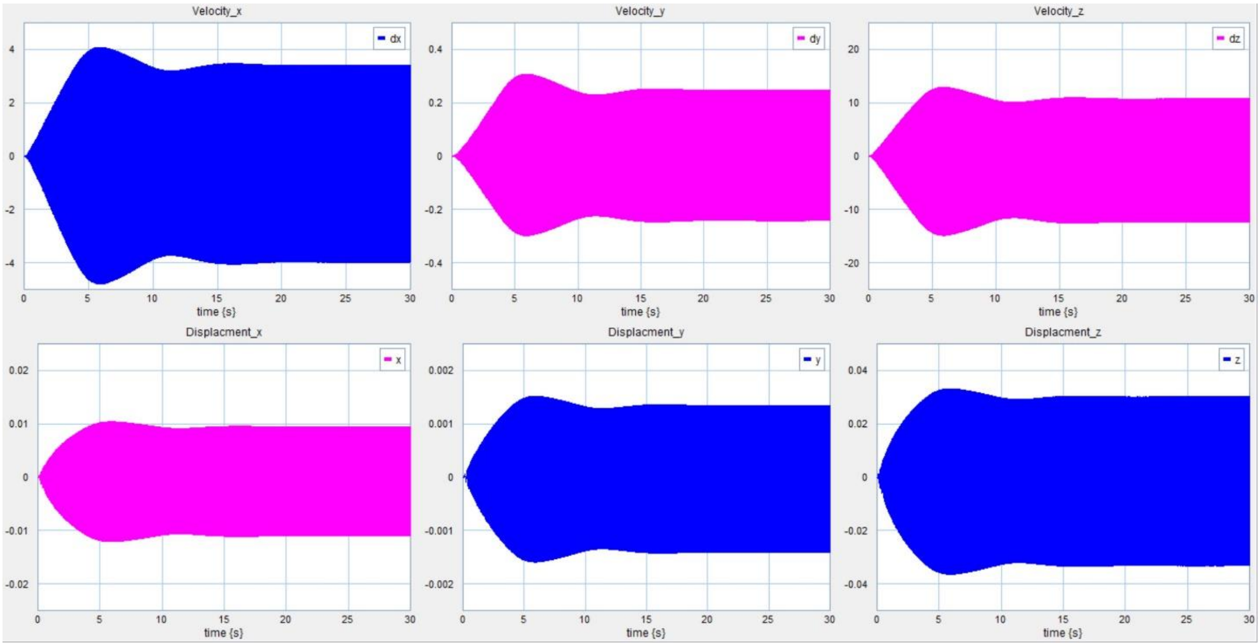


Figure A.3 – Velocity and displacement in pipe connection.

B. DIMENSION DRAWING OF CANOPY AND FRAME

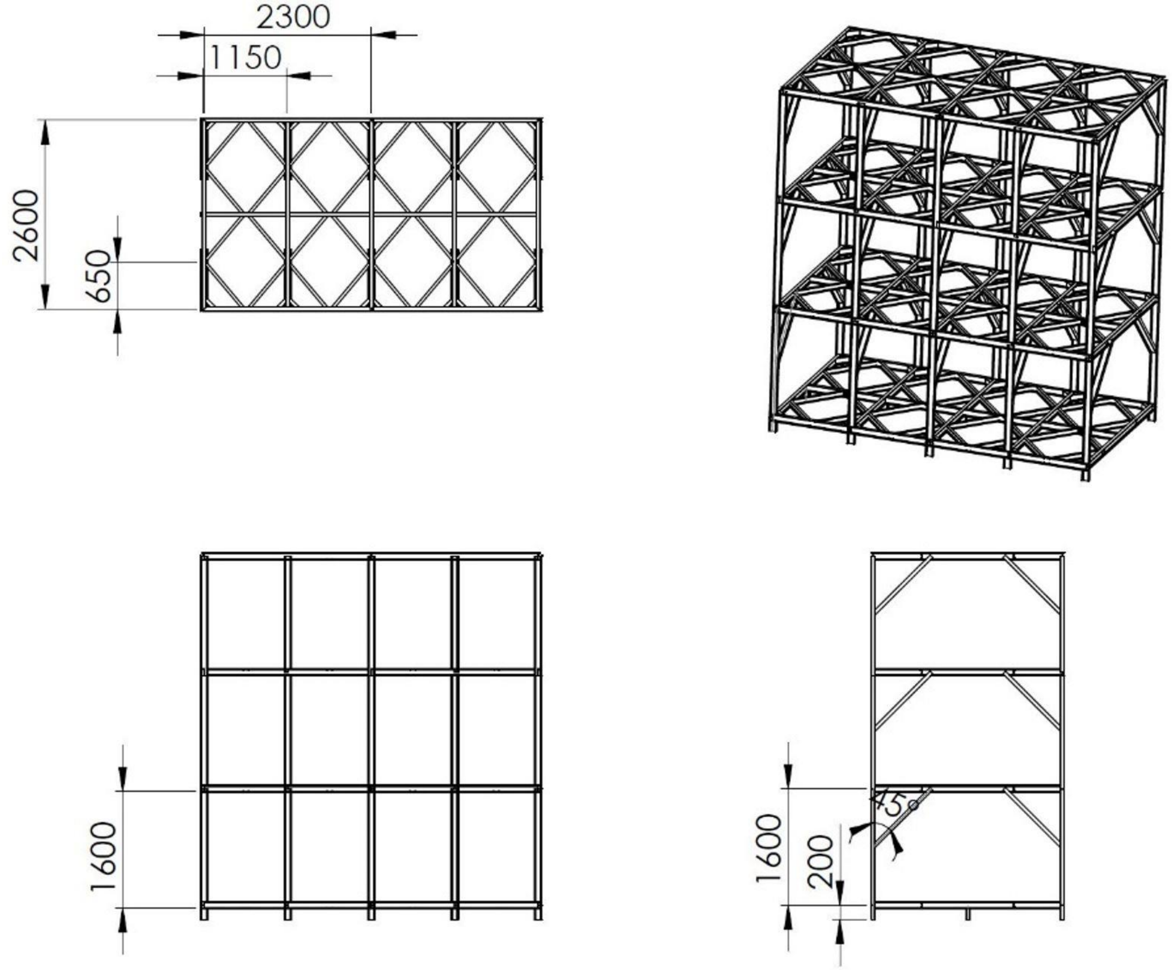


Figure B.1 – Dimension drawing of final frame design

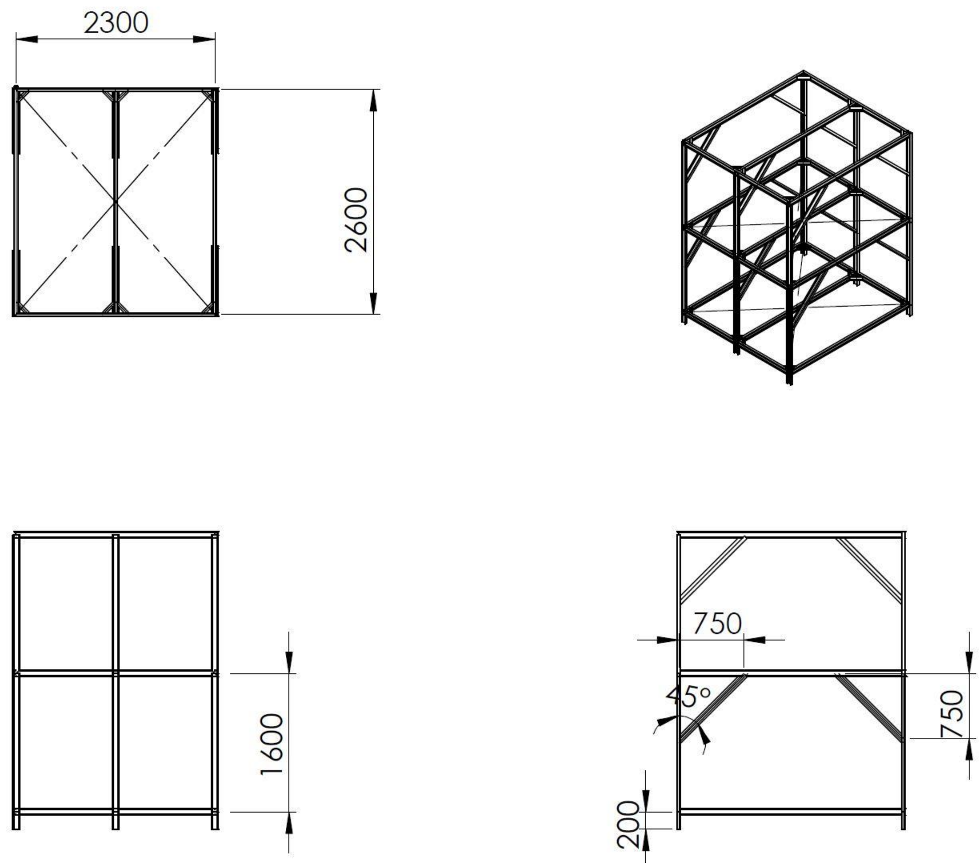


Figure B.2 – Dimension drawing of initial frame design

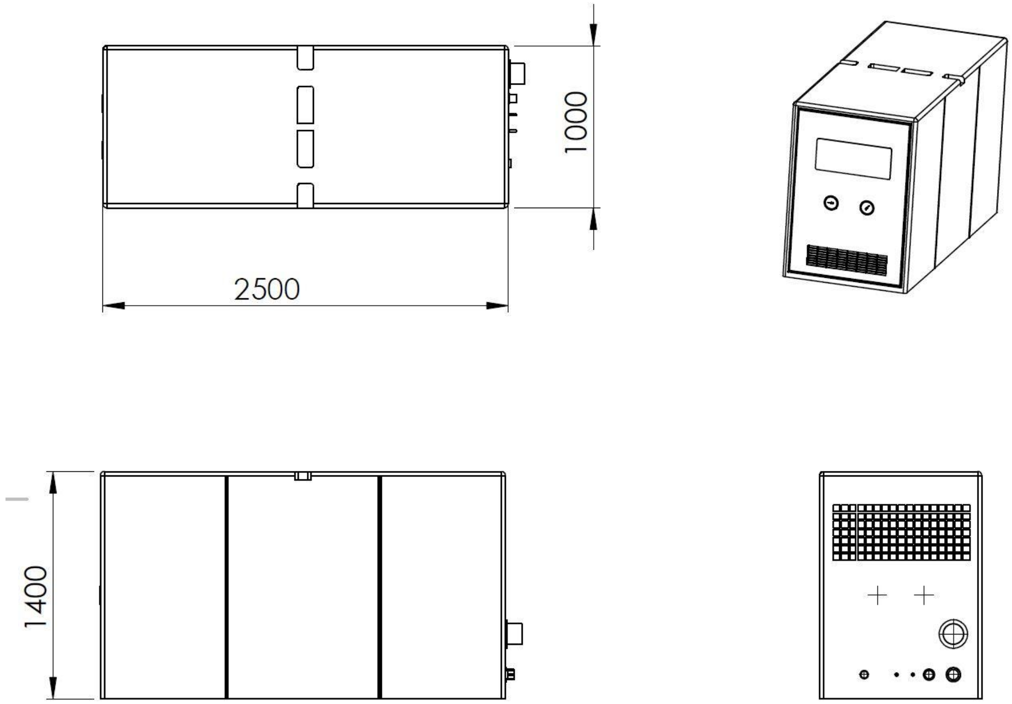


Figure B.3 – Dimension drawing of canopy.

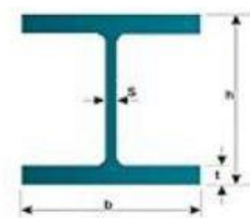
C. PROPERTIES OF IPE BEAM AND MATERIAL

SPAHR METRIC, INC.

YOUR DIRECT SOURCE FOR METRIC STEEL & FASTENERS

Metric I Beam IPE

Material Available: CARBON STEEL - S235JR / S355JO / S355J2 - EN 10025
Stainless Steel - 304, 304L, 316
Available in 3 meter and 6 meter lengths
All sizes shown may not be in stock, please call us for more information



Description	Profile Size	Dimension h	Dimension b	Dimension s	Thickness t	Weight	Finish Options						Length Options	
Metric I Beam IPE	80 mm	80 mm	46 mm	3.8 mm	5.2 mm	6 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	100 mm	100 mm	55 mm	4.1 mm	5.7 mm	8.1 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	120 mm	120 mm	64 mm	4.4 mm	6.3 mm	10.4 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	140 mm	140 mm	73 mm	4.7 mm	6.9 mm	12.9 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	160 mm	160 mm	82 mm	5 mm	7.4 mm	15.8 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	180 mm	180 mm	91 mm	5.3 mm	8 mm	18.8 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	200 mm	200 mm	100 mm	5.6 mm	8.5 mm	22.4 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	220 mm	220 mm	110 mm	5.9 mm	9.2 mm	26.2 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	240 mm	240 mm	120 mm	6.2 mm	9.8 mm	30.7 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	270 mm	270 mm	135 mm	6.6 mm	10.2 mm	36.1 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	300 mm	300 mm	150 mm	7.1 mm	10.7 mm	42.2 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	330 mm	330 mm	160 mm	7.5 mm	11.5 mm	49.1 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	360 mm	360 mm	170 mm	8 mm	12.7 mm	57.1 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	400 mm	400 mm	180 mm	8.6 mm	13.5 mm	66.3 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	450 mm	450 mm	190 mm	9.4 mm	14.6 mm	77.6 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	500 mm	500 mm	200 mm	10.2 mm	16 mm	90.7 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	550 mm	550 mm	210 mm	11.1 mm	17.2 mm	106 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter
Metric I Beam IPE	600 mm	600 mm	220 mm	12 mm	19 mm	122 kg/m	S235JR	S355JO	S355J2	304	304L	316	3 meter	6 meter

Figure C.1 – Data for IPE beams.(Matric, 2017)

Table C.1 – Properties of ASTM A36 steel.(Materials, 2017)

Chemical Composition

Element	Content
Carbon, C	0.25 - 0.290 %
Copper, Cu	0.20 %
Iron, Fe	98.0 %
Manganese, Mn	1.03 %
Phosphorous, P	0.040 %
Silicon, Si	0.280 %
Sulfur, S	0.050 %

Physical Properties

Physical Properties	Metric	Imperial
Density	7.85 g/cm^3	0.284 lb/in^3

Mechanical Properties

Mechanical Properties	Metric	Imperial
Tensile Strength, Ultimate	400 - 550 MPa	58000 - 79800 psi
Tensile Strength, Yield	250 MPa	36300 psi
Elongation at Break (in 200 mm)	20.0 %	20.0 %
Elongation at Break (in 50 mm)	23.0 %	23.0 %
Modulus of Elasticity	200 GPa	29000 ksi
Bulk Modulus (typical for steel)	140 GPa	20300 ksi
Poissons Ratio	0.260	0.260
Shear Modulus	79.3 GPa	11500 ksi

D. DIGITAL FILES.

- 20-Sim model of genset-module
- Drawing of frame and sub systems
- Matlab plot and developing Lagrange