

Procedure for torsional-vibration calculations in ice

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Faculté : Faculté des Sciences appliquées

Diplôme : Master : ingénieur civil mécanicien, à finalité spécialisée en "Advanced Ship Design"

Année académique : 2020-2021

URI/URL : <http://hdl.handle.net/2268.2/13312>

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Procedure for Torsional-Vibration Calculations in Ice

Submitted on 2nd August 2021

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ABSTRACT

The safety of navigation at sea is closely connected to the reasonable operation of the marine power transmission system of a vessel. The propulsion unit inside a ship is considered as one of the most significant on-board systems that is mostly subjected to dynamic loading. The dynamic behaviour can cause various vibrations along the propulsion line that will substantially affect the propulsion shaft's reliability and efficiency. In general, the most vital factor that can influence the shafting design is the shafting vibration behaviour, particularly the torsional vibration behaviour. The demand for proper vibration analysis grows rapidly in the last decade due to the growing adverse effects of fatigue failures in the shaft machinery systems, and also to define and identify the main torsional vibration characteristics as they differ from other types of vibrations to an extent. Hence the validity of a proper shafting design can only be checked through a detailed torsional vibration analysis.

This thesis focuses on developing a FEM based Torsional Vibration Calculation (TVC) method to identify and examine the vibratory stresses arising in the propulsion shaft line of a container vessel. The work starts with performing the TVC analysis of the vessel in open water conditions or normal deep-water conditions following the design loads proposed by Det Norske Veritas (DNV). A numerical model of the propulsion system is investigated and the results are compared with the analytical design approach. The condensed model is studied further to validate the effects of critical speed range and safety of shaft components by assessing the shaft response due to torque excitations.

Later, a new design approach of TVC analysis for the same vessel navigating in ice conditions is presented. A complete numerical based approach is being followed to examine the results. The aim is to improvise and model the ice-based propulsion system for different loading conditions and obtain the results in time-domain. The developed approach gives the dynamic response of the shafting system subjected to torsional vibrations for different engine operation conditions that are based on the Finnish-Swedish Ice Class Rules (FSICR).

Keywords: Torsional Vibration Calculation, Propulsion Shafting

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1. INTRODUCTION

In the ship building applications and design process of ships, the topic of ship and structural vibrations have a prominent role. The studies of vibration as an aid to the marine sector were already given paramount importance at the end of the nineteenth century itself, at a time when numerous shipyards and the marine industry were facing continuous problems regarding complicated mechanical vibration effects. Normally the analysis of vibration and its effects are to be carried out from the preliminary design phase to the operational stages of a vessel. Vibration has always been considered as one of the most severe problem on board ships, due its severe effects upon the ship's structure and upon the comfort of passengers and crew. Due to this fact, many shipbuilders and marine engineers are focusing and showing a great interest on this subject nowadays. Talking about ship machinery vibrations, it is equally important for every type and classes of sea going vessels. The scenario of vessel on-board vibrations is considered as a vital behaviour in recent times because of the fact that ships play an important role in the modern transportation system and world's trade.

The vibration levels in a ship are usually analysed by studying the behaviour of the complete vessel or a particular component to given excitations. The behaviour or response vibrations can be safe or even critical in some cases. In addition to the vibration, as an after effect, noises can also be propagated along the vessel causing problems and recently all the studies related to both these areas are given much importance on the basis of ship design. But this thesis deals with the fundamental factors of vibration and a further detailed analysis of torsional type of vibrations that can occur in the mechanical propulsion system of a ship.

1.1 General Overview of the Topic

This thesis is based on the vibration analysis, specifically torsional vibrations of the local shafts that can occur in the propulsion system of a ship. Recently, the importance of vibration study on vessels has been rising up and to cope up with that, a realistic and convenient approach can be developed, by making the 3D structure model preparation of the shafting arrangement and investigating further. Normally torsional type of vibrations is seen for all rotating machinery which include the shafting also. But in the case of large vessels that are installed with very long propulsion lines, many aspects of torsional vibration are similar to that of shaft vibration.

The torsional vibration is one among the most severe type of drivetrain vibrations that can happen to any rotating mechanical systems on board a vessel. This vibrational behaviour is induced as a result of the excitation of the natural frequencies of a mechanical system when a torque is being applied. The analysis of torsional vibration is somewhat easier for a single degree or two degree of freedom system, but when it comes to a system with many rotating components or rotary masses the analysis will be hard and specific methods should be followed to determine the natural frequency and angular displacements that are sufficient for torsional vibratory stress determination.

1.2 Scope of the Topic

The topic is focused on developing a FEM procedure for torsional-vibration analysis and study the propulsion characteristics of a shafting system in the design phase of shaft construction. In addition to the main task, it is also focused on comparing the developed numerical FEM model with a suitable analytical method of torsional vibration analysis. Nowadays, most of the shipyards, ship owners and even clients approach external design offices and consultancies which are specialised in performing the torsional vibrational analysis and corresponding calculations of a new building ship. This is because, the Torsional Vibration Calculations (TVC) are being done with the help of specialised software's and tools which are easily adaptable to different designs and programmed in a way to analyse specific shaft behaviours like bending and torsional characteristics. Hence the scope of this work is to develop a feasible FEM method for the TVC analysis that can be easily adaptable in the shafting design stage.

Nevertheless, the normal vibration analysis of a shafting arrangement can be done by following the guidelines that are provided by the respective classification societies, but the torsional-vibration calculations are little bit complex and also not much data have been provided by the class rules regarding this topic. Moreover, when it comes to the TVC analysis for an ice going vessel, it will be much more complicated due to the fact that, an additional ice excitation loading or force is formed in conjunction to the normal hydrodynamic loads at the propeller end. So, after taking all this into account, basically the TVC analysis of the propulsive shafting system of a vessel in open water conditions should be performed separately prior to the TVC ice class calculation. Furthermore, vibration calculations performing both in analytical and numerical

approaches gives a complete understanding about the differences and accuracy of the results at the end. This thesis is done in collaboration with Mecklenburger Metallguss GmbH (MMG), where all the practical works are carried out. For executing these tasks, the software's used in MMG for 3D modelling and FEM simulations are Siemens NX and ANSYS. The ANSYS Mechanical has been used for the structural dynamic analysis and simulations on the developed model.

1.3 Background Study and Motivation

In order to compromise the ride comfort and well-being of passengers and crew in a vessel, effective noise and vibration regulating measures needs to be implemented to ship structures. The vibration study of large and complex vessels is difficult, since the vibration characteristics exhibited by the complete ship hull vibration or individual components are quite different. Ship vibrations are generated from different sources that can either be formed on board the vessel or even from some external factors. The vibrations can be dangerous in some situations where it can cause rapid failures to structures, bearings, gears, shafts, machine parts and even for a whole machine in some cases. Over the past years, many researchers and marine engineers are focusing towards the various types of vibration behaviours, mainly torsional vibrations that can happen in a ship machinery while it is in operational condition.

1.3.1 Literature Review

Many extensive researches and experiments are conducting in this particular field of torsional vibrational analysis. However, the current developments on different types of vibration systems are not so promising and they are simply an extension of the traditionally followed methods. There are currently different researches that are being done with theoretical methods for determining natural frequencies and corresponding modes shapes of torsional systems, but publications related to the further torsional amplitude and vibratory stress calculations are minimal. Considering the fact that, every machinery is composed of some mechanical components, there will be a response from these parts when it is subjected to any kind of internal or external type of forces regardless of the method of application. Some papers related with the approaches for torsional vibration analysis in open water and ice water conditions and also discussing the various background and sources for shaft line vibrations are detailed below.

Adam Charchalis and Lech Murawski (2014) describes a simplified approach of torsional vibration calculation where they are able to determine the natural and forced torsional vibration frequencies and modes. It is given that the presented analysis is much faster and easier to perform for a power transmission system. Normally the torsional analysis for a marine power transmission system is conducted only by considering sufficient rotary masses, but in this paper, they have also considered the propeller added water moment of inertia.

The influence of engines and propellers are a must important factor to be considered in the torsional analysis. It is also important to understand that the drive trains or power transmission system of a machine or even in the case of ships can be either of direct coupled, geared systems, multiple drive shafts, branched and non-branched types. Gray C and Edwards A.J (1973) presented some interesting analysis technique in correlation with a program for the study of multi-branched systems with excitation forces or torques from engines, compressors and marine propellers with the effect of damping too. Even though, this approach is an effective one for iterated use with specific part analysis in a system. The torsional vibration analysis is normally solved or studied by using preferred mathematical models of a system. It was also noted that (Abhary, 1995) shows an approach for modelling lumped-parameter torsional systems. He used a graphical approach for the modelling and performing calculations for branched systems.

Wilson (1956) presents a more fundamental based review of the modern torsional analysis. He mentioned that the sources of dynamic torsional behaviour in machinery are the ones leading to failures in marine and aeronautical drive trains. Another well explained paper was about the different methods for modelling the torsional systems on basis of the equivalent mass inertias and torsional stiffnesses, which is proposed by (Nestorides, 1958). He referred the methods for a series of machinery parts like crankshafts, flywheels, etc. In his book, he describes a theoretical method called Holzer method which can be effectively used for branched and single line shaft systems.

The investigations related to the torsional vibration effect during ice impacts on a propeller blade are pretty much fewer. But Rosca Johan (2017) presents a study on the shaft line torsional vibration of an ice class vessel. This thesis paper mentioned about the different dynamic responses in a shaft line system which can be varying depending on factors like, the region where the vessel is navigating, cavitation effects on propeller and hull and ice navigation also.

The external induced loads like ice loads will be periodic but will vary as the duration of ice impact increase.

Barro and Lee (2011) describes about the sources of excitation for polar class propulsion systems and give a clear idea related to the propeller varying loads during ice passage. They defined that there can be contact and non-contact loads acting on a propeller in these conditions. The contact loads refer to the hydrodynamic loading on the propeller blades that are always present no matter where the vessel is operating (open water conditions) and the non-contact loads are the ones which are induced by the ice milling process or during ice-propeller interaction. They have mentioned and reported about the forced vibration analysis caused due to these excitation forces with the torsional stress level amplitude estimation.

1.3.2 Important Considerations of the Topic

An efficient and adaptive method to minimise or eliminate the vibrations on board is by properly designing and installing structures and machine components after following a proper vibration analysis procedure. Even though, there are some improvements in terms of machinery vibration as a result of the researches and studies done by different classification societies over the past years, still it is not clearly evident about a proper procedure or methodology to be followed for the distinct system vibrations. The data available are limited only for the normal preliminary vibrational analysis which is just the determination of natural frequencies and mode shapes of the shafting system. The changes made to the shaft or even to the ship (retrofitting cases) can seriously affect the vibrational characteristics.

Shafting vibration analysis in the vessels that are redesigned or retrofitted with new propellers are the area where the behaviour of whole shafting (bending line and other vibrations) can significantly vary. In this cases, proper vibration study needs to be conducted to facilitate the smooth operation of the shafting without any restrictions. Normally the vibrational analysis studies of vessels in open water conditions are available to a certain limit but for ice class or vessels operating in the polar regions, the information to conduct or perform a torsional vibrational analysis is not sufficient. The available results of the investigations and experiments are usually made into a simplified form for a practical view and also come into daily use in ship machinery design.

1.4 Objectives and Aims

The main objectives and aims for this thesis are detailed below:

- To understand the fundamentals of vibration phenomenon that can occur in the marine power transmission system of a sea going vessel. Following that, study and investigate the specific torsional vibration behaviour that can be typically found on a shafting system inside a vessel. In the first objective, the Torsional Vibration Calculation (TVC) of a 3000TEU Container vessel in open water condition (without ice condition) is to be investigated. The task was to develop and modify a numerical calculation method to determine the natural frequency and corresponding mode shapes of the shafting arrangement of the given Container vessel. The study then followed by comparing the numerical model results with the analytical results after conducting the simulation in a FEM environment. All the stress calculation, limit curves and design features have to be done and calculated by following the DNV classification rules.

The aim is to develop an idealised model of the complete shafting arrangement and use the different excitation forces to estimate the torsional stresses in various shaft segments for different engine firing conditions. All the calculations of torsional stresses and amplitudes in the considered shafting locations have to be estimated in frequency domain.

- To perform the TVC analysis for an ice-class vessel. The analysis should be carried out for the same 3000TEU Container vessel but here she had been retrofitted with a new propeller to make it able to operate in ice channels. The second objective is to study and analyse the retro-fit propeller characteristics and its influence on the torsional vibrational stresses on the shafting of the container vessel. In addition to the excitation loading from the main engine and propeller, an additional excitation force will be generated due to the propeller-ice interaction and this torque is to be estimated as per the Finnish-Swedish Ice Class Rules (FSICR, 2017).

Furthermore, a complete FEM numerical simulation has to be performed by taking into account of the external hydrodynamic force and the ice excitation force. For that an updated 3D model should be developed with the influence of propeller blades, as the ice impact have to be in correlation with the propeller blades to give the exact results.

In this task, the torsional stresses occurring at the shafting system has to be estimated for the three different operation conditions of the main engine of the vessel in conjunction with the different excitation cases of ice loading on the propeller. The main aim here is to perform time domain torsional vibration simulations and estimate the vibratory stresses for the propulsion shafting of the vessel. Validation of the applied analytical procedure will be done by comparison with the FE results in the two main tasks.

1.5 Structure of the Report

The thesis is outlined as follows,

- **Chapter 2** gives the state of the art regarding all the basic concepts, theories and information of vibration, influence of vibration on the propulsion systems in a ship.
- **Chapter 3** describes the main theory and important considerations of torsional vibration phenomenon, the vibration excitation sources, torsional stress limits, vibration damping and firing operations, which are all significant factors for a TVC analysis.
- **Chapter 4** presents the first task of the thesis work, where all the detailed methodology and final results for the torsional vibration analysis of the marine power transmission system in open water conditions (without ice conditions) are described. Moreover, some detailed concepts of the proposed analytical method are also introduced. Secondly, the topic of FEM modelling of shafting systems with different Degrees of Freedom (DOF) are also summarized.
- **Chapter 5** covers the second task of the thesis, where the torsional vibration analysis is performed for the vessel operating in ice conditions. A complete FEM based procedure is introduced and summarized in this chapter with all the relevant results according to the Classification Rules.
- **Chapter 6** shows the evaluation and interpretation of the results obtained in the TVC analysis for open water conditions and ice conditions. Moreover, a general overview of the possible counter measures for regulating the high vibratory stresses are also detailed.
- **Chapter 7** gives the conclusion and summary of the work conducted.
- **Chapter 8** shows some recommendations for the future works related with the normal torsional vibration calculations.

2. STATE OF THE ART

2.1 Basics of Vibration

Vibration can be defined as the dynamic behaviour of a system which indicates the oscillation of a system about its equilibrium position. A vibratory system can be depicted as a system with the means for storing potential energy (typically spring type elements), storing kinetic energy (mass, weight or inertia) and by which the energy dissipates (mostly by use of dampers). In normal cases, the vibration analysis for a system composed of many structural components or parts is complicated. There are some fundamental factors that can influence the vibration problems in general. These include the excitation loading, damping of the system, stiffness behaviour, and the mass of the system. All these can be compromised or altered to achieve a system with reduced level of vibrations. Based on these basic components in a vibrational dynamic system, the equations of motion representing the vibrations can be derived in a matrix form. A typical arrangement for a single degree of freedom system having translational motion is shown in Figure 1.

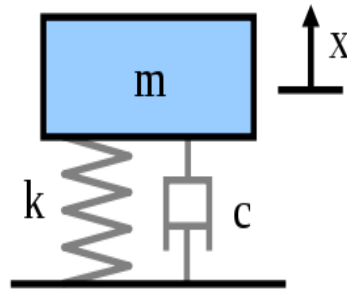


Figure 1: Mass Spring Damper Model System

The equation of motion formed based on the above arrangement is given below, where the mass, damping and stiffness coefficients are considered. This is the general ordinary differential equation of a simple mass spring model vibration happening at each instant of time. The mathematical equation of vibration for a single DOF system is given in Equation 1.

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t) \quad (1)$$

Where, m , c and k stand for mass matrix, damping matrix and stiffness matrix respectively. The differential components of $x(t)$, $\dot{x}(t)$ and $\ddot{x}(t)$ stands for displacement, velocity and acceleration vectors. $F(t)$ will be the externally applied force. This differential equation of a dynamic system is also formed by following the Newton's third law of motion. The above equation of motion is only used for a forced vibration system or we can say a system that is excited or vibrating with an externally applied force. Generally, the vibration analysis discussion is based on two general classes of vibration and that are discussed in the next sections.

2.1.1 Free Vibration Analysis

This type of vibrations normally happens when a system is oscillating or vibrating under its own internal forces or can be due to some pre occurred disturbances on the stationary system. Normally this analysis is performed by assuming the damping as negligible and there is not external load applied to the mass of the system. Some important considerations while doing a free vibration analysis is listed down.

- From a free vibration analysis, one or more natural frequencies of a system can be determined, which could be of different vibration modes.
- The dynamic system properties like eigen frequency and eigen modes can be clearly estimated.
- The natural frequency of a system clearly depends upon the mass and stiffness properties of the system.

Free Vibration Without Damping - Even though in free vibration, there will not be an influence of the external force, but there can be an influence of the damping in some occasions. For a system vibrating without the effect of damping, the ordinary differential equation of vibration will be modified and this is given in Equation 2 as follows.

$$m\ddot{x}(t) + kx(t) = 0 \quad (2)$$

Free Vibration with Damping - Normally there are various types of dampers that can be in corporate to a system to adjust the vibrational phenomenon. If the added damping to a system is very small or negligible, the system will be in vibration but eventually stops vibrating after

some time. This case is referred as underdamping. The case where the damping value is increased to a limit where the system no longer vibrates is termed as critical damping. If the damping limit exceeds beyond the critical damping, then it will be over damped system. These all varies according to the design and requirement criteria for different mechanisms and structures. Since in this case, there is an additional influence of damping is coming into action, the differential equation after summing all the forces on the mass can be written as shown in Equation 3.

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = 0 \quad (3)$$

2.1.2 Free Rotational Vibration Phenomenon

The scope of this thesis is dealing with the rotational or torsional type of vibration and rather than the translational type. Hence more importance and focus are given to this rotational mode of vibration. All the same concepts and principles apply to rotational vibrations as with translational vibrations, except with the Degrees of Freedoms (DOFs) and physical properties which are replaced with those associated to rotational modes (Antti Kangasperko, 2018). In rotational scenario, forces will be replaced by moments and the displacements will be considered as angular displacements, commonly denoted by angle θ which is measured either in radians or degrees from the initial resting position of the body before excited by a moment. The two main properties of these type of dynamic rotational systems will be the mass moment of inertia (related to the mass) and the torsional stiffness (related to the beam or shaft). Figure 2 presents an understanding about the torsional model system of a single rotor system, where a rigid disk is attached to the one end of a flexible shaft and the other end of the shaft is fixed.

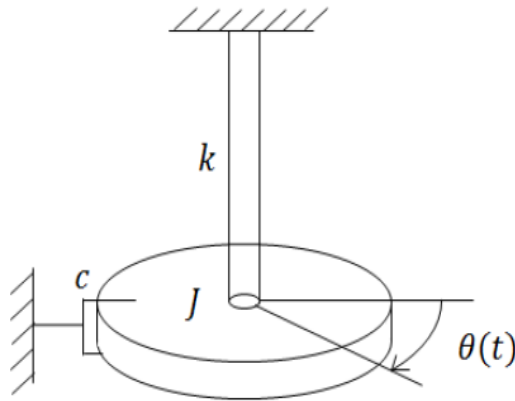


Figure 2: One Mass Rotor Torsional Vibration Model (Antti Kangasperko, 2018)

In the case of rotational vibration, the ordinary differential equation shown in the free vibration analysis of translational motion will be modified into a form as given in Equation 4. The equation of motion of these types of systems are generally represented in this format.

$$J\ddot{\theta}(t) + c\dot{\theta}(t) + k\theta(t) = 0 \quad (4)$$

Where, J , c and k stands for the moment of inertia, the damping and stiffness matrices. The different vectors like $\theta(t)$ is the angular displacement, $\dot{\theta}(t)$ is the angular velocity and $\ddot{\theta}(t)$ is the angular acceleration. The natural angular frequency of a single rotor system can be found out using the following Equation 5.

$$\omega_n = \sqrt{\frac{k}{J}} \quad (5)$$

Where, ω_n is the natural angular frequency in rad/sec, k is the torsional stiffness of the shaft in Nm/rad and J is the mass moment of inertia in Kgm².

2.1.3 Forced Vibration Analysis

The forced type of vibration occurs only after applying an external load or force to a body. This external disturbance excites the system and the body starts to vibrate in the frequency of the excitation loads. The main characteristics of forced vibration analysis is described below.

- If the excitation force or external force is harmonic, the system will be vibrating at a frequency called excitation frequency and while the system vibrates and when the excitation frequency coincides with any one of the natural frequencies of the system, resonance occurs. The resonant condition is the worst case where the system experience large amplitude vibrations and eventually it can lead to the failure of the structure. Most of the mechanical components that can undergo vibration at any stage in its lifetime, they are normally designed and constructed based on their resonance values.
- Frequency response of the system can be identified in a forced vibration analysis with or without the damping effects. The role of damping will be high in order to limit the amplitude of oscillation in case of resonant conditions.

2.2 Ship Propulsion System and Vibration

The ship can be considered as an elastic structure which will be forced to have elastic vibrations when subjected to some periodic forces that can arise within the ship or from external means. The marine propulsion system in a vessel is normally designed after considering numerous factors and parameters like ship type, class and capacity, material used for construction, purpose of the ship, vessel's travelling route, choice of prime mover and related components, etc. In the shipping industry, the engine room of a vessel is commonly referred to as the heart of a ship.

However, the engine can be the principal power producing unit, but the ship propulsion unit which is starting from the engine room and extend up to the aft part of the ship should also be considered as a supreme part of the ship. This is because of the fact that, the function of the propulsion unit is not only limited to ensure the motion of a ship from one location to another but also to regulate the vessel speed according to different operating conditions, navigating in shallow and deep channels, etc. Along with that, the vital factors that a propulsion unit influence in a ship operation can be further pointed out and some of them include pollution control, fuel consumption, the weight estimation and overall cost control of shipbuilding, etc.

Normally, the marine power transmission system on a ship transmits power from the main engine to the propeller. This power transmitted is converted into the thrust at the propeller side, which will enable the vessel to move forward. Typically, the power transmission system in a vessel consists of a set of rotary components with some couplings. The power transmission system will be made up of shafts, bearings, couplings, gear drive units, generator sets and a propeller. The thrust developed in the propeller is normally transmitted to the ship hull by the propulsive shaft through the bearings installed along the shaft line. As there will be so many rotating members in this power transmission unit, there can be local vibrations developing. Hence most of the marine engineers are focusing nowadays to give more importance to the free and forced local vibrations of the propulsive shafting system (Jong-Shyong Wu, 2013).

The marine power transmission unit of a vessel will be different for various powered ships. It will vary for slow speed, medium speed and high-speed engine installed vessels. The shafting arrangement of a vessel can be in different ways based on a series of factors which are explained above. There can be propulsion systems that are directly driven, geared systems, branched

systems, etc. For large sea going vessels, a typical propulsion shafting system is composed of main engine (including pistons, connecting rods, crankshaft and flywheel), flanges, gear boxes, intermediate shafts, bearings, couplings, propeller shaft and propeller. But in the case of small ships, the shafting arrangement will be installed with reduced number of components and parts. A typical arrangement of the shafting inside a vessel is depicted in Figure 3. The shafting line shown in Fig. 3 is a directly driven power unit where the crankshaft followed by intermediate shafts and finally the main propeller shaft is connected to the propeller as a straight-line system.

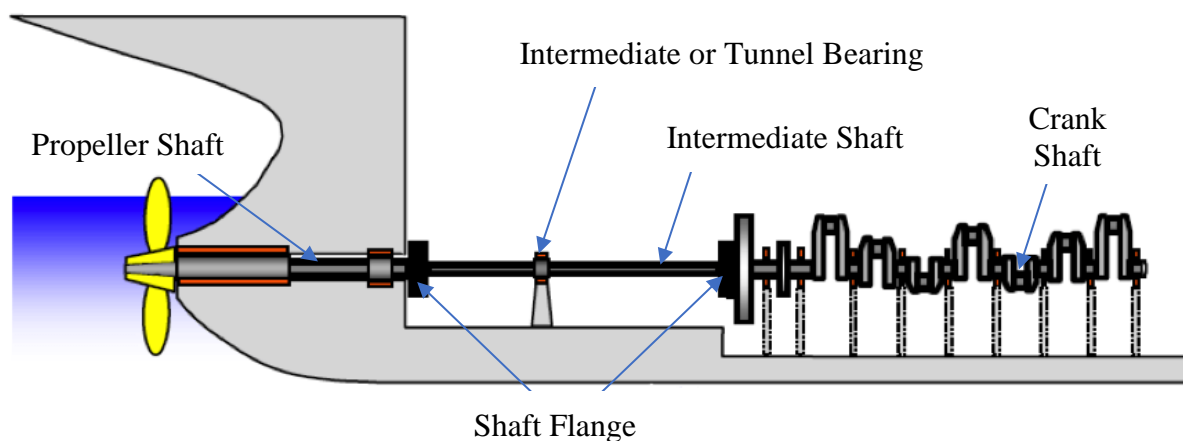


Figure 3: A Typical Power Transmission Unit of a Vessel (Ajay Menon, 2021)

The study of vibrations in these type of shafting systems is very much important in a ship design point of view. As there will be many reciprocating or rotary type of machinery involved in this power transmission, the magnitude of forces producing by these components will be large with lower frequency. The main parts that are involved in the vibration phenomenon or the parts which are having a greater influence on the vibration on a shafting are the combustion engine and its parts and the marine propeller. To analyse a case of vibration, it is required to identify the excitation sources of that particular type of vibration. The operation of the whole ship machinery system can cause different types of vibration on the propulsion shaft line and they are briefly discussed below:

- **Axial or Longitudinal Vibration:** The type of vibration can be caused due to the shaft misalignment (enable the back-and-forth movement of shaft along the axis). This type of axial mode of vibration makes the propulsion unit behave like a spring-mass system linearly. There will be multiple degrees of freedom linearly causing this mode of vibration to occur.

- **Lateral or Transverse Vibration:** This mode of vibration occurs in the transverse direction of the shaft. This means in a direction perpendicular to the axis of rotation of the shaft. Hence the bending phenomenon of shafts are mostly seen in this type of vibration. During the analysis of transverse vibrations, bearings play an important role as that they will be acting as support ends between the shaft line. So basically, the shafts are considered as beams. In the design phase of shafting, the number of shaft bearings and the placing of them is a crucial factor in the happening of this type of vibration.

- **Torsional Vibration:** Torsional vibration is an angular or rotational vibration that will happen along the axis of the shafting. Normally this vibration affects the whole shafting system of the propulsion unit. So, this torsional mode of vibration can affect the functioning and operation of almost all rotary systems that are participating in the power transmission process like engines, crankshaft, flywheel, flanges, intermediate shafts, bearings, propeller shafts, propeller hub and finally the propeller also.

3. THEORY OF TORSIONAL VIBRATION PHENOMENON

Mechanical vibrations are greatly affected on propulsion shafting systems. Among the different type of mechanical vibrations, the most important and prominent one will be the torsional vibration caused due to torsion. Torsion is an oscillatory angular motion that will result in the relative twisting in the rotating members of a system (Vishwajeet Kushwaha, 2012). A system subjected to continuous torsional vibration can lead to the twisting of the shafting arrangements and eventually at a critical time of its lifetime, cracks and fatigue starts to occur. This twisting of the shafts happens at fairly high frequencies and is driven by two different uneven torque forces originating from the two ends of the shafting. Figure 4 shows the fundamental principle of torsion in a shaft.

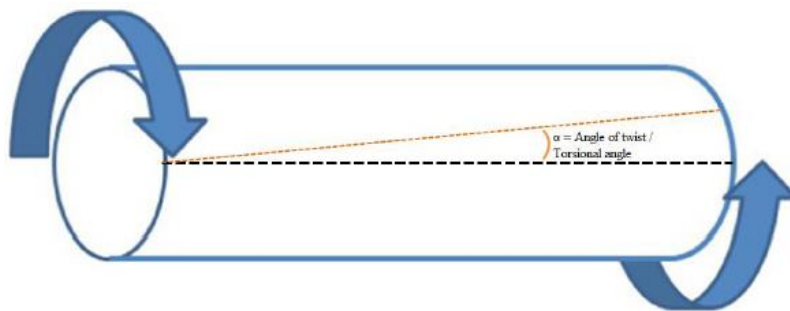


Figure 4: Basic Principle of Torsion in a Shaft (Jonas Holm, 2014)

Specifically, shafting torsional vibrations are usually originated by variable speeds of rotation of shafts. Compared with other types of vibrations like axial and lateral vibrations, torsional vibrations are normally undetectable, but it can cause serious damages in certain circumstances.

3.1 Torsional Vibration Sources

The torsional vibrations that are normally seen on the propulsion shafting of a vessel can be originated from different sources. The various type of forces from the ship structure or from any machinery components will lead to the twisting of the shafts, which results in the generation of shear stresses or also can be called as torsional stresses in this case. The most common sources of torsional vibrations occurring in the propulsion shafting are:

- Mass or inertia forces from reciprocating devices (e.g.: slider-crank mechanism inside an internal combustion engine)

- Varying gas forces developing inside an engine because of its firing order conditions (normal firing and mis-firing situations).
- Shock loads happening to electrical equipments (generator line defects or failure followed by the connected shaft line).
- Marine propeller inertia at the free end of the propulsion shaft (looks like a large mass acting at the free end of a beam).
- Varying and fluctuating torques from the gear box units, branched shafts, propeller blade excitations in sea water flow and special case of ice-propeller interactions.

From the vibration point of view, among these, only a couple of excitations will induce high torsional vibratory stresses in the ship propulsion system. The main and inevitable of these sources of torsional vibration excitations are discussed in this section. The first and foremost among them is the excitations that are originating from the diesel engine and these forces can cause serious vibration problems. There will be local vibrations (to be precise, variation of excitation torque) causing on the crankshaft which are normally created by the rotating and reciprocating parts in the engine cylinder. Due to that, the shaft loads will consist of a variable torque component, as well as a static torque component that will depend on the power transmitted and engine speed respectively.

The second major source of vibration causing torsional stresses in the propulsion shafting of a vessel is by the propeller induced excitations. The normal case of excitation from propeller will happen during the operation of the vessel, because of the hydrodynamic loading in the homogeneous wake field. But in this thesis, a special case of propeller excitation is also considered for the torsional vibration analysis, which is the ice excitation loads that are originating from the propeller-ice interaction in the case of ice class vessels operating in ice regions. This is also an important torsional excitation source that should be clearly analysed for the vibrations under ice conditions.

3.1.1 Diesel Engine Excitations

The excitations produced by the internal combustion engine of a ship are considered as the dominant vibration source not only for the torsional vibration of propulsion shafting but also for the whole ship hull vibration. Normally the excitations from a diesel engine are considered to have a set of three periodic forces and three periodic moments that will act on the foundation

of the engine (Soumya Chakraborty, 2019). The forces and moments that are originating inside a diesel engine will have a periodic nature. The forces acting along the axis of the crank shaft of the engine is taken as zero because it will be eventually cancelled out by the thrust. The other two force components are the one which are responsible for the engine excitations. Engine excitations normally begins only during the combustion process and the two forces responsible for the vibrations originating during this gas combustion in cylinders are the variable gas pressure forces and inertia forces. In addition to these forces, there will be some small excitations caused by piston slaps, diesel knock, faulty or worn parts, combustion problems during firing inside cylinders and so on; but their combined effects are small and hence they are neglected in the torsional vibration case. Figure 5 presents a diagram showing the different forces and moments that are acting on a marine engine.

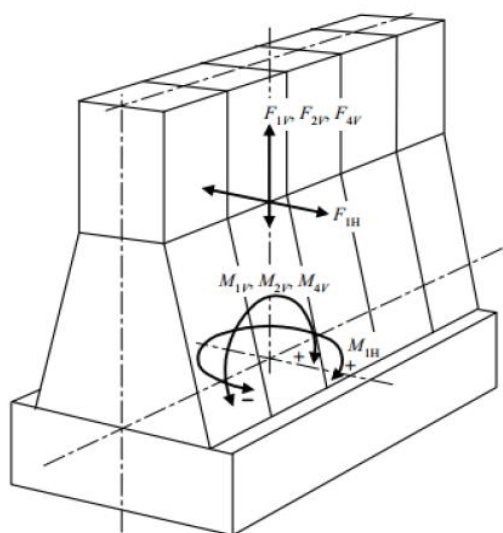


Figure 5: Forces and Moments Acting on a Marine Diesel engine (Mika Miettinen, 2019)

Depending on the engine model, excitation forces and corresponding moments can be varied. But in the torsional vibration point of view, the main sources of vibration considered for the vibration analysis is the gas forces which are due to the compression and combustion inside the engine cylinder and the inertia or mass forces due to the rotating and reciprocating parts inside an engine.

Both the forces (variable gas forces and inertia forces) have radial and tangential components. The tangential component of the cylinder gas force and mass force results in the cylinder torque. The tangential force will be having two components (constant part and variable part). The constant part contributes to the ship propulsion, whereas the variable part is the source for the torsional vibrations of the shaft line (Ivo Senjanović, 2019).

The frequency of the excitation forces cause by a diesel engine can be determined based on the running speed of the engine. For two-stroke engines, excitation frequencies happen at the rotational speed of an engine and its multiples and for a four-stroke engine they appear at the half order harmonics or vibration orders. This is because of the fact that in the case of four-stroke engines, the engine operating speed is twice the cycle speed (D Woodyard, 2009). The general formula for the excitation frequency calculation is shown in Equation 6.

$$f = k \times \frac{n}{60} \text{ Hz} \quad (6)$$

Where, k is the firing order of an engine cylinder ($k = 1, 2, 3, \dots, N$ for two stroke engines and $k = 0.5, 1, 1.5, 2, 2.5, \dots, N$ for four stroke engines) and n is the operating speed of the engine in revolutions per minute.

The excitations produced by different type of engines are different. The engine of the Container vessel presented in this thesis is a marine slow speed diesel engine. Talking about slow speed diesel engines, they normally operate approximately at a speed range between 60 to 130 RPM. These engines will induce both horizontal and vertical first order moments. Also the main interest will be the fact that in most cases, the engines are directly connected to the propeller shaft. A clear idea about the different modes of vibrations due to gas forces and inertia forces in a structure are given in Table 1.

Table 1: Summary of Engine Excitation Forces

Excitation Force	Mode of Vibrations	Influence on Structure
Gas Force	<ul style="list-style-type: none"> Torsional vibrations Engine body vibration Axial vibrations 	<ul style="list-style-type: none"> Strength of the structure depends on engine load. Considered as the most significant engine excitation source.
Inertia Force	<ul style="list-style-type: none"> Torsional vibrations Bending mode vibrations Rigid body vibrations 	<ul style="list-style-type: none"> The strength of the structure depends on the operating speed. The engine balance design have an influence on dominant vibrating modes.

3.1.2 Inertia Forces

The inertia forces or mass forces developing inside an engine cylinder is basically the reaction force of the moving parts. The mass force is considered as the sum of reciprocating and rotating masses inside an engine. Since an internal combustion engine follows a typical slider crank mechanism, the reciprocating parts consists of the piston body and the rotating mass consists of the connecting rod that is attached with the crankshaft. After considering the masses of each part that are significant for rotating and reciprocating movements, the forces exerted by these components are typically called as oscillating forces. The general equations used for the calculation of these oscillating masses are given in Equations 7 and 8.

The force exerted by the rotating masses is determined by the following formula,

$$F_{rot} = m_{rot} \times r \times w^2 \times \cos \varphi \quad (7)$$

Where,

m_{rot} is the rotating mass in Kg

r is the crank radius in mm

φ is the crankshaft angle in degree

w is the angular frequency in rad/sec

The rotating force component acts in the outward direction from the crank shaft or it is acting radially outwards causing the rigid body motion of the engine body.

The force exerted by the reciprocating masses inside an engine is given by,

$$F_{rec} = m_{rec} \times r \times w^2 \times (\cos \varphi + \lambda \cos 2\varphi) \quad (8)$$

$$\lambda = \frac{r}{l} \quad (9)$$

Where,

m_{rec} is the reciprocating mass in Kg

λ is the connecting rod ratio

l is the length of the connecting rod in mm

The reciprocating force exerted mainly by the moving piston is acting vertically along the axis of the engine cylinder. These forces are the main reason that causes bending and torsional vibrations. The acceleration of the reciprocating engine parts results in very high inertia forces as these rotating parts can be of high mass. Hence for the inertial/mass force calculation, the reciprocating mass force is chosen and this force can be further decomposed into different orders or coefficients based on the connecting rod ratio. But for typical engine pressure calculation, only the first four order coefficients are taken and remaining higher order components are neglected due to its insignificance in the total engine torque. The representation of the reciprocating and rotating forces that are induced on the engine cylinder is shown in below Figure 6.

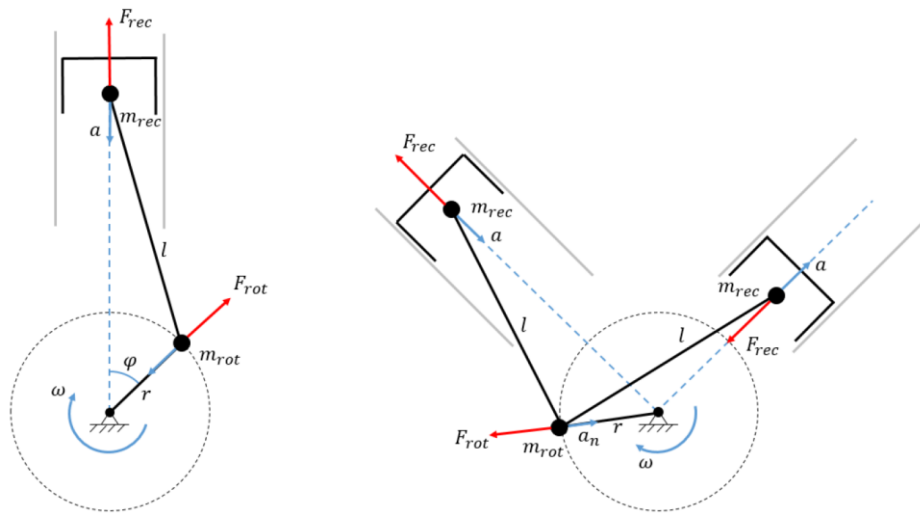


Figure 6: Different Excitation Forces Acting in an Inline and V-Type Diesel Engines (Antti Kangasperko, 2018)

Hence the different engine oscillating forces are known, the pressure developing inside a cylinder can be calculated easily with the data of piston surface area. Finally, the reciprocating mass force is transmitted to the crankshaft through the connecting rod and correspondingly an engine torque will be produced at the flywheel end. This torque is further used for the torsional vibration stress calculation.

3.1.3 Gas Forces

The gas forces are induced by the combustion process happening inside an engine cylinder. During the process of combustion, high pressure will be developing inside a cylinder and this pressure forms the gas forces. The pressure exerts forces on the piston, cylinder head and also

on the cylinder walls (Antti Kangasperko, 2018). Generally, the force exerted by these gas components can be obtained using the following formula,

$$F_{gas} = P_{cyl} \times A_{cyl} \quad (10)$$

Where,

P_{cyl} is the cylinder pressure in MPa.

A_{cyl} is the area of the piston surface in mm².

Similar to the mass forces, gas forces are also playing an important role in generating the engine torque at the crankshaft. Here also the gas forces developed at the piston surface are transmitted through the connecting rod to the crankshaft to get the torque. Normally the torque analysis from gas forces of a diesel engine is a very complicated process, due to the inability or deficiency to use the harmonic order functions to represent the real scenario of periodic nature of process but happens in the crankshaft rotation angle. Some other additional problems which again make this internal source excitation more complex is the fuel injection phenomenon into the engine system, possible change in torque for the individual cylinders, etc.

Hence in most cases, the gas force values or the gas harmonics data required to calculate the torque excited due to the gas forces are obtained from the engine manufacturer. The values related with the consecutive harmonics and phase angles are commonly used in the gas harmonics data. This complex gas forces or pressure waves can also be decomposed by using Fourier series transformation of the measurement series, here the cylinder tangential force can be expanded into Fourier series. but with the mean effective indicated pressure and its harmonic data. The gas tangential pressure can be estimated with the mean effective pressure coefficients from harmonics data. In the case of mass tangential pressure, the data required for the pressure calculation is simply the engine specifications with the maximum continuous rating speed.

3.1.4 Propeller Excitations

The propeller induced excitations normally consists of varying pressure on the ship hull, displacement effect of propeller blades on the rudder, cavitation of the propellers and normal hydrodynamic loading effect and ice induced loads on the propeller blades. In torsional vibration point of view, hydrodynamic loading, propeller inertia around the shaft rotation axis

and ice induced loading are the ones having a greater influence on the vibration characteristics of the shafting system. The main cause for propeller induced vibration is due to the primary contact between propeller blades and water while the ship is operating. As discussed in the section of engine excitations, the exciting frequency or blade frequency of the propeller can be determined by using a simple formula as shown below.

$$f = k \frac{n \times z}{60} \text{ Hz} \quad (11)$$

Where, k is the firing or harmonic order ($k = 1, 2, 3, \dots, N$ for two stroke engines and $k = 0.5, 1, 1.5, 2, 2.5, \dots, N$ for four stroke engines), n is the propeller rotating speed in revolutions per minute, z is the number of propeller blades. Fluctuating propeller loads can be formed during the propeller and water interaction. The non-uniform wake flow is the primary reason for fluctuating torques from marine propellers. Figure 7 depicts the different forces and torques acting on a marine propeller. The torque acting around the rotation axis of the propeller shaft (denoted in green colour) is the one having influence on torsional vibration on shafts.

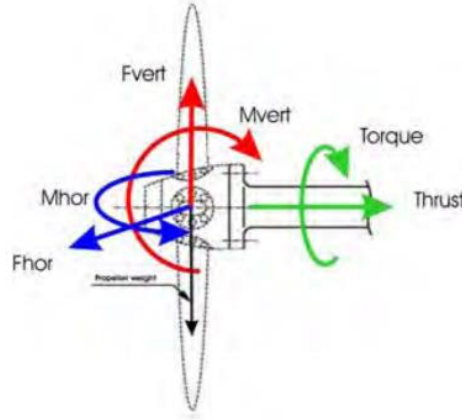


Figure 7: Propeller Hydrodynamic Loads (Soumya Chakraborty, 2019)

The ice induced excitation will be caused due to the propeller-ice interaction. Due to this interaction, there will be a torque formation on the propeller which is in correlation with the duration of ice impacts by the propeller blades. As there will be a non-uniform torque developed in the propeller side and this can induce a torsional vibration in the propulsion shafting of a vessel. Hence for those type of vessels, an additional torsional design load should be estimated to ensure the safety of shafting arrangement.

3.2 Torsional Vibratory Stress Limits

Torsional vibration in a ship propulsion system causes vibratory stresses in the propulsion line. Like all other vibrations, torsional vibration also has defined maximum stress limits that should be followed while designing a shaft segment in the propulsion system. These permissible limits are normally provided by the Classification societies under torsional design loads of shafting. Prior to the TVC analysis of a vessel, a background study regarding the torsional vibratory stress limits in different shaft segments are important. As it is important to identify the maximum permissible stresses a shaft can withstand and for that DNV specifically provides the allowable limits of torsional vibratory stresses for engine crankshafts, intermediate shafts and propeller shafts.

It is recommended by the Classification societies that the vibratory stresses for each shaft type have to be under the permissible values of two stress limits such as:

- Lower stress limit (τ_1) is the maximum allowable stress level for the continuous operation of an engine. This is applicable for the entire engine speed range.
- Higher stress limit (τ_2) is the ultimate limit state under transient engine operating conditions. This is applicable only for the 80% of the engine's Maximum Continuous Rating (MCR) speed. This stress limit should not be exceeded in any case.

Another advantage of these stress limit curves is the identification of the barred speed range of the vessel. This is a speed range that can possibly occur in the engine operating speed while the vessel is seagoing. At this particular range, the torsional stresses in the shafting will be higher (around main torsional resonance condition), where the excitation frequency in the shafting will be same as to the natural frequency of the shafting. The torsional stress limits of different shafts will be different due to some factors like the general materials used for the construction of shaft, dimensions and intended operation, purpose, form factors based on shaft type and design, size factor, etc. These limits are not constant and varies as a function of the engine speed. The classification society have provided defined formulas to calculate the two stress limits and they are given below.

$$\tau_1 = \pm c_W \times c_K \times c_D \times (3 - 2 \times \lambda^2) \quad \text{N/mm}^2 \quad (12)$$

For speed ratio values $\lambda < 0.9$

$$\tau_1 = \pm c_W \times c_K \times c_D \times 1.38 \quad \text{N/mm}^2 \quad (13)$$

For speed ratio values $0.9 \leq \lambda \leq 1.05$

$$\tau_2 = \pm 1.7 \times \frac{\tau_1}{\sqrt{c_K}} \quad \text{N/mm}^2 \quad (14)$$

Here, d – Shaft diameter (mm)

λ – Speed ratio $\left(\frac{n}{n_0}\right)$

n – Speed (rpm)

n_0 – Nominal speed (rpm)

R_m – Tensile strength of shaft material (N/mm²)

c_W – Material factor $\left(\frac{R_m+160}{18}\right)$

c_D – Size factor $(0.35 + 0.93 \times d^{-0.2})$

c_K – Form factor

The different permissible torsional stress limit curves in the shafting system in accordance with these formulas for shaft materials with a tensile strength of 450 N/mm² is depicted in Fig. 8.

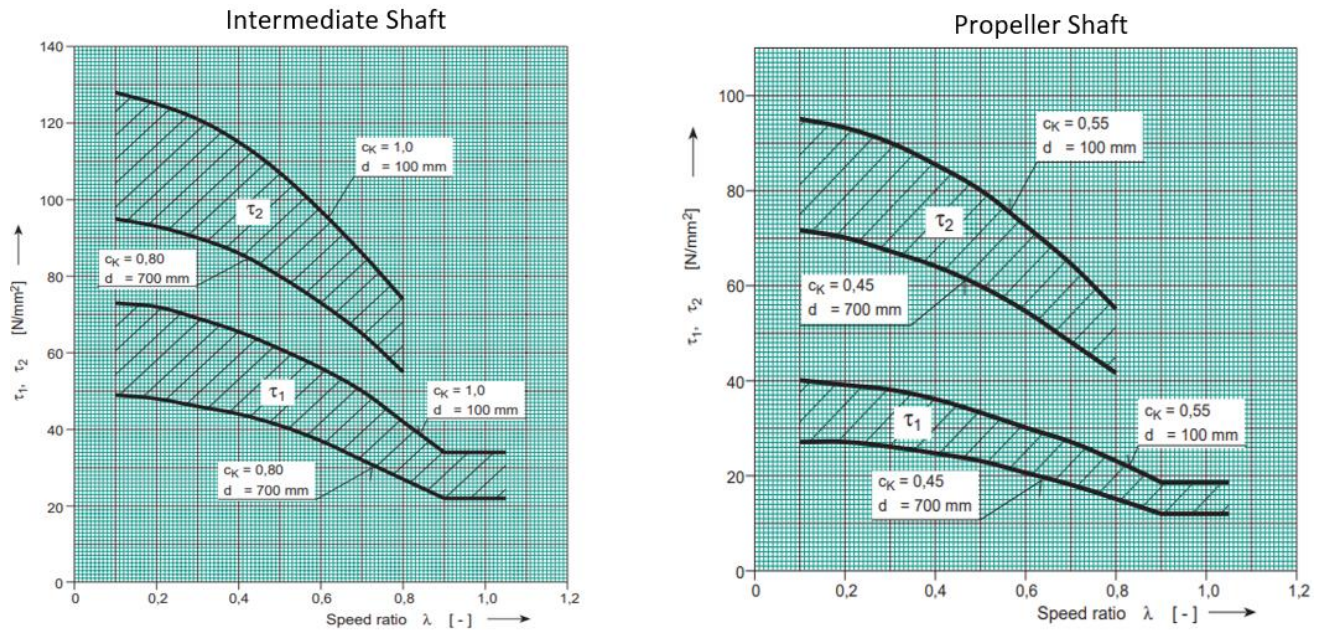


Figure 8: Limit Curves of Intermediate Shaft and Propeller Shaft (DNV-GL, 2008)

3.3 Damping in Torsional Vibration Phenomenon

Regarding the torsional stresses in a propulsion system, the effect of damping factors is also usually considered. The torsional damping determination is still a great task for the marine shafting designers. It is only possible to identify the damping data introduced to be correct, by the validation of results by means of measurement on-board (Batrak, Y., 2011). Damping is a method that tends to suppress the vibratory amplitude of an oscillating system. On basis of damping specification, the accuracy of torsional stress determination is lower in comparison with the natural frequency calculation even for the numerical methods also. The total damping in the dynamic system has a very small influence on the stress amplitude outside the resonance range. For a torsional vibration in the marine power transmission unit, the typical Damping Coefficient (DC) values will be at a range between 3 and 5% (Lech Murawski and Adam Charchalis, 2014). If there is no damping around the main resonance of the propulsion system, the stress will rise to infinity value. A clear understanding of the significance of damping coefficient in resonance curve of the power transmission system is illustrated in Figure 9.

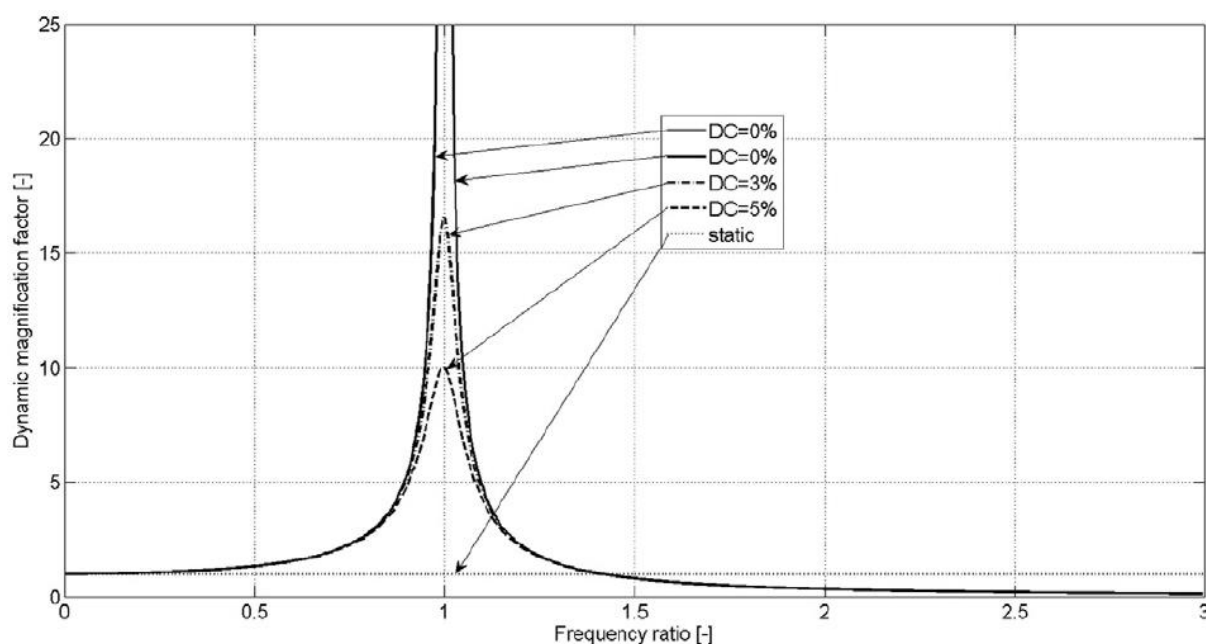


Figure 9: Resonance Curve of the Power Transmission System with Different Damping Coefficients
(Lech Murawski, Adam Charchalis, 2014)

In most cases, the dampers are intended to protect the engine machinery and not necessarily the driven shafts. The excitation forces are mainly originating from the engine side and due to that the location of dampers are often be at a point with high angular velocity, normally near the

front end of the crankshaft of the engine. For the marine propulsion torsional vibration calculation, four main types of damping are significant (Batrak, Y., 2011):

- a) Viscous damping
- b) Fluid damping
- c) Internal damping
- d) Structural damping

Viscous damping is caused due to the loss of energy happening in the lubricating liquid between the different parts that are in relative motion. This damping is directly proportional to this relative velocity of the dynamic system. The viscous damping is considered as absolute damping, if it is between a moving part and a stationary environment whereas it can be also relative damping, if it is between two parts that are in relative motion (Batrak, Y., 2011). Viscous type of dampers is commonly used in reciprocating engines which will help to limit the torsional vibration and crankshaft stresses to a large extent (Troy Feese and C. L. Hill, 2009).

Fluid damping is formed by the hydrodynamic interaction between the propeller and the surrounding water medium. The reason for internal damping or material damping is the energy loss within the material of the shafting, material of couplings or even in the torsional vibration dampers. As the shaft components are mechanically connected to each other through couplings and flanges, this mechanical energy dissipation happens during its operation and that leads to the material damping by itself. The resistance or relative friction between the shafting elements that are in direct contact causes the structural damping (Batrak, Y., 2011).

From all the above-mentioned damping types, only the linear viscous damping type enables a simplified analytical calculation method and so all the remaining damping types are in practice remodelled to the equivalent viscous damping. This makes the fluid damping as absolute viscous damping, internal and structural damping as relative viscous damping (N. Vulić, 2018). The damping data provided in various torsional vibration calculation programs developed by some engine manufacturers, shafting design offices and even some shipyards can be having slight differences. Owing to the fact of this difference, the following definitions like absolute torsional damping (in % of critical damping) and physical damping (between the actual and previous inertia) have been used in the presented TVC analysis.

For the engines and cylinders the most reliable damping data are provided by the engine suppliers where it is measured for each engine. The propeller damping also has a major role in TVC analysis and here the propeller damping is being presented as an equivalent absolute viscous damping. Two often used methods for this damping calculation is Frahm approach and Archer approach. In Archer approach, the propeller damping is taken as a resultant of the propeller's torque characteristics. The propeller law is used to estimate the damping coefficient. There are also several formulas for estimating propeller damping known from publications and here the dimensionless equations used to calculate the damping coefficients are the one implemented by Archer's damping factor.

3.4 Influence of Firing Conditions in TVC

One other main factor that can influence and determine the intensity of torsional stresses (vibratory stresses) in the propulsion shafting is the firing order of an internal combustion engine. Firing order is the sequence or order by which ignition of fuel happens inside the engine cylinders. The firing order greatly affects the vibration, sound and even the final power output from the engine. Normally, the propulsion engines inside a vessel will consist of a number of cylinders and the total engine torque that is transmitted to the crankshaft is the summation of all the simultaneous events (firing order) of all cylinders, by reckoning the phase angle between them due to the firing order (Gojko Magazinović, 2002).

Typically, a marine diesel engine can be subjected to two kinds of engine firing conditions. The first one is the normal firing condition where all the cylinders are in operating condition by the correct setting of fuel supply into each cylinder by an injector. The burning and combustion of the air-fuel mixture takes place in all the cylinders in this condition. The latter one is the mis-firing condition, where a cylinder that isn't firing or it is not contributing to the overall power output from the engine. Figure 10 shows the cumulative engine torque for a five-cylinder engine for both conditions. The blue line represents the normal operation whereas the red line denotes the case when cylinder 3 lacks ignition or mis-firing.

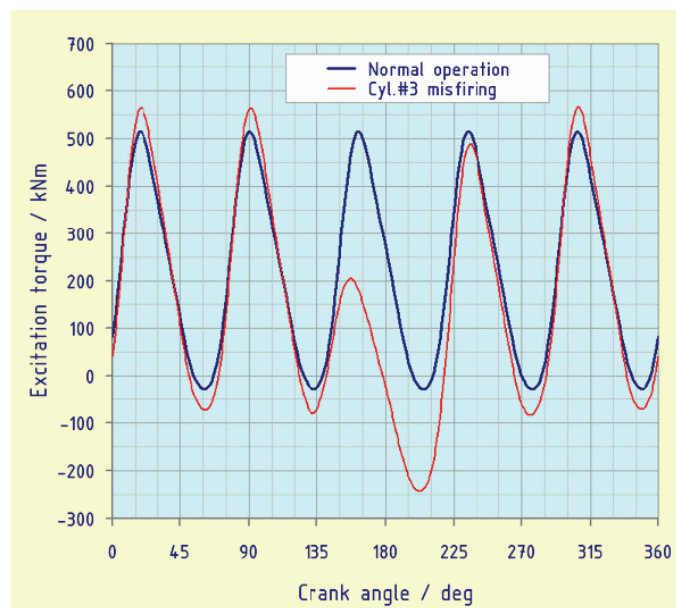


Figure 10: Excitation Torque in a Typical Five Cylinder Diesel Engine (Gojko Magazinović, 2002)

In mis-firing case, the remaining cylinders are forced to work in addition to compensate the resulting cylinder pressure on to the crankshaft. This leads to the rise in stresses by having a strong counter torque and eventually the varying torque will be transmitted through the shafting up to the propeller. This initiates higher torsional stresses in the complete propulsion shafting. Hence the critical influence of normal firing and mis-firing situations in the torsional stresses in the intermediate shafting of a propulsion unit is shown in Figure 11.

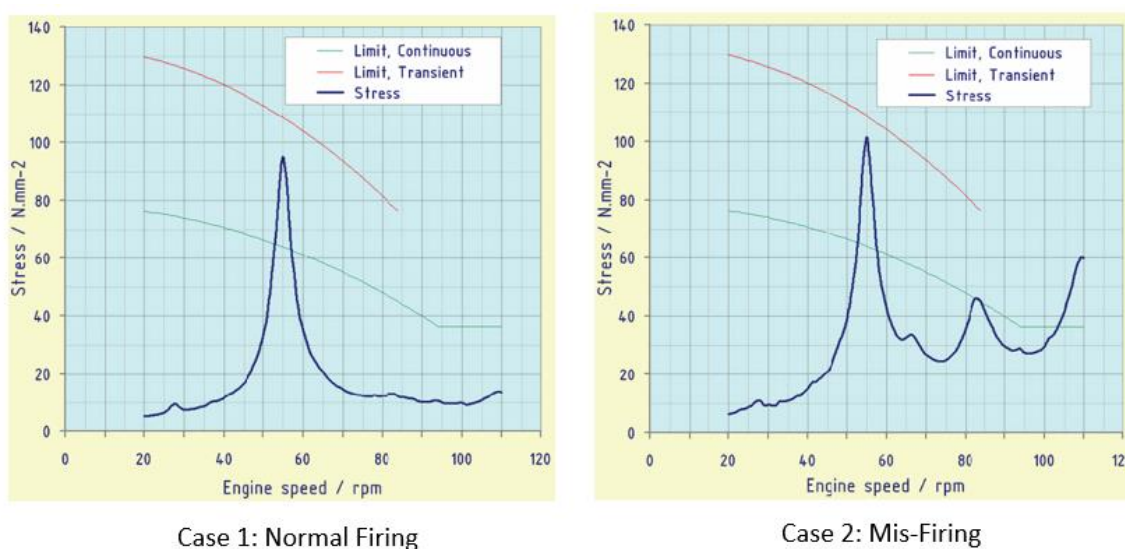


Figure 11: Influence of Firing Conditions in the Vibratory Response for Intermediate Shaft of a Conventional Low Speed Diesel Engine (Gojko Magazinović, 2002)

4. TORSIONAL VIBRATION ANALYSIS IN OPEN WATER CONDITION

4.1 Problem Description

During the operation stage of a sea going Container vessel, the propulsion system of the ship can be subjected to various excitation loads and hence it is made to investigate and analyse the torsional vibration characteristics of the shaft line. The vessel is retrofitted with a new propeller, optimised to a changed operation profile of the ship. The new propeller has been designed for slow steaming with decreased engine power. Because of the changed propeller characteristics, the vibration behaviour of the shafting line deviates from the original condition. Hence the objective is to calculate the torsional stresses in the new propulsion line of the container vessel having re-designed propeller. The normal TVC data of the shafting with the old propeller is previously done by an external agency and already available as a prerequisite. Due to the various loads, there can be significant vibrations at different locations in the shafting and hence the task is to evaluate these vibratory stresses developing at corresponding shaft segments (crankshaft, intermediate shaft and propeller shaft).

The vessel can also experience a jerk in the shafting system while the engine is operating, probably it occurs at the time when the stresses at the shafting are very high or we can say the vibrations in the rotating shafts are extreme at that situation. Hence it is also required to determine that specific engine speed or speed range where the maximum shafting vibratory stresses can occur. While analysing this critical speed range also known as the barred speed range, the vessel will be in operating condition, and bearing in mind to avoid this speed range as quick as possible to eliminate the higher torsional stress development in shafting.

As the firing conditions inside an engine can be varying in normal practical cases of a ship operation scenario, it is also required to investigate the torsional stress calculation analysis for both normal firing and mis-firing conditions of the vessel. All the torsional vibration responses in the different shaft segments of the ship propulsion system have to be evaluated for these two firing conditions.

4.2 Vessel Considered for the TVC Analysis

This thesis aims at analysing the vibration behaviour of the propulsion system of a 3000TEU Container vessel. This vessel taken for the TVC analysis is an ice-class vessel operating in the Northern Baltic Sea region. The vessel was retrofitted with a new re-designed 5-bladed propeller capable to operate in polar regions. In the case of torsional vibration calculation, the main point of interest will be the complete shafting arrangement starting from the engine side to the propeller end. The vessel taken for the study and TVC analysis is shown in Figure 12. As the propeller is replaced, the propeller properties will change and this can cause alterations in the torsional stresses that are developing along the shaft line.



Figure 12: 3000TEU Container Vessel

Sea going vessels like container ships normally operate at a speed range well below the Maximum Continuous Rating (MCR) speed of the engine and the main engines are regulated to operate below capacity to ensure fuel consumption over long distance voyages. The fuel consumption by container ships is influenced mainly by its size and speed. The TVC analysis also ensures the calculation of the critical speed range (indicates the region where the natural frequencies of the shafting and the excitation force coincides) that a vessel should avoid during its voyage. The vessel performance is normally monitored under different engine operating conditions and identifying the critical condition (dangerous for propulsion shaft) is significant.

4.2.1 Characteristics of Propulsion System

The power transmission system of a 3000TEU Container ship, equipped with a slow-speed two-stroke seven-cylinder main engine - MAN B&W 7L80 MC type, and a five-bladed fixed pitch propeller, is considered. The engine power is $P_W = 21700$ kW and the nominal speed $N_0 = 88$ rpm. The power transmission unit equipped with the slow speed diesel engine have a short and directly coupled straight shaft line. The main characteristics of the propulsion system are presented in Table 2.

Table 2: Characteristics of Main Engine and Propeller

System	Characteristics	Value	Unit
Engine	Type	MAN B&W 7L80 MC	
	Maximum continuous power, P_W	21700	kW
	Nominal speed, N_0	88	rpm
	Number of cylinders, z_c	7	-
	Cylinder bore diameter, D_{cb}	800	mm
	Cylinder stroke, s	2592	mm
	Mean indicated pressure, p_0	2.20	N/mm ²
	Connecting rod ratio, λ	0.4150	-
	Oscillating mass per cylinder, m_{osc}	12413	Kg/cyl
	Crankshaft diameter, D_{cs}	820	mm
Propeller	Propeller diameter, D	7700	mm
	Number of blades, z	5	-
	Expanded area ratio, EAR	0.500	-
	Mean pitch ratio, P/D	0.976	-
	Propeller shaft diameter, D_{pr}	715	mm

4.3 Methodology for Analytical Approach

Preliminary analysis should be done on the simplified shafting system considering all the rotary components that can affect the transmitting torque on the shafting system. The methodology followed to do the TVC analysis of the shafting systems of the Container vessel with the new propeller in the open water conditions are discussed below.

- a) The first step in analytically determining the torsional/vibratory response is by calculating the torsional natural frequencies of the system. To calculate the natural frequencies, the mass moment of inertia and stiffness properties of each component and shafts should be required. These parameters can be well estimated from the mathematical model (lumped-mass model system or discrete-parameter system) of the propulsion shafting arrangement of the vessel.
- b) The main engine data, propeller data and gas harmonics data (obtained from the engine manufacturer for both normal firing and mis-firing) can be used for the forced torsional vibration analysis using a proper analytical scheme. A proper and effective method called Benz scheme can be implemented for the free and forced vibration analysis with the effect of damping also.
- c) With the correlation between the material data, damping factors and external moments (caused due to the oscillating masses inside the engine - explained in Section 3.1.1), the torsional stresses can be estimated for each shaft components in the system.

4.4 Modelling Method and Natural Frequency Estimation

4.4.1 Lumped-Mass-Elastic System

The propulsion system of the considered container vessel is a directly coupled system which means the engine unit is connected directly to the propeller without any means of gear box or other drives. Here the low-speed diesel engine is in direct connection to a fixed pitch propeller through a couple of straight shafts with some couplings and bearings. A clear diagram of the complete shafting arrangement of the container vessel taken for the TVC analysis is shown in Figures 13 and 14. The propulsion system consists of a generator shaft, intermediate shaft and the main propeller shaft. The generator shaft is mounted with a diesel generator and the generator shaft and intermediate shafts are coupled by a flange coupling. To ensure proper support and maintain the correct position of the rotating shaft, two bearings (tunnel and plummer bearings) are placed at proper distance along the intermediate shaft. In between the intermediate shaft and propeller shaft, another flange coupling is being installed to ensure that the two shafts rotate together to transmit motion and torque. Finally, the propeller shaft is connected to the propeller hub via a keyless fitting through the stern tube.

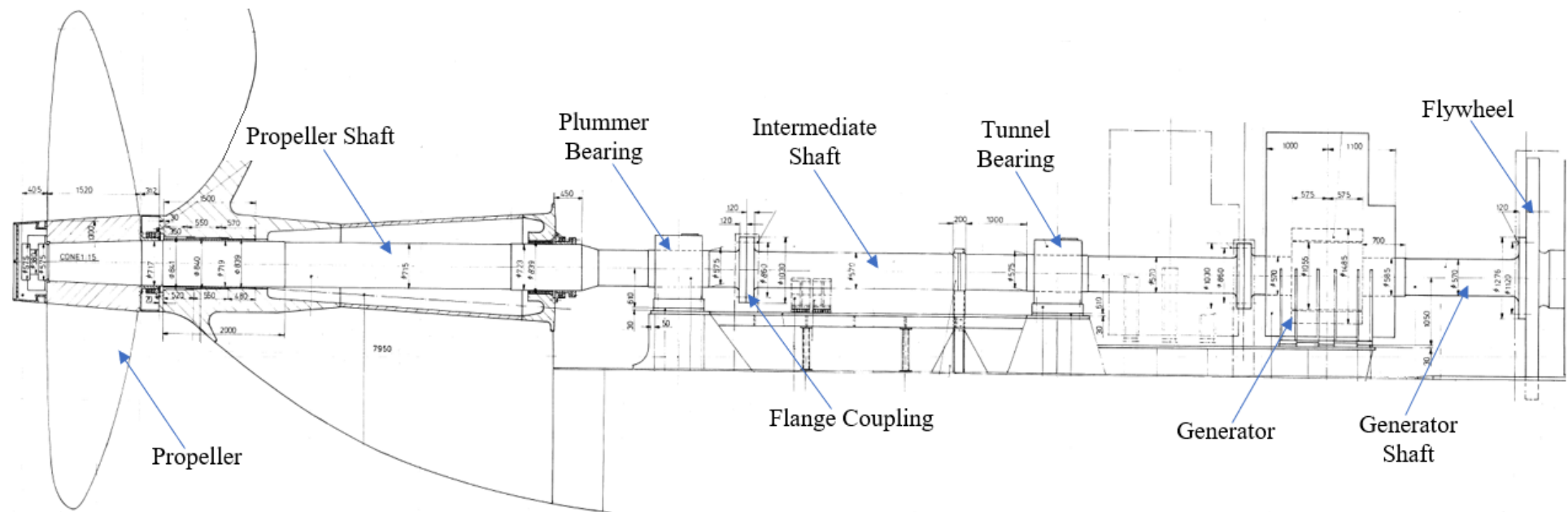


Figure 13: Shafting Arrangement of the Container Vessel (Layout from Flywheel upto Propeller)

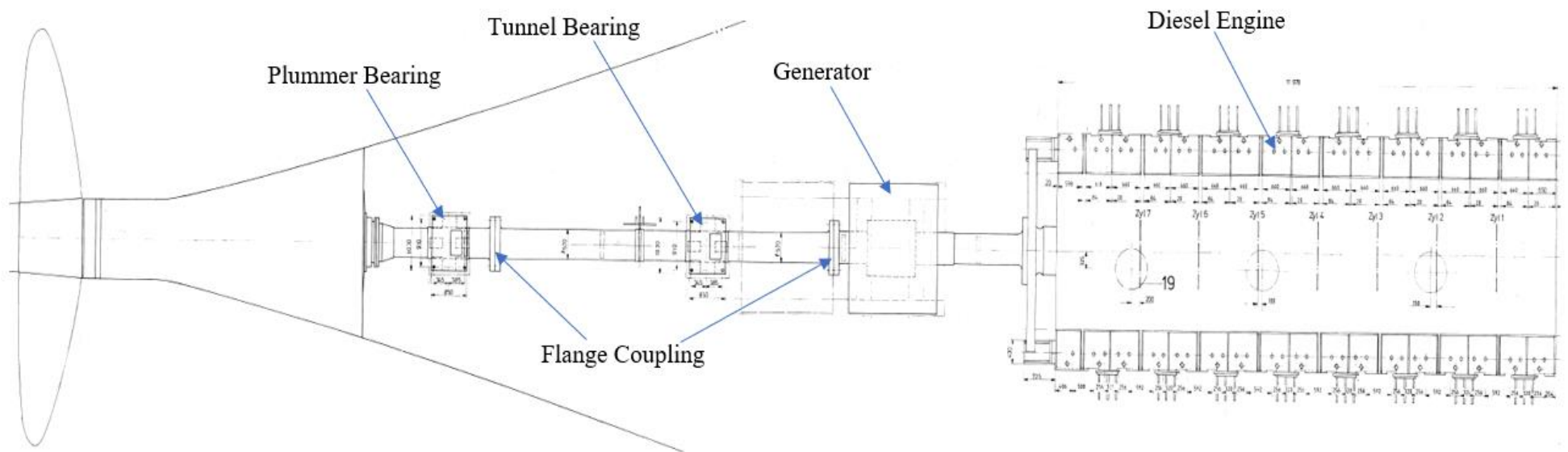


Figure 14: Shafting Arrangement of the Container Vessel - Topview (Layout from Engine Block up to Propeller)

To begin with the analysis the first step is to simplify the original shafting system having distributed characteristics of a continuous system into a discrete-parameter system. So, to proceed with the calculation of the torsional natural frequencies of the whole system, a mass-elastic model for the system is required. In order to do so, the reciprocating and rotating masses of the engine including crankshaft, generator shaft, intermediate shaft, propeller shaft, couplings, bearings and propeller are modelled as a system of rotating masses or inertias connected by the torsional spring (Lech Murawski and Adam Charchalis, 2014). This approach of reducing the system configuration or remodelling the structure of the system is done properly to keep the vibration characteristics unchanged. This type of simplified mathematical model is called as a lumped-mass system or lumped parameter model.

In the case of the Container vessel propulsion plant, the lumped mass model will consist of 14 inertias and 13 elastic shafts as shown in Figure 15. In the simplified model, the crankshaft is represented by the first 10 elements that consists of 7 cylinders and a moment compensator, followed by a thrust bearing, flywheel and then by generator shaft, intermediate shaft and the propeller shaft.

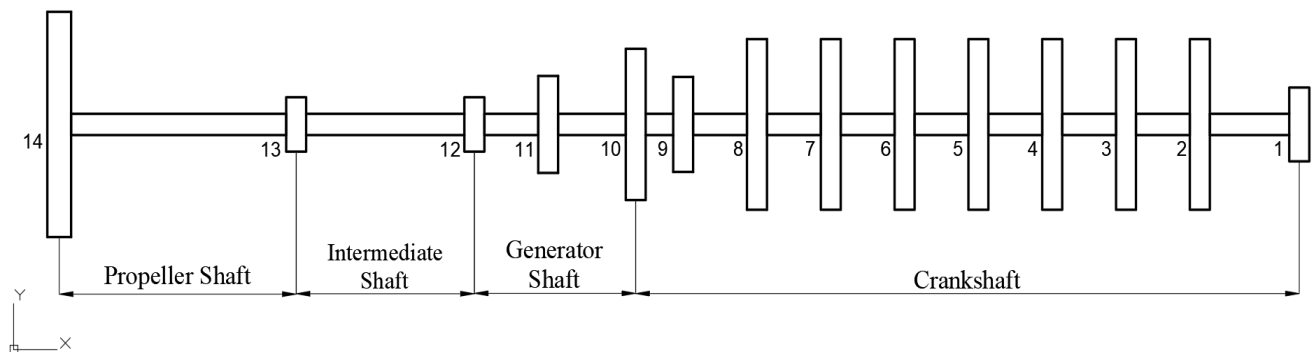


Figure 15: Lumped-Mass Model of the 3000TEU Container Vessel

This model of the power transmission system is now a system of finite lumps or masses deduced from a system of continuous arrangement of rotating parts (Drazen Polic, 2013). This developed mathematical model will be used for the calculation of natural frequencies of the propulsion machinery. If the lumped inertia-stiffness model truly represents the propulsion train of the vessel, then the calculating torsional natural frequencies of the system will match with the original system's frequencies.

4.4.2 Material Data Calculation

In order to determine the torsional natural frequencies, the material data of the dynamic system subjected to the rotational moment should be considered. To proceed to the frequency calculation, two important parameters called as mass moment of inertia and torsional stiffness of the simplified lumped mass model are required. As the whole system is a rotating machinery consists of mass and a defined length, there will be moment of inertia and stiffness properties exhibiting on this system. Here the structure is modelled with 14 lumped masses (rotating components) and each of them connected together with the torsional spring elements (shaft components) without any mass and stiffness loss of the whole system.

As we are dealing with the torsional vibration, the effect or influence of bearings are negligible. Normally, the bearings have a huge role in the vibration analysis of the rotating machinery but that is limited to only lateral and bending mode analysis in shafting. In torsional vibration, the shaft will be forced to twist by the external torque from either one end of the shaft (other end will be assumed as fixed) or by two moments from both ends of the shaft causing vibration nodes. Hence the modelling of the original system is done by excluding the two shaft bearings of the propulsion shafting. Finally, the representation of the lumped-mass model with the corresponding mass moment of inertias (J) for masses and torsional stiffness (K) for shaft segments are depicted in Figure 16.

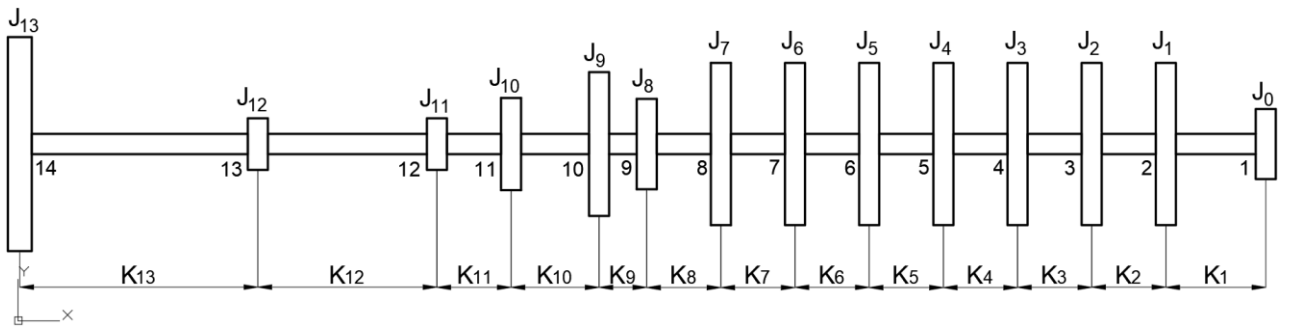


Figure 16: Shaft Model with Moment of Inertias and Torsional Stiffness

After the reduced model is finished, the material data can be calculated directly by using general formulas of mass moment of inertia and torsional stiffness. Assuming the system is subjected to torsion, there will be the formation of shear stresses inside the material and this stress increases linearly from the centre to the radial direction. Due to torsion, shear strain can also be present and both shear stress and shear strain are correlated and directly proportional. Hence the required

material data can be calculated by using the formulas of shafting system subjected to torsion. The torsional stiffness of a shaft can be determined by a well-known equation:

$$K = G \times \frac{j}{L} \quad (15)$$

$$j = \frac{\pi}{32} \times D^4 \quad (16)$$

Where, G is the shear modulus of the material, j is the second torsional moment of area or polar moment of inertia of shaft and L is the length of the shaft and D is the shaft diameter.

The mass moment of inertia (J) should be calculated for each and every rotating solid part in the mass-elastics system developed. Normally the moment of inertia is a parameter that determines the moment required for a desired angular acceleration with respect to the axis of rotation. As the model is a dynamic system, it will be dealing with mass moment of inertia and not with the rotational inertia which is connected to the rigid body concept. Here the moment of inertia of a body about the rotational axis passing through its centre and perpendicular to its face is calculated by assuming each part of the reduced lumped-mass model as a solid disc with a constant thickness t , radius r and density ρ . The general formula to calculate the mass moment of inertia (J) is given in the following equation.

$$J = \frac{1}{2} \times m \times R^2 \quad (17)$$

$$m = \pi R^2 \rho t \quad (18)$$

Where, m is the mass of the part or component and R is the radius of that corresponding part. Hence Equations 15 and 17 can be used to determine the required material data for all the lumped masses and torsional springs in the reduced shaft model. The mass moment of inertia will vary according to the shape of the masses and hence a general idea about the different moments of inertia for different bodies are given in Figure 17. In that figure, the term R denotes the radius of each part and M represents the mass of each body respectively.

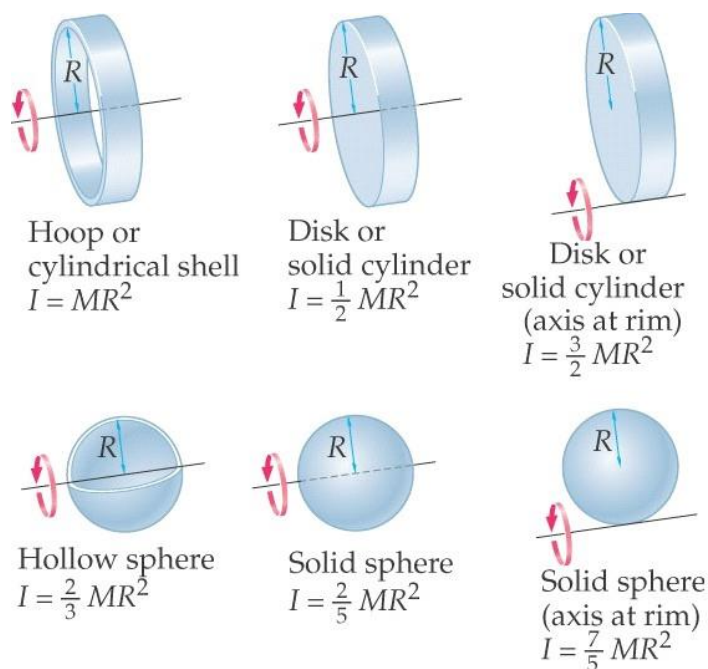


Figure 17: Moment of Inertia for Different Mass Arrangements
(<https://slideplayer.com/slide/6606268/>)

However, the above-mentioned formulas can be directly used to calculate the moment of inertias of the rotor geometries in the shafting. Usually for TVC, as a prerequisite this material data is being provided by the engine manufacturers. Hence in the scope of this thesis work, the data regarding the inertia moments of the diesel engine (inertias of each cylinder, moment compensator and flywheel) have been given by the engine supplier. Along with that, the material data for all the remaining components in the shafting arrangement are already given as the input data prior to the TVC analysis. The task is to do the TVC analysis for the following case:

- Normal torsional vibration calculation of the main engine propulsion plant of the 3000TEU Container vessel with the new propeller.

In both cases (system with old and new propellers), only the propeller properties are changing (mainly the moment of inertia) and all other component parameters are unchanged, but this one change can cause significant impact in the whole propulsion shafting system on basis of vibration. The new re-designed propeller has a smaller mass compared to the previous one and as a result the moment of inertia of the marine propeller decreases. A detailed description about the position of each component in the shafting line and the corresponding mass inertias and torsional stiffnesses for the given scenario are given in Table 3.

Table 3: Main Particulars of Shaft Line for the TVC Analysis with Re-Design Propeller

System No.	Components	Mass moment of Inertia, J (Kgm ²)	Torsional Stiffness, K (MNm/rad)
1	Moment Compensator	5995	3759
2	Cylinder 1	30557	2865
3	Cylinder 2	31231	2865
4	Cylinder 3	30557	2865
5	Cylinder 4	30666	2865
6	Cylinder 5	31231	2865
7	Cylinder 6	30557	2865
8	Cylinder 7	31231	3477
9	Thrust Bearing	13180	4525
10	Flywheel	15889.10	329
11	Generator	3589.80	1155.90
12	Flange 1	515	107.20
13	Flange 2	1262.80	151.40
14	Propeller	121508	-

4.4.3 Approximation Methods for Computing Natural Frequency and Mode Shapes

The natural frequencies or eigen frequencies of the torsional shaft line vibration can be determined analytically by different calculation methods that are in practise. The methods that are commonly used for the mechanical vibration analysis are approximation methods, since as the number of degrees of freedom increases in a dynamic system, the solution to get converged becomes difficult. The different methods that have been widely used for finding the natural frequencies and mode shapes of any vibrating system are as follows:

1. Holzer's Method:

This method can be easily implemented to the continuous as well as discrete parameter vibration model systems. Hence the natural frequency and mode shapes can be extracted from a multi mass lumped elastic system in an iterative method as developed by Holzer. It is suitable for straight and branched type of vibration problems. It's application also extend to free, forced, damped, undamped and semi-definite systems too (Vishwajeet Kushwaha, 2012).

2. Rayleigh's Method:

This method is based on energy theory where the vibration frequency has a static value nearby to the natural mode. It will give an upper bound approximation to the basic frequency of the model.

3. Transfer Matrix Method:

This method is an extend of the Holzer's method where the concept of state vector and matrices of transfer are used to find the natural frequencies for distributed parameter system and also for lumped systems.

4. Dunkerley's Method:

It is used generally for the transverse vibration scenarios. The method is based on matrix theory. It can be implemented effectively in systems with low damping and having higher frequencies than the fundamental frequencies. This method is commonly used for solving fundamental frequency of the vibration model. The method gives a lower-level approximation as that the estimated frequency value is always lower than exact frequency, since the harmonics are neglected in the calculation (Yuwen Yao, 2004).

On basis of the extend of applications in multi-branch systems, these methods have certain limitations. But for an implicit analysis in the TVC of in-line and as well as branched systems Holzer's method and Transfer Matrix method are the basic and accurate ones. The method chosen for the Torsional Vibration Calculation (TVC) in the framework of this thesis is Holzer's Method.

4.4.4 Basic Principle of Holzer's Method

Holzer's method is an effective method based on iteration which can be used find the natural frequencies and eigen modes of any dynamic vibrating system with multiple Degrees of Freedom (DOF). This method is applicable to systems with rectilinear and angular motions, semi-defined, branched and straight systems with fixed ends, damped or undamped models and so on. The theory of Holzer's method is connected to the lumped-elastic model, where the shaft segments are taken as massless elastic springs with a torsional stiffness and the disks or solid cylinders are considered as lumped masses. This iteration procedure assumes a trial frequency

at the start and a solution is formed when the assumed frequency matches the constraints of the problem. The iterations or trials will be several, but it is able to determine more than one torsional natural frequencies of the system. The whole method is a tabulation technique where the fundamental as well as mode shapes of the system can be obtained.

The basic and detailed theory of this effective method for a straight system is explained hereafter, since the investigating propulsion system of the 3000TEU Container vessel is a straight-line system (main engine is connected directly to the propeller through shafts). Firstly, the procedure to evaluate and analyse a three-disk torsional vibrating system as in Figure 18 by the Holzer's method is explained.

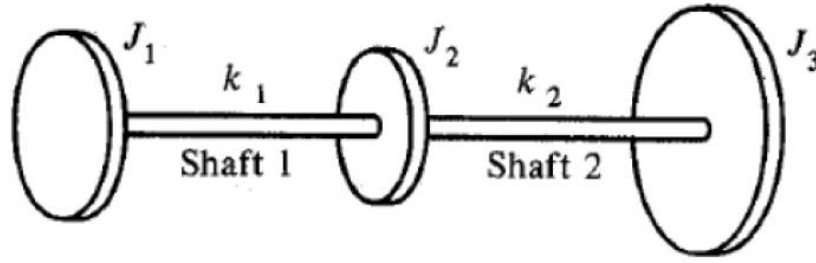


Figure 18: A Three Disk Straight-Line Torsional Vibration System (Vishwajeet Kushwaha, 2011)

Consider a three-rotor torsional straight system and according to Newton's 2nd law, the equations of motion for a torsional system can be written as follows:

$$J\ddot{\theta} + k\theta = 0 \quad (19)$$

Where, J is the mass moment of inertia of the solid disk, k is the torsional stiffness of the shaft, θ is the angular displacement and $\ddot{\theta}$ is the angular acceleration due to the twisting of the system. Now formulating the general equation of motion for a three rotor system will be as follows,

$$J_1\ddot{\theta}_1 = -k_1(\theta_1 - \theta_2) \quad (20)$$

$$J_2\ddot{\theta}_2 = -k_1(\theta_2 - \theta_1) - k_2(\theta_2 - \theta_3) \quad (21)$$

$$J_3\ddot{\theta}_3 = -k_2(\theta_3 - \theta_2) \quad (22)$$

The motions are harmonic at principal mode of vibration or we can say that the nature of vibration will be harmonic at principal mode and this allows us to write the formula for angular

displacement in harmonic nature. The general equation of the angular displacement will be as follows (Yuwen Yao, 2004),

$$\theta_i = \theta_i \sin(\omega t + \psi) \quad (23)$$

Simplifying the equations of motion by substituting the value of $\theta_i = \theta_i \sin(\omega t + \psi_i)$, for the term $i = 1, 2, 3$ in the motion Equations 16-18, factoring out the $\sin(\omega t + \psi)$ term, and rearranging the equations,

$$-\omega^2 J_1 \theta_1 + k_1(\theta_1 - \theta_2) = 0 \quad (24)$$

$$-\omega^2 J_2 \theta_2 + k_1(\theta_2 - \theta_1) + k_2(\theta_2 - \theta_3) = 0 \quad (25)$$

$$-\omega^2 J_3 \theta_3 + k_2(\theta_3 - \theta_2) = 0 \quad (26)$$

Where, ω is the natural frequency of the equivalent mass spring system or torsional rotor system. Hence, the above equations can be generally written as,

$$\sum_{i=1}^3 J_i \theta_i \omega^2 = 0 \quad (27)$$

Likewise, for a n rotor system, the condition to satisfy the frequency estimation is,

$$\sum_{i=1}^n J_i \theta_i \omega^2 = 0 \quad (28)$$

According to Holzer, the above formula must be followed that the total sum of the inertia torque of the vibration system must be zero. Also, the initial or even the trial frequency ω must satisfy this constraint. So, the Equation 28 can be considered as an alternative form of the frequency equation. An idea regarding the mass moment of inertia, torsional stiffness and the angular displacement of a lumped mass-elastic model can be obtained from the Figure 19.

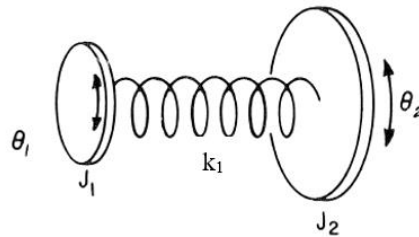


Figure 19: Small Mass-Elastic Model with Two Rotors (Ronald L, 1977)

Generally, the procedure of trials inside the Holzer method begins by assuming an initial value of frequency ω and angular displacement θ . The values of ω for which the Equation 28 is satisfied are the eigen frequencies and the values of θ_i give the mode shapes of the system. To start with the Holzer tabulation, the first step is to assume a trial natural frequency value of ω and a relative amplitude or angular displacement of $\theta_1 = 1$ arbitrarily. Following that, the second angular displacement θ_2 will be calculated using the Equation 24. Then the next relative amplitude θ_3 will be formulated from the next equation in that series (Yuwen Yao, 2004).

$$\theta_1 = 1 \quad (29)$$

$$\theta_2 = \theta_1 - \frac{\omega^2 J_1 \theta_1}{k_1} \quad (30)$$

$$\theta_3 = \theta_2 - \frac{\omega^2 (J_1 \theta_1 + J_2 \theta_2)}{k_2} \quad (31)$$

Now by substituting the corresponding values of θ_1 , θ_2 and θ_3 into the Equation 28 to check if the condition is satisfied. If the constraint is satisfied, that is if the summation of the inertial torque of each component of the vibration system becomes zero, then the assumed frequency will be the Torsional Natural Frequency (TNF). If the condition is not satisfied, the whole iteration procedure is repeated with a new assumption of frequency ω until the condition of $\sum(\text{Inertial Torque}) = 0$ is met. From the Equations 29-31, we can form a general expression suitable for angular deformations for a n rotor system can be formed as,

$$\theta_j = \theta_{j-1} - \left(\frac{\omega^2 \sum_{i=1}^{j-1} J_i \theta_i}{k_{(j-1)}} \right) \quad j = 1, 2, \dots, n \quad (32)$$

In summary, this iterative method consists of the repeated execution of the Eqs. 28 and 32 for different trial frequency values. If the assumed frequency is not a natural frequency of the model, then automatically the Equation 28 will not be satisfied and the left hand side of that equation is called the residual or inertia torque and can be expressed as,

$$T_i = \sum_{i=1}^n J_i \theta_i \omega^2 \quad (33)$$

The inertial torque is the torque being applied at the last disk in the torsional system. It is similar to a condition of steady state forced vibration (Yuwen Yao, 2004). The resulting residual torque T_i versus the assumed frequency ω for a typical 3 rotor system is illustrated in Figure 20.

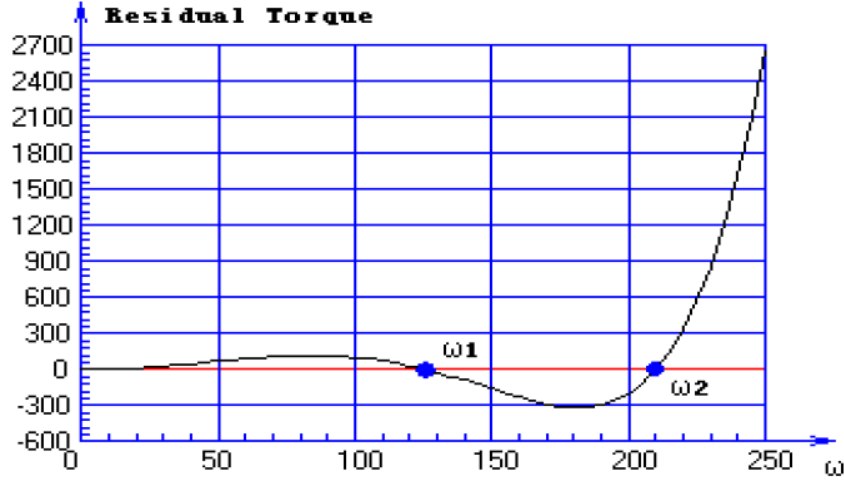


Figure 20: Residual Torque Versus Frequency Curve (Yuwen Yao, 2004)

Here, the natural frequencies of the torsional systems are the frequency values that are close to zero (ω_1 and ω_2), or we can say that when the total residual torque in the system changes its sign from positive to negative or vice versa at a corresponding frequency, that frequency value will be the natural frequency of the system.

4.4.5 Natural Frequency Calculation Using Holzer's Method

By applying the Holzer's method into the actual vibration system of the Container vessel, the natural frequencies can be obtained. The main data required prior to the frequency calculation is the mass moment of inertia and torsional stiffness values of each rotor and shaft segments in the equivalent lumped-mass model of the system and those values are already known.

Hence after following the iteration procedure of Holzer's method, the natural frequencies of the shafting system of the vessel are found out. where a three-rotor system is considered. Here the shafting system consists of 14 rotors with inter connected shafts. After performing the iteration procedure for different frequency values of ω , torsional natural frequencies are obtained as per the residual torque constraint; i.e., a natural frequency occurs when there is a change in sign in the sum of inertial torque. The calculated natural frequencies of the shafting vibration system is tabulated in Table 4.

Table 4: Torsional Natural Frequency of the Shafting System of the Vessel with New Propeller

Torsional Natural Frequency, ω				
Modes	1 st Mode	2 nd Mode	3 rd Mode	4 th Mode
ω (rad/sec)	24.19	117.02	223.85	292.18
ω (Hz)	3.85	18.624	35.627	46.50

For instance, to verify the residual torque condition, a tabular column is developed showing the relative amplitude or angular displacement for each part and also the occurring inertial torque corresponding to the first torsional natural frequency value of 24.19 rad/sec is given in Table 5. The formulas used to calculate the angular displacement θ and residual torque T_i in this analysis are already given in Section 4.4.4.

Table 5: Holzer Tabulation for the First Torsional Natural frequency

System No.	Components	Mass Moment of Inertia, J (Kgm ²)	Torsional Stiffness, k (Nm)	Angular Displacement, θ (rad)	Residual Torque, T_i (Nm)
1	Moment Compensator	5995	3759000000	1	3508010,82
2	Cylinder 1	30557	2865000000	0,99906677	21371939,04
3	Cylinder 2	31231	2865000000	0,991607106	39493568,97
4	Cylinder 3	30557	2865000000	0,977822265	56977632,38
5	Cylinder 4	30666	2865000000	0,957934784	74167194,41
6	Cylinder 5	31231	2865000000	0,932047456	91200371,14
7	Cylinder 6	30557	2865000000	0,900214866	107296766,5
8	Cylinder 7	31231	3477000000	0,862763987	123063787,2
9	Thrust Bearing	13180	4525000000	0,827370318	129444762,8
10	Flywheel	15889.10	329000000	0,798763741	136871351,5
11	Generator	3589.80	1155900000	0,382741396	137675335,6
12	Flange 1	515	107200000	0,263634782	137754783,3
13	Flange 2	1262.80	151400000	-1,021391182	137000041,5
14	Propeller	121508	-	-1,926279171	-39382,22881

From this table, it is evident that at system no. 14 (Propeller), the restoring torque has become negative or there is a sign change compared to the previous value. That indicates that the

frequency at which this torsional moment is excited is a natural frequency. Similarly, it can be seen for all the calculated natural frequencies. A clear understanding of the sign change in the summation of inertial torque and the corresponding occurrence of the natural frequencies are visible from the frequency curve. Natural frequencies are calculated by plotting these curves of Residual torque (Nm) vs Frequency (rad/sec). The graph plotted below depicts the occurrence of more than one torsional natural frequencies of the vibrating shafting system of the vessel. Among those only the first four modes of natural frequencies are considered for the TVC analysis since they are the most critical ones in analysing torsional vibratory stresses in general cases.

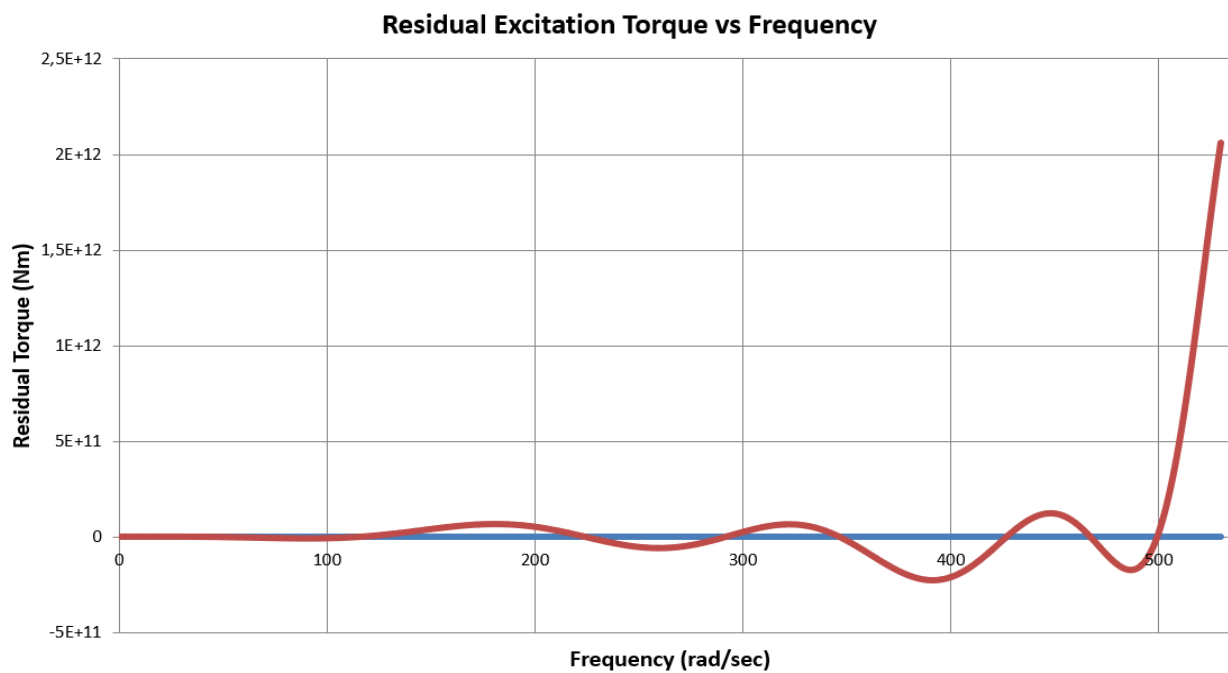


Figure 21: Residual Torque vs Frequency Curve

The relative amplitude of the rotors in the vibrating system gives the vibration pattern or mode shapes to a particular frequency. Hence the mode shapes corresponding to the first three orders of torsional natural frequency are represented by Figure 22.

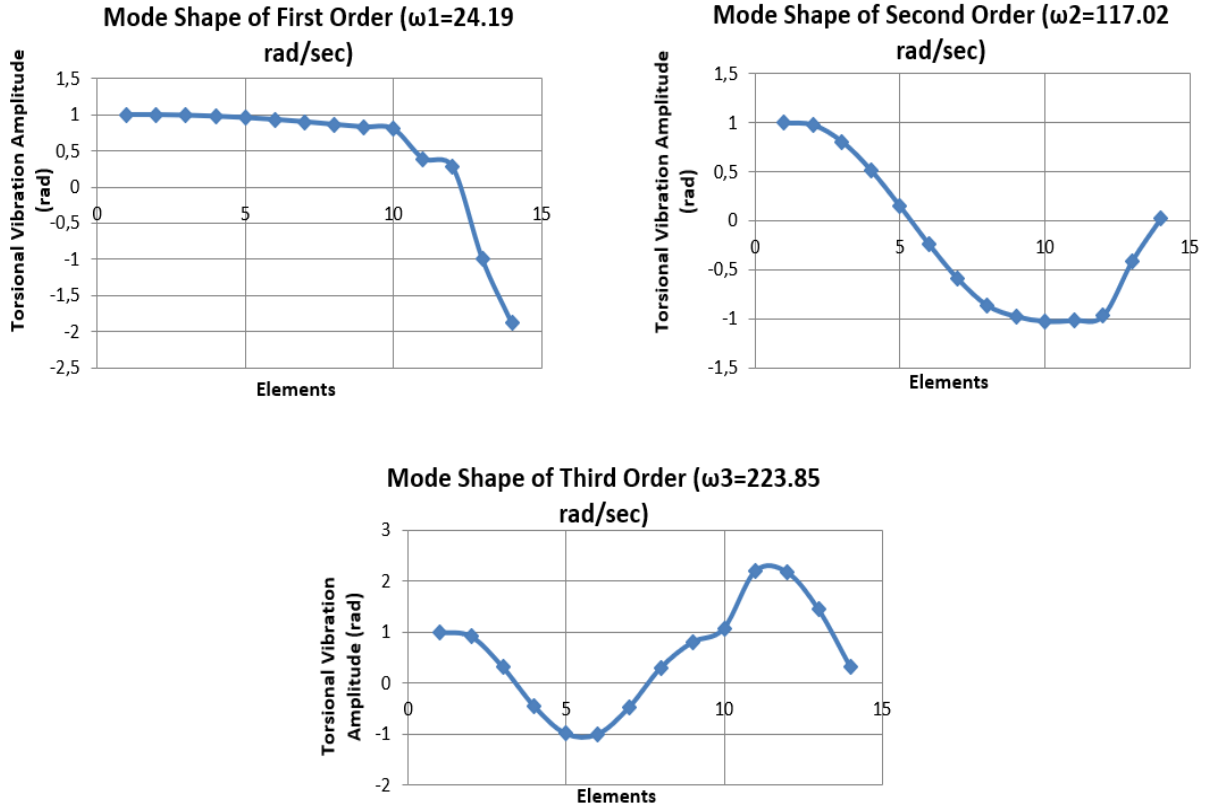


Figure 22: Mode Shapes of First Three Torsional Natural Frequencies

4.5 Forced Dampened Torsional Vibration Analysis

4.5.1 Forced Torsional Vibration Calculation

The forced Torsional Vibration Calculation (TVC) of the propulsion system is usually performed in a frequency domain approach for without ice conditions and it is due to the fact that the shaft rotation speed is assumed to be constant. Since there is no external ice torque, it is considered that the mean torque developed by the prime mover is in equilibrium with the hydrodynamic torque given to the propeller side (Yuriy A. Batrak, 2012). The determination of forced damped oscillations of multi-mass systems had been achieved through the Holzer method in the earlier section. Following to that a calculation scheme known as Benz scheme can be used for systems with any number of masses, damping and excitations.

Basically, the procedure is carried out according to Holzer at the beginning, then the influence of angular oscillations of each mass are taken into account. Hence the main advantage of this methodology is that all known parameters like torsional angles and torques which are significant for torsional vibrations are considered in this scheme. The scheme is divided into

two parts where at first the free vibration of the system is calculated without considering the influence of the excitation forces from the engine cylinders. In the second part, forced vibration is performed by taking into account the cylinder forces (gas harmonics data). In both cases, vibrations are calculated for damped systems with proper damping coefficients on each component.

For this analysis, the external moment at each rotor is calculated on basis of the angular deflections caused at those rotors. It is considered that when one of the components with a mass θ_1 oscillates at a defined angle φ_1 , this rotating mass brings the moment $\varphi_1 \theta_1 \alpha^2$ and also causes the twist $\Delta\varphi_{12} = -\varphi_1 \theta_1 \alpha^2 l_{12}$, where α will be the assumed natural frequency of the complete shafting system. For schematic calculation, it is considered that the estimated moment and twist are decomposed into two mutually perpendicular components, horizontal sin-members and vertical cos-members. So, in the 1st part of schematic calculation, the sum of all the sin and cos members of moment and twist for all the components are calculated directly, and in the 2nd part, the same step is repeated but an external moment from the engine (pressure forces given as gas harmonics data from the engine manufacturer) is added to the moment and twist. Finally, the resulting angular displacement and amplitude of the residual torque will be determined. This torque is calculated at each component's position of the system and from that the shear stress or torsional stress can be easily determined if the diameter of that particular component is known.

System No.	Components	Mass Moment of Inertia, J	Torsional Stiffness, k	Absolute Damping	Relative Damping	Crank Angle	Shaft Diameter		Gear Ratio
		(kgm ²)	(MNm)	Damping Coeff. V or Archer Factor	Non Dimensional Damping Factor, tan ε		Outer	Inner	
						(Deg)	(mm)	(mm)	(-)
1	MOMENT COMPENSATOR	5995,00	3759,0		0,010		896	150	1
2	CYLINDER 1	30557,00	2865,0	58,8	0,010	0,00	896	247	1
3	CYLINDER 2	31231,00	2865,0	58,8	0,010	102,86	896	150	1
4	CYLINDER 3	30557,00	2865,0	58,8	0,010	257,14	896	247	1
5	CYLINDER 4	30666,00	2865,0	58,8	0,010	205,71	896	226	1
6	CYLINDER 5	31231,00	2865,0	58,8	0,010	154,28	896	150	1
7	CYLINDER 6	30557,00	2865,0	58,8	0,010	308,57	896	247	1
8	CYLINDER 7	31231,00	3477,0	58,8	0,010	51,43	896	150	1
9	THRUST BEARING	13180,00	4525,0		0,010		896	150	
10	FLYWHEEL	15889,10	329,0	100,0	0,010		570		
11	GENERATOR	3589,80	1155,9		0,010		570		
12	SHAFT FLANGE 1	515,00	107,2				570		
13	SHAFT FLANGE 2	1262,80	151,4				715		
14	PROPELLER	121508,00		9,1					

Figure 23: Input Data for Torsional Stress Calculation for the Vessel with New Propeller

In the input data, all the material characteristics like mass moment of inertia, torsional stiffness, crank angle for each cylinder are provided by the engine manufacturer and the external design office (who conducted TVC analysis for the propulsion unit with old propeller). The damping coefficients (absolute and relative) are taken based on the values that are generally used for the TVC of cylinders, bearings, flywheel and the propeller. A detailed view of the scheme used for the forced vibration calculation is added in Appendix A1 section.

4.6 Results for the TVC Analysis in Open water Conditions

CASE: Marine Propulsion System of the Container vessel with New Propeller

The excited torque in the shafts will consist of a number of single harmonic excitations, and each of them has its own frequency (firing order frequency), which is a multiple of the rotating shaft frequency. This multiple is termed as the vibration order. So, there exists first order excitation, the second order excitation and so on. It happens like an i^{th} order excitation produces the i^{th} order response (Gojko Magazinović, 2002). The final shaft torsional response graphs are made from the synthesis of a number of single harmonic responses. All the limit curve calculation of different shafts are added in Appendix A2 section.

4.6.1 Case 1: Vibratory Stress in the Crankshaft

The torsional stresses in the crankshaft are calculated as a function of the engine speed. As per Class rules, the allowable stress limit for continuous operation for two stroke engines are between 25 and 40 N/mm². Hence a value of 27 N/mm² is provided by the engine manufacturer.

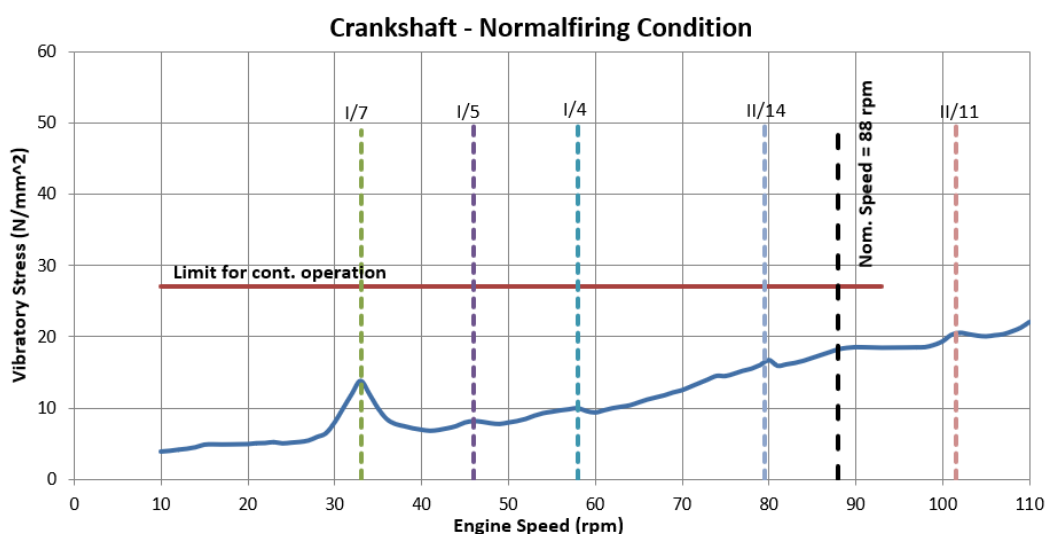


Figure 24: Synthesis Graph of Vibratory Stresses in the Crankshaft for Normal firing

The dotted vertical lines in the graphs represents the speed at which torsional natural frequency mode occurs for a particular firing order (the stresses are observed to be higher at these regions). The horizontal thick line shows the maximum limit the stress can rise. For instance in Fig 24, the green dotted line shows the occurrence of first torsional natural frequency of the shafting (24.19 rad/sec) for the firing order 7 at a speed of 33 rpm. Similarly, the purple dotted line denotes the speed (102 rpm) at which the second order natural frequency (117.02 rad/sec) occurs for a firing order 11. Throughout the engine speed range, the stresses are well below the limit, but some larger peak stress values are visible in the speed range between 30 to 36 rpm.

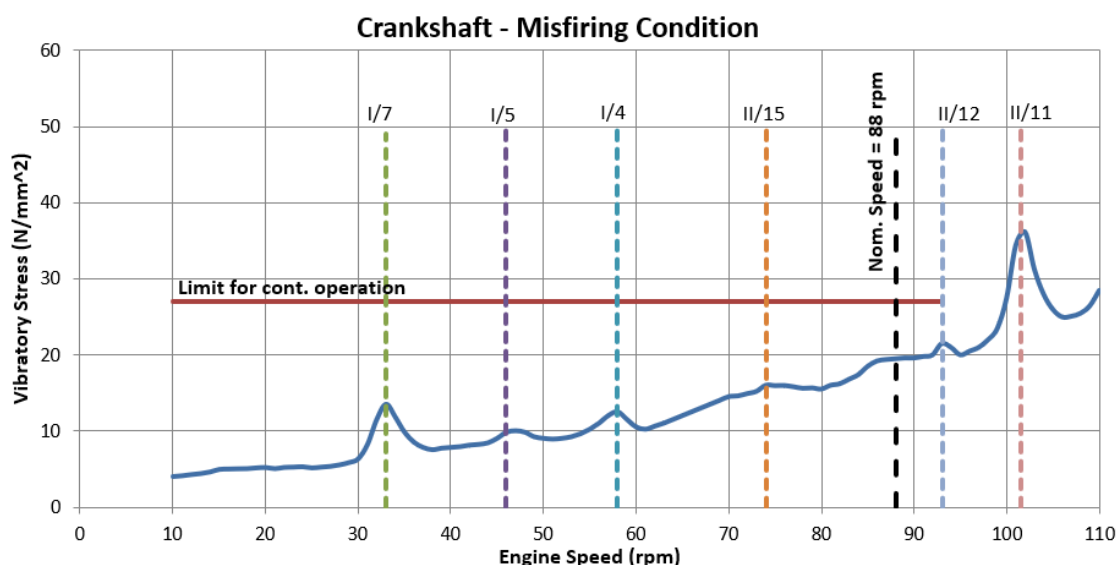


Figure 25: Synthesis Graph of Vibratory Stresses in the Crankshaft for Mis-firing

This representation of dotted lines showing the correspondence of natural frequency and firing order follows for all the remaining graphs in the coming sections.

4.6.2 Case 2: Vibratory Stress in the Intermediate Shaft

Here, the torsional stresses in the intermediate shaft are calculated. The limit curves are plotted for the shafts that are made of forged, low alloy carbon steel, which are often used for intermediate shafts in Figures 26-27. After observing the graphs, it is clear that the stresses are under the limit for the normal firing condition whereas in mis-firing situation, small peaks are appearing at speeds 33, 46, 57 and 77 rpms. Also, all of these are occurring at the first order natural frequency of the shafting. In misfiring condition, the dangerous ones are the stresses forming at 33 rpm and 77 rpm. Hence it is advisable to operate the main engine well below the threshold value of 77 rpm.

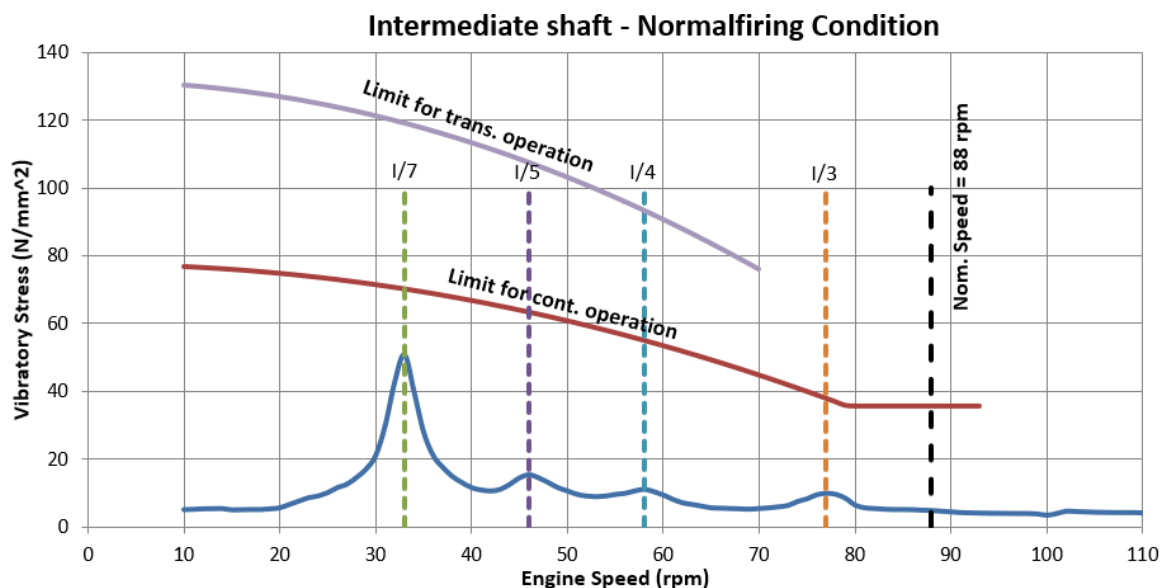


Figure 26: Synthesis Graph of Vibratory Stresses in the Intermediate Shaft for Normal firing

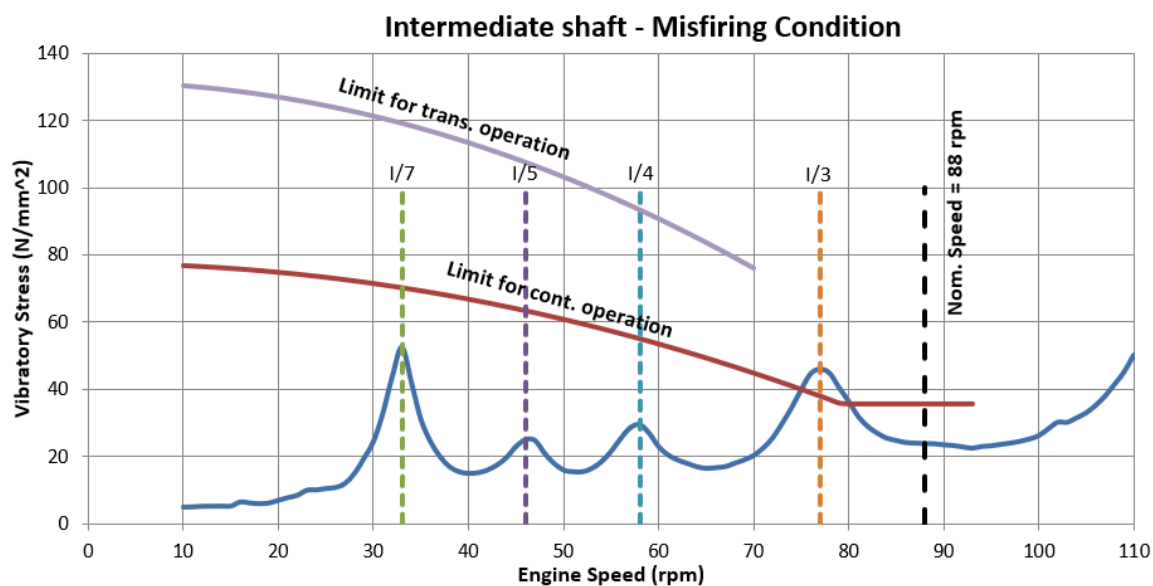


Figure 27: Synthesis Graph of Vibratory Stresses in the Intermediate Shaft for Mis-firing

4.6.3 Case 3: Vibratory Stress in the Propeller Shaft

In the propeller shaft, the trend of the curves looks similar to the intermediate shaft, but the stress limit values are different. Engine speed ranges below 25 rpm and over 35 rpm are characterized by moderate, even low stress values can be seen. Above 90 rpm, the vibratory stress is exceptionally low, i.e., in the vicinity of the nominal engine speed. Here also, the peak vibration stress is reached at 33 rpm, like the remaining shaft segments. This is due to the

resonance between the excitation torque and the system's natural frequency. So, this engine speed is usually called as the critical speed.

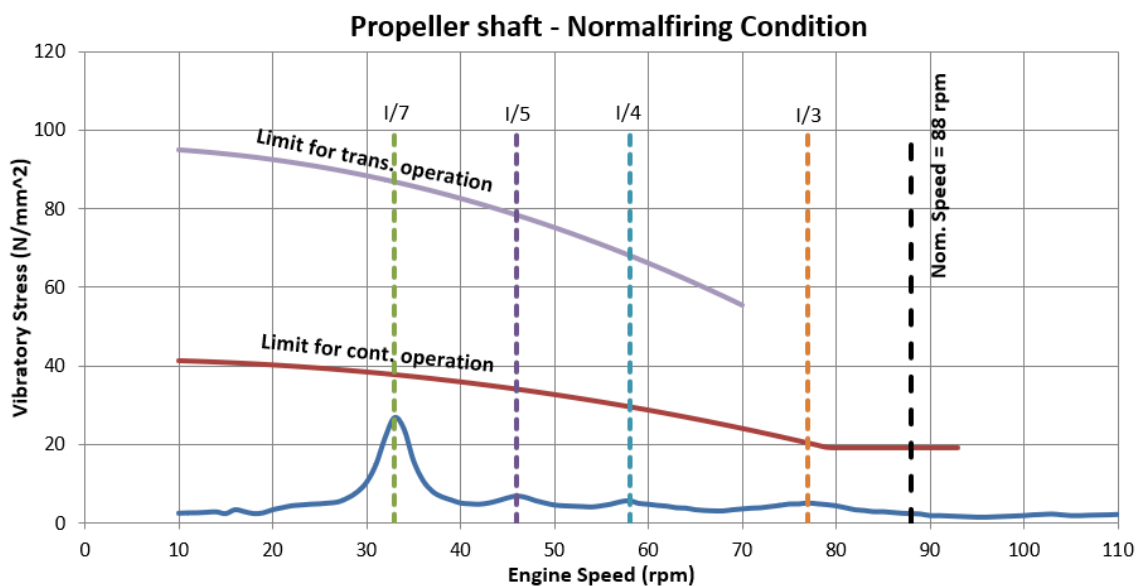


Figure 28: Synthesis Graph of Vibratory Stresses in the Propeller Shaft for Normal firing

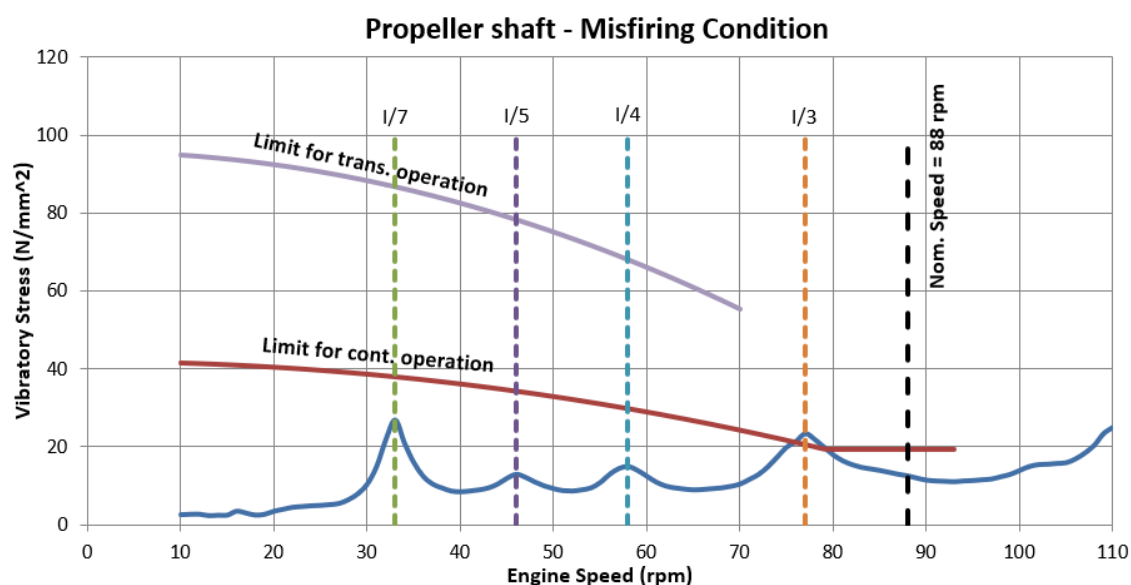


Figure 29: Synthesis Graph of Vibratory Stresses in the Propeller Shaft for Mis-firing

Observations:

- The excitation torque is composed of a number of single harmonic excitations which has its own system response (frequency). The various peaks represent this resonances.
- The critical speed range (30-36 rpm) represents the barred speed range of the vessel that must be covered rapidly during voyage.
- Torsional stresses in the shafting requires some time to develop fully, and if the barred speed range is passed soon, gives a great chance that high stress level would never reach.

4.7 Development of FEM Procedure

Finite Element Method (FEM) is used to analyse the vibration behaviour of the multi-body shafting system consisting of several flexible or rigid bodies in a more accurate and practical way. In order to validate and analyse the analytical results, a numerical analysis of the marine propulsion shafting system was performed using the ANSYS software to conduct and simulate the vibration analysis. To begin with the shafting torsional vibration analysis, a simulation model of the 3000TEU Container vessel propulsion system has been built, including flywheel, shafting, flanges, intermediate bearings and propeller hub. The detailed parameters and material properties of each components have to be set based on the real shafting. The free vibration is being done with the Modal Analysis where the natural frequency and mode shapes of the shafting system can be obtained during its own self vibrating nature. Later, a Harmonic Response Analysis is performed to evaluate the effect of the external moment causing the rise of stresses in the system and to identify the critical frequency range. The application of the FEM vibration analysis will be similar for any propulsion shafting inside a vessel; since the vibration analysis (free and forced analysis) is mainly depending on the material characteristics.

4.7.1 3-D Model of Marine Propulsion Shafting

In the analytical part, the original shafting arrangement of the vessel is modified into a lumped-mass elastic system, where the mass is concentrated in the rotors and the shaft segments denotes an elastic system. Coming to the numerical analysis, the process of modelling of shafting is slightly different. The complete shafting system is modelled with its real geometric attributes and appropriate material, which is then discretised into finite elements (Husak, E., and Haskić, E. (2018). To improve the efficiency of simulation, some complicated structural parts having small influence on the results are simplified. It includes the engine block section where all the cylinders are modelled to be a solid shaft with an appropriate diameter of an equivalent crankshaft. So, the equivalent part of crankshaft in result gives the total added effect of mass properties of the cylinders. All the components in the shafting line are modelled as solid disks with exact dimensions in order to reduce the cost of complexity and also computational time. Two bearings (plummer and tunnel bearings) are placed at their exact positions to provide the boundary conditions to fix the position of shafts in the propulsion shafting. Both the bearings

are coming at the intermediate positions of the whole system; hence they are locating type that are normally used at the drive end parts of the shafting.

In the case of open water conditions, the influence of propeller blades in the FEM analysis of vibration behaviour of shafting is very small. So, only a propeller hub is modelled based on the original material characteristics and dimension parameters at the end part of the propeller shaft. All the part are modelled based on the original dimensional properties of the vessel propulsion system in the 3000TEU Container. The modelling of the system is done in Siemens NX software where the complete arrangement is modelled as solid elements for the result accuracy at the simulation stages. The simplified model of the marine propulsion system of the 3000TEU Container is shown in Figure 30.

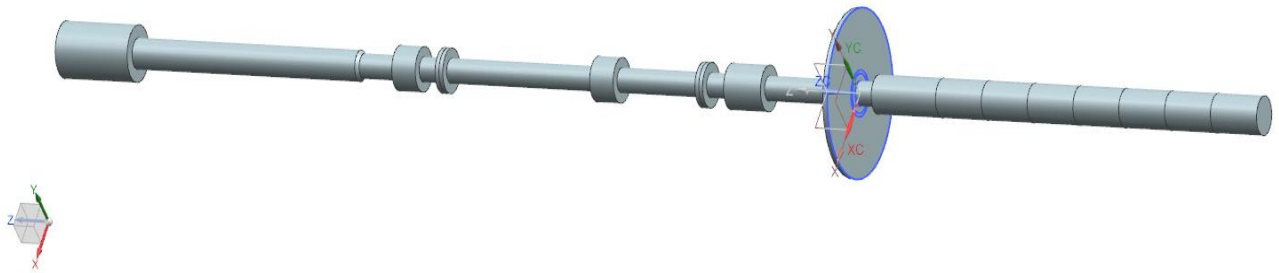


Figure 30: Model of Ship Propulsion Shafting of the 3000TEU Container Vessel

The main parameters of the vessel shafting used to model the system are given in Table 6.

Table 6: Main Parameters of Ship Propulsion Shafting

Components	Length (mm)	Diameter (mm)
Equivalent Part of Crankshaft	11978	896
Flywheel	140	3842
Generator Shaft	4360	570
Generator	1150	1055
Flange Coupling 1	240	1030
Tunnel Bearing	730	1030
Intermediate Shaft	10418	570
Flange Coupling 2	240	1030
Plummer Bearing	730	1030
Propeller Shaft	9157	715
Propeller Hub	2237	1300

4.7.2 Modal Analysis of the System

The modal analysis is done to check the natural frequencies of the physical system and also to know about the corresponding mode shapes at those particular frequencies. The modal analysis is also called as free vibration analysis and this procedure is almost similar to that of a linear static analysis. The main steps involved in this analysis procedure is given below:

- Geometry Modelling
- Assign Material Properties
- Define Contact Regions (applicable for flexible and rigid part contacts in some case)
- Define Mesh Controls and Settings
- Provide Boundary Conditions or Supports
- Request Frequency Results (maximum number of modes, frequency range, etc.)
- Solve the Model

In the geometry modelling stage, the complete system can be modelled as either solid bodies, surface bodies or even line bodies. The surface modelling should only be done with appropriate thickness defined and the line bodies with appropriate cross-sections defined. These two models require less computational time in overall but solid model gives a more accurate result but with the cost of computational time. While modelling the geometry, point mass feature is used in the equivalent part of crankshaft and also at the region of the propeller hub. Point mass basically add mass to a certain body or selected surface so the effect is to add only mass (not stiffness) to a structure. This feature is used to ensure the total mass of these components are made to be similar to the mass of the original shafting system, as the natural frequency estimation depends on the mass properties of a system. A total of 7-point masses representing the exact mass and mass moment of inertia of each engine cylinders are placed at equal distances from the flywheel end. At the propeller side, to enable the effect of a complete propeller, another point mass with the propeller mass is given to the hub surface.

Regarding the material properties, the material chosen for the shaft material and also for the remaining components in the shafting is S355. This category of steel is a medium tensile strength low carbon steel which is easy to weld and also possess good impact resistance. This material is widely used and also a preferable material in the construction of shafts. The properties of the S355 material are as follows:

Table 7: Material Properties of S355 Material

Sl.No.	Properties	Value	Units
1.	Density	7850	Kg/m ³
2.	Young's Modulus	210000	N/mm ²
3.	Poisson's Ratio	0.3	-
4.	Tensile Yield Stress	355	N/mm ²
5.	Tensile Ultimate Strength	470	N/mm ²

While defining the contact regions, the contact behaviour will be different for the non-linear contact types in a purely linear analysis. Here Bonded contact type is used between all the rotating components that are in relative motion to each other. The Bonded and No Separation contact types normally depend on the pinball region size and this region is automatically determined by the software.

The boundary conditions or support are a significant part in vibration analysis, as they can affect the mode shapes and natural frequencies of the system. The boundary conditions are applied at the locations of bearings in the shaft line. For both bearings, the boundary condition - Cylindrical Support is used, as this kind of support is suitable for rotating parts that move relative to its counter parts (Benjamin Grunwald, 2018). Their motion can be translation as well as rotational to the adjacent parts. Here, both the bearings in the shafting system are drive end bearings and so they will come under locating or fixed bearings, for which the radial and axial DOF's are constrained but the tangential DOF is free. The boundary conditions are applied in a way by which the simulation gives a nature of the actual realistic shafting running status.

The meshing also has an important role in the vibration analysis. The model was made to undergo a mesh study with different mesh sizes (200mm and 400mm) and it is noticed that the change in mesh size doesn't affect considerably the convergence of results. So, a reasonable meshing size of 250mm is chosen to reduce the calculation time and improve the result accuracy. The meshing method chosen for the model is Hex dominant method where Hex elements are preferably used for meshing parts, but other elements will be used by default to fill in the missing geometry gaps, edges and so on. The simulation of the ship propulsion system is being performed with the help of the ANSYS Mechanical to simulate the actual working conditions of a shaft and the members attached to it. Figure 31 shows the final geometry with all the boundary conditions and point masses used for the modal analysis.

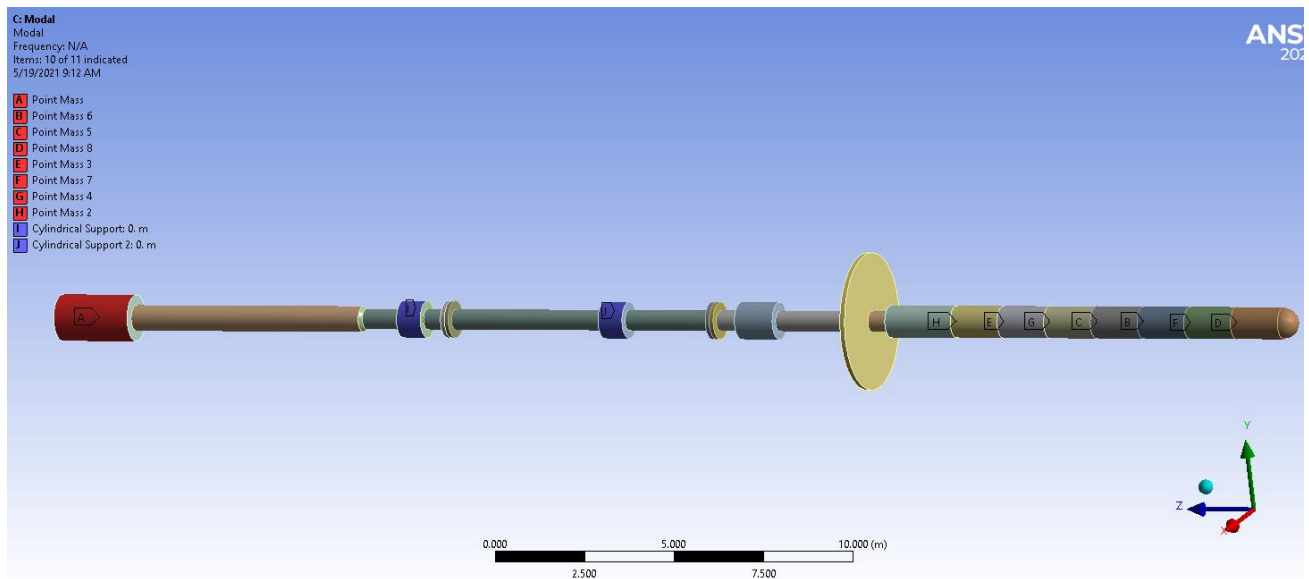


Figure 31: Geometry of Ship Propulsion Shafting with Supports and Point Masses

4.7.3 Natural Frequency and Mode Shapes

The modes of frequencies of the shafting system have been obtained from the Modal analysis. The first two modes of torsional natural frequency and the resulting mode shapes are illustrated in Figures 32-33.

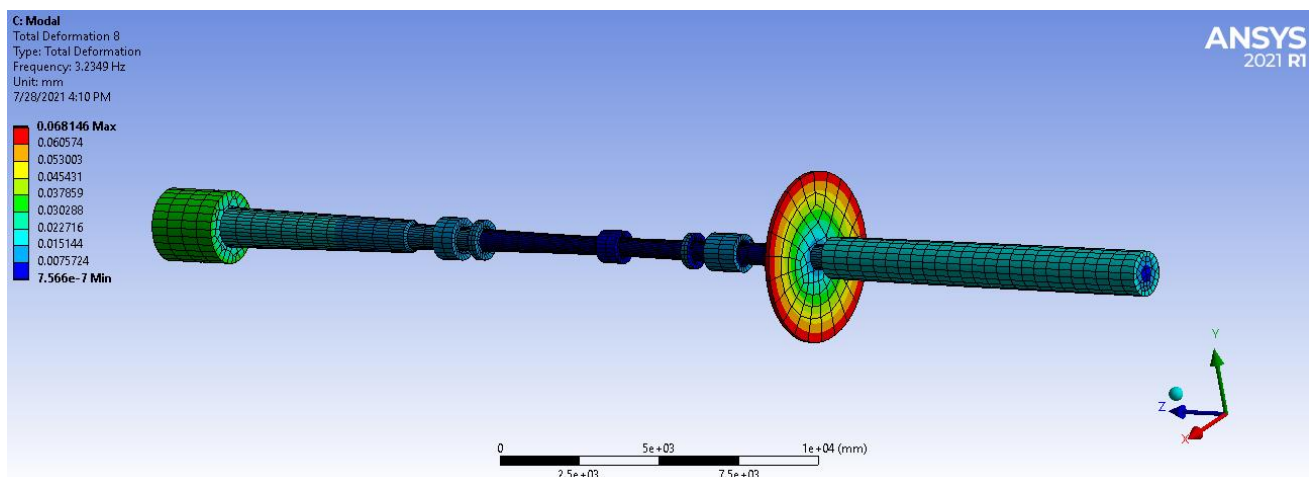


Figure 32: Mode 1 Torsional Natural Frequency of Shafting Design

The first value of the torsional natural frequency obtained is 3.23 Hz which corresponds to 20.29 rad/sec and the first mode shape for this frequency can be seen in Figure 32.

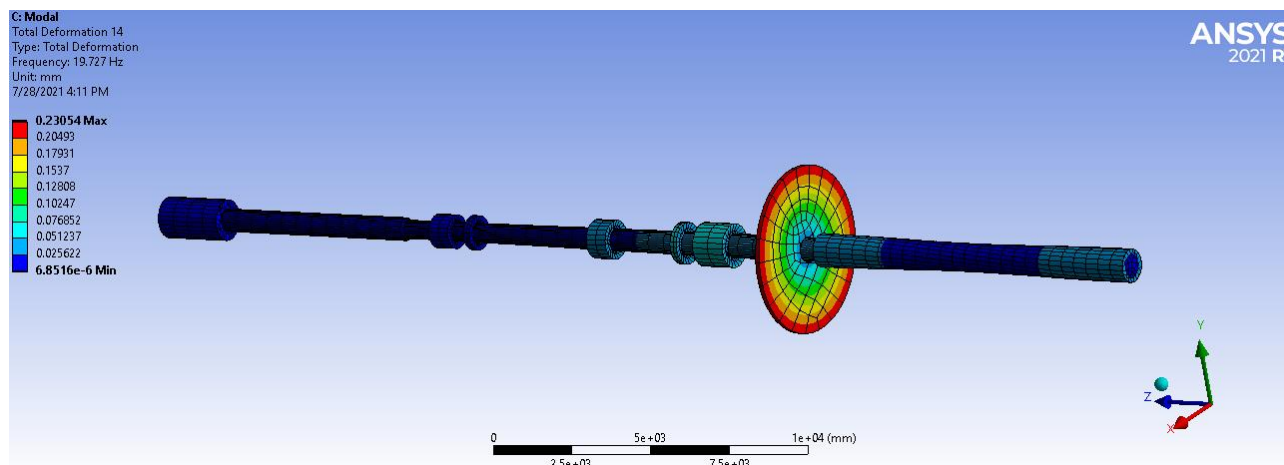


Figure 33: Mode 2 Torsional Natural Frequency of Shafting Design

The second torsional natural frequency has a value of 19.72 Hz and that corresponds to 123.90 rad/sec. The mode shape of this mode of frequency is shown in Figure 33.

4.7.4 Harmonic Response Analysis of the System

The harmonic response analysis is performed to calculate the response of the structure at several frequencies. This is a frequency-response analysis where the system response under a steady-state sinusoidal (harmonic) loading at a given frequency is determined. A known external force is applied to the system and due to this excitation force the system will be subjected to displacements at certain frequencies. To understand the dynamic characteristics of the shafting system it is better to do the free vibration analysis or modal analysis prior to this harmonic response analysis always. The steps involved in this analysis is similar to that of Modal analysis except in the part of load application.

There are normally two solution methods applicable for doing a Harmonic analysis – Mode Superposition method and Full method. Among those, Mode Superposition method is chosen for the forced vibration analysis since it is very quick and efficient and basically it is much faster than the Full method. Also in harmonic analysis, based on Mode superposition, the peak response will meet up with the natural frequencies of the system. Also, the simulation can cluster the results near the natural frequency range and this captures the peak response way better than evenly-spaced values.

Along with that, damping is also specified as a global property for the whole model and not specified as a material property. Damping results in the energy dissipation in a dynamic system

and in this case study a constant damping ratio is given over the entire frequency range. The damping ratio is the ratio of actual damping over the critical damping. This value will be used directly in Mode superposition method.

The external loading occurs at a frequency known as excitation frequency. So compared to the free vibration analysis that has done before, here an external force (moment) is applied to the model. This external moment is the torque developing at the crankshaft-flywheel connection point which was calculated analytically in the previous Section 4.5.1. The idea to conduct this forced vibration analysis in TVC is to ensure that the maximum stress values are occurring at the calculated natural frequency range itself. It can also give a clear idea about the most critical torsional natural frequency at which the maximum response of the structure occurs. Peak responses and maximum displacements can be identified from this analysis.

4.7.5 Frequency Response Plot

After performing the harmonic analysis, the response of the shafting system is recorded in a displacement-frequency plot. Figure 34 shows the harmonic response of the model when subjected to an external moment. It is evident from the graph that the displacement rapidly increased at the vicinity of the two torsional natural frequency ranges of the system. But the maximum displacement is happened at the 1st natural frequency value of about 3.85 Hz then drastically reduced and again starts to rise as the frequency increases. This peak response graph denotes the possible critical frequency that is dangerous for the engine and correspondingly the shafting operation

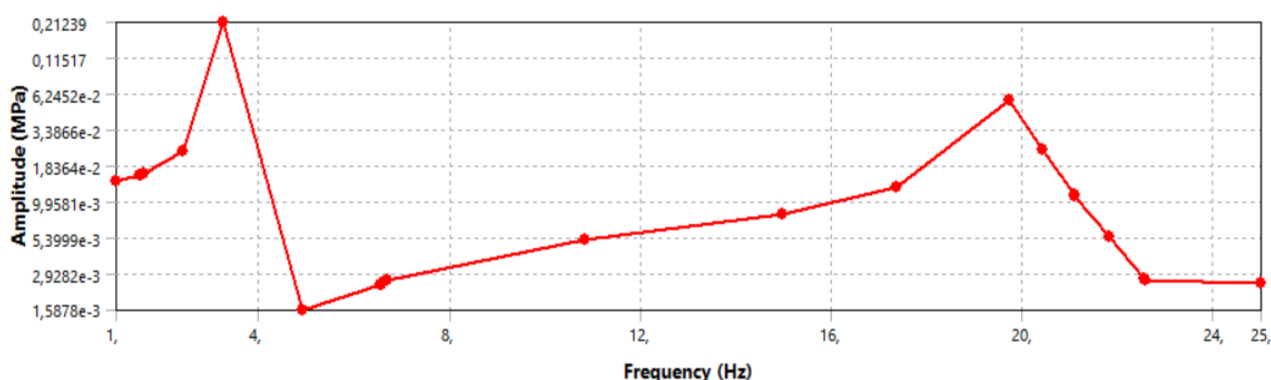


Figure 34: Displacement-Frequency Response of Shaft

5. TORSIONAL VIBRATION ANALYSIS WITH ICE CONDITION

5.1 Problem Description

In recent times, the number of vessels operating in ice conditions increases and the understanding of ice related loads acting on the hull and also at the propeller become significant for safe and convenient design and performance of ice going vessels. In this task, the same 3000TEU Container vessel which was operating in the open water condition is retrofitted with a new propeller capable of navigation in ice covered waters. Figure 35 shows a container vessel which is navigating in ice covered waters. As the propeller is re-designed, it can affect the characteristics of the propulsion shafting system on basis of vibration behaviour. So, the subject of investigation is to determine the ice impact load response on the complete shafting system for TEU Container vessel that is equipped with new 5-bladed propeller.

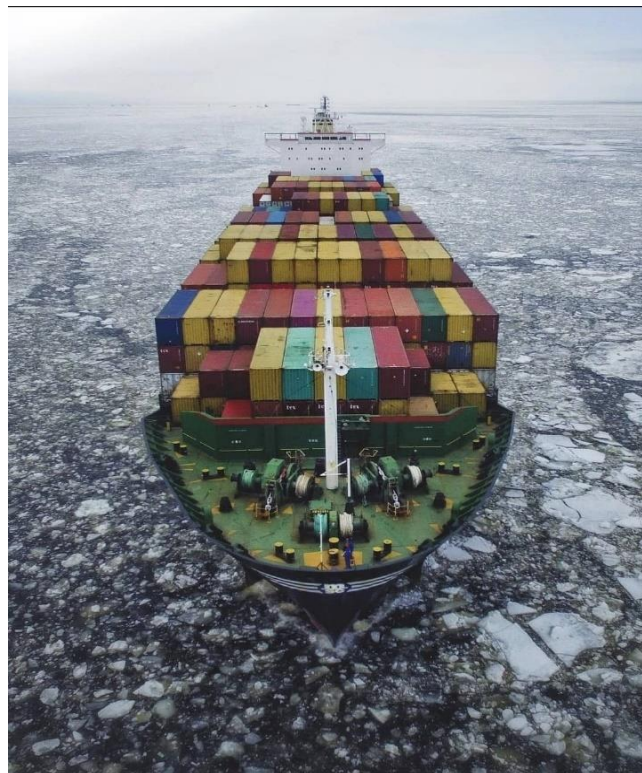


Figure 35: Container Vessel Operating in Ice Channel (https://www.instagram.com/p/Bs6DMEQD-ID/?utm_medium=copy_link)

Here the Torsional Vibration Calculation (TVC) for the propulsion machinery is to be done by a time-domain approach. Earlier in the open water condition when no ice torque is applied, the TVC is performed in a frequency-domain approach by taking the loading as steady-state

oscillations since the rotational speed of shaft is assumed as constant and as well as the remaining shafting system parameters (Yuriy A. Batrak, 2012). The investigation and analysis of torsional vibration of shaft line of a vessel during the propeller-ice interaction is a complicated process due to the non-uniform ice-induced propeller loads.

In ice loading conditions, the torque balance applied to the propeller is interrupted by ice impacts. This non-harmonic behaviour of the ice impact load requires the TVC to be done in the time-domain, where transient loads can be applied and the responses can be calculated. Therefore, a simple model of the propulsion machinery of the Container vessel (with the new propeller characteristics) is to be developed to investigate the influence of ice impacts on the complete shafting system response. It is also recommended to follow the DNV and Finnish-Swedish Ice Class Rules for ice strengthened vessels to identify the ice excitation cases and load responses.

5.2 Propulsion Machinery in Ice

The propeller of a ship navigating through ice channels will be continuously subjected to ice impacts causing severe vibration problems in the shafting. The propulsion system of the Container vessel is a coupled system connecting the marine engine with the propeller through a series of shafts to create a directional thrust (Drazen Polic, 2013). When the vessel is operating in ice covered waters, it typically involves the cutting of ice blocks and their submergence in the direction of the flow of water around the ship hull. This greatly affect the hydrodynamic performance of the propeller, because the hydrodynamic torque applied to the propeller caused by the water inflow to the propeller (wake field) is disturbed. The ice induced contact loads happens on the propeller blades when the submerged ice pieces move towards the advancing propeller. When the propeller blades made contact with the ice blocks, the milling process occurs which is nothing but the cutting of ice blocks into smaller pieces. This operation is known as ice-propeller interaction. Figure 36 depicts a typical scenario of the ice milling process by the propeller blades.

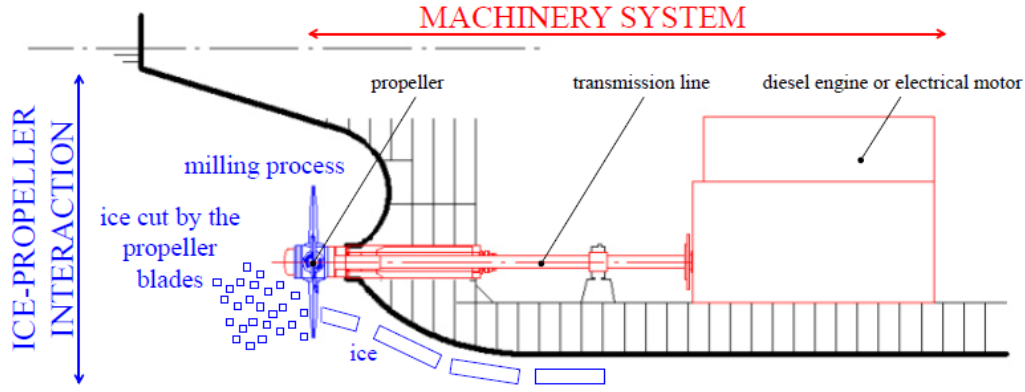


Figure 36: Propulsion Machinery in Propeller-Ice Interaction (Polić D, Ehlers and Pedersen 2014)

Normally, the main engine of the vessel transmits the engine power or torque to the propeller through the shafting and in turn, the propeller transmits the hydrodynamic load resulting from the propeller water interaction. But in this special case of propeller-ice interaction, the load transmitted by the propeller to the propulsion machinery will be a coupled torque that is formed due to the ice contact load together with the hydrodynamic load. This relationship of coupled torque transmission affects the individual components and results in the rise of vibratory stresses.

5.3 Finnish-Swedish Ice Class Rules

The Finnish-Swedish Ice Class Rules (FSICR) define the ice class regulations and requirements for the vessels which are intended to navigate in ice and polar regions. According to this Class rules, there are torsional design loads which should be followed by the marine propellers and shafting that are installed on board. Based on that, the maximum torque on a propeller resulting from propeller-ice interaction during the service life of a vessel is given as (FSICR, 2017)

When $D \leq D_{limit}$,

$$Q_{max} = 10.9 \left(1 - \frac{d}{D}\right) \left(\frac{P_{0.7}}{D}\right)^{0.16} (nD)^{0.17} D^3 \quad (34)$$

When $D > D_{limit}$,

$$Q_{max} = 20.7 \left(1 - \frac{d}{D}\right) \left(\frac{P_{0.7}}{D}\right)^{0.16} (nD)^{0.17} D^{1.9} H_{ice}^{1.1} \quad (35)$$

Where,

$$D_{limit} = 1.8H_{ice} \quad (36)$$

And, d is the hub diameter of the propeller, D is the propeller diameter, $P_{0.7}$ is the propeller pitch at 0.7 radius, D_{limit} is the propeller diameter limit, H_{ice} is the maximum ice block thickness entering the propeller and it is different for different ice class vessels and n is the rotational propeller speed at MCR in bollard condition and its value varies according to the type of propeller. So basically, Q_{max} depends on ship class, propeller and hub parameters

Table 8: Rotational Propeller Speed for Different Propeller Types (FSICR, 2017)

Propeller Type	Rotational Speed, n
Controlled Pitch Propellers	n_n
Fixed Pitch Propellers driven by turbine or electric motor	n_n
Fixed Pitch Propellers driven by diesel engine	$0.85n_n$

Here n_n is the nominal speed at MCR in the free running open water condition of the engine.

Prior to the ice load excitation calculation of the intended propeller, the maximum design ice torque has to be calculated. This is done to perform the time domain calculation of the torsional response in each excitation loading cases. According to the FSICR rules, the time domain calculations shall be done for three engine operating conditions :

- MCR Condition,
- Ice Bollard Condition and
- Blade Order Resonant Condition.

The Finnish-Swedish Rules describes four excitation cases containing different scenarios where a propeller is milling an ice block, for the strength and stress evaluation of the propulsion line. These cases are defined with an intention to reflect the operational loads on the shafting system, when the propeller gets in contact with ice blocks, and the corresponding response of the complete system. This transient dynamic analysis of ice torque excitation containing a series of blade impacts are defined as half sine shape functions. This excitation cases are further used for the ice torque loading profiles in the shaft line dynamic simulation stage. The four excitation cases and the relevant factors for torque calculation are presented in Table 9.

Table 9: Torque Excitation Factors for Different Ice Cases (FSICR, 2017)

Torque Excitation	Ice-Propeller Interaction	C_q	α_i (deg)			
			$Z = 3$	$Z = 4$	$Z = 5$	$Z = 6$
Excitation Case 1	Single ice block	0.75	90	90	72	60
Excitation Case 2	Single ice block	1.0	135	135	135	135
Excitation Case 3	Two ice blocks (Phase shift $360/(2 \times Z)$ deg)	0.5	45	45	36	30
Excitation Case 4	Single ice block	0.5	45	45	36	30

In the above table, C_q is the ice impact magnification factor and α_i is the duration factor. The 3000TEU Container vessel taken for the TVC analysis has a 5-bladed propeller intended to operate in the ice region. Hence the values of duration of the propeller blade-ice interaction expressed in rotation angle (α_i) corresponding to the blade number ($Z = 5$) is taken for the ice torque loading calculation. During the ice interaction sequence, the excitation frequency shall follow the propeller speed. The ice torque developed due to a single blade impact on the ice with respect to the propeller rotation angle can be then defined as,

For $\varphi = (0, \alpha_i)$

$$Q(\varphi) = C_q Q_{max} \sin(\varphi(180/\alpha_i)) \quad (37)$$

For $\varphi = (\alpha_i, 360^\circ)$

$$Q(\varphi) = 0 \quad (38)$$

Where φ is the angle of rotation when the first impact occurs and α_i is the duration of the ice and propeller blade interaction shown in terms of propeller rotation angle. Figure 37 represents the ice torque graphs for the proposed 5 bladed propeller at a constant propeller speed for a duration of three rotations of the propeller corresponding to each excitation case.

In the graphs, Case 1 and Case 2 represents a 72° and 135° single blade impact sequence respectively. While Case 3 denotes a 36° double bladed impact sequence and finally Case 4 shows a 36° single blade impact sequence. These curves are further used for the torque response calculation by the transient analysis approach. According to the proposed Class Rules, these

excitation ice torques are used at three different operating points in the engine running condition in order to evaluate the vibratory torque on the propulsion system.

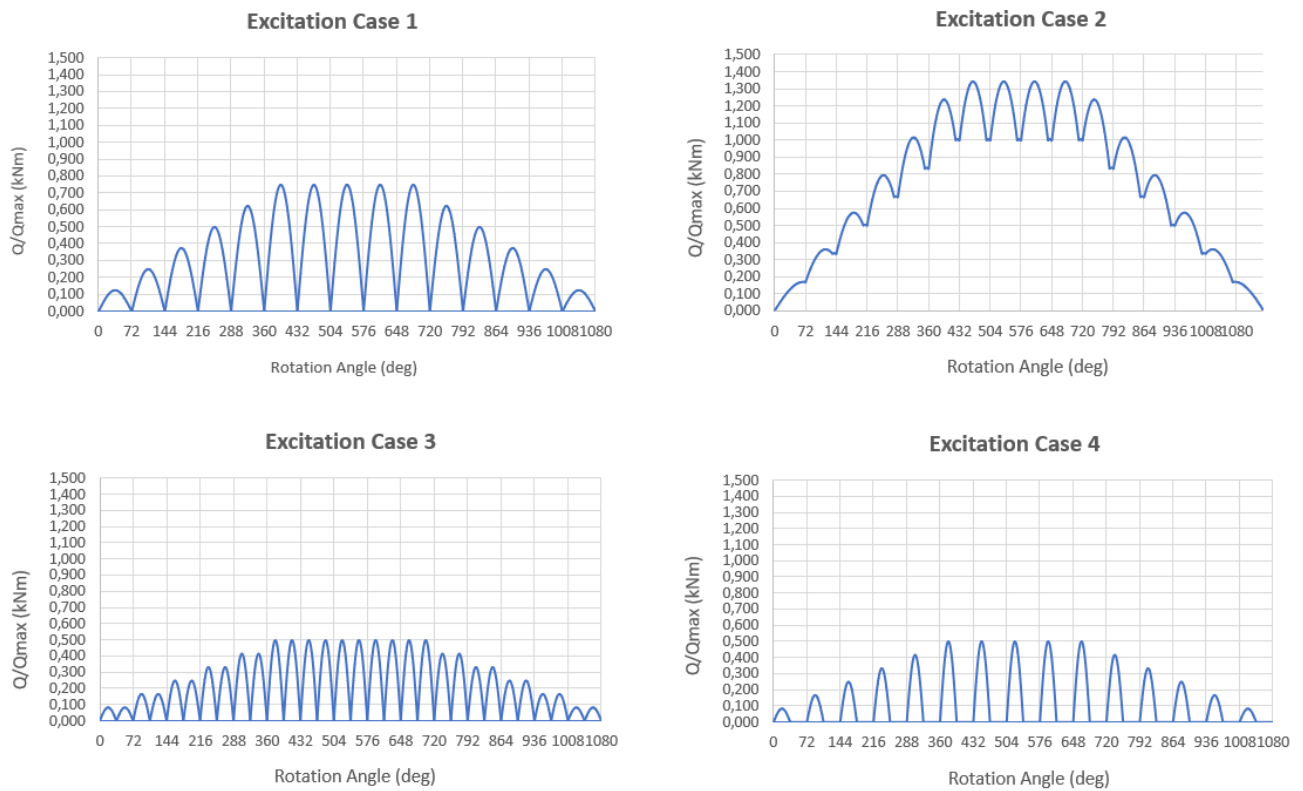


Figure 37: Ice Excitation Torques for 5-bladed Propeller

5.4 Calculation Model of the Ship Propulsion System

In order to model the propulsion system for the TVC analysis in ice conditions, the characteristics of the main engine and the technical data of the propeller are used. All these data are provided in Section 4.2.1. Compared to the shafting model used in the open water conditions, the model intended to use for the ice loading simulation must have an additional effect of all the propeller blades. Hence a modified solid model of the system is developed with the help of Siemens NX and ANSYS Spaceclaim software.

5.4.1 General Aspects of the Model

The model contains all the sufficient components required for the transient analysis. The main shafting model consists of a flywheel, generator shaft, generator, flange couplings, intermediate

shaft, propeller shaft, propeller hub and equivalent parts representing propeller blades. All the parts are modelled as solid elements in the 3D modelling software. In this condensed model of the system, the intermediate bearings are not modelled because the effect of loading is happening from the two end sides of the propulsion system. There will be an engine torque applied at the connection point between crankshaft and flywheel and similarly there will be a hydrodynamic torque or propeller moment and in addition the ice torque applied from the propeller side. This as a whole restrict the movement of the dynamic system as a rigid body while doing the simulation. Figure 38 depicts the propulsion 3D model used for the TVC analysis in ice condition.

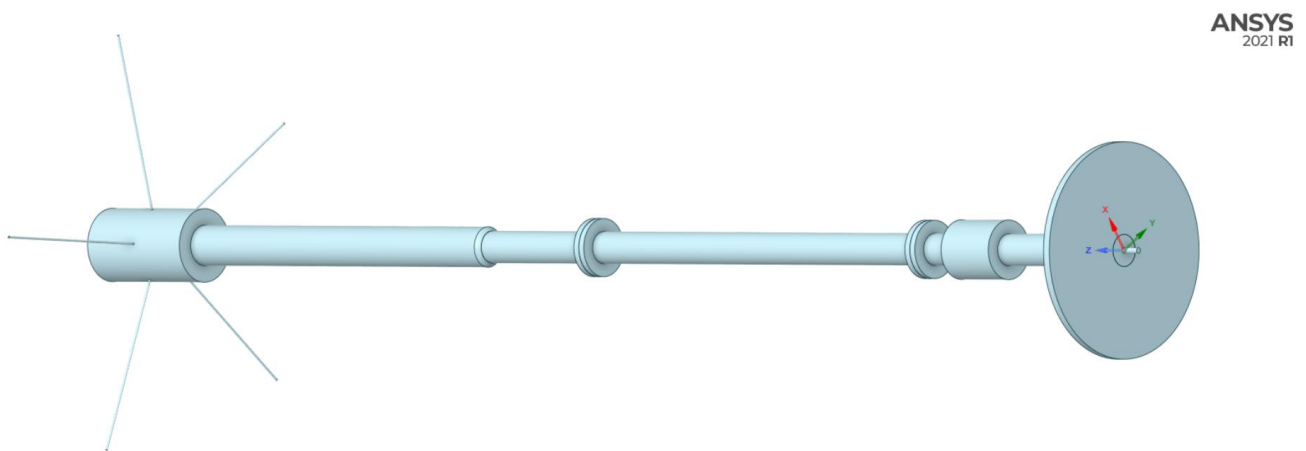


Figure 38: Model of the Propulsion Shafting of the Ice Class Container Vessel

5.4.2 Mechanical Properties of the Model

The parameters sufficient for the vibration analysis of the analysed system are listed below. The values are the ones that are used in the TVC analysis of the free running open water conditions.

Table 10: Mechanical Properties of the Model

Components	Mass Moment of Inertia, J (Kgm ²)	Torsional Stiffness, k (MNm/rad)
Flywheel	15889.10	329
Generator	3589.80	1155.90
Flange Coupling 1	515	107.20
Flange Coupling 2	1262.80	151.40
Propeller	121508	-

5.5 Excitation Sources for Vibration

In the case of ice class vessels equipped with propulsion systems intended to operate in ice channels, the vibration excitation forces on the shafting are mainly emerged from machinery components like the main engine, the disturbances or resistance from the surrounding water of the propeller and finally the ice-propeller blade impacts. The origin of engine excitation forces are already discussed in detail in Section 3.1.1. The description about the engine torque determination (in time-domain) at the connection between flywheel and crankshaft, together with the tangential pressure components estimation are given in the next sections.

5.5.1 Engine Torque Determination

The engine torque is basically the sum of the tangential components of periodically recurring gas forces and inertia forces of the engine moving parts multiplied with the crank radius. These forces are transferred to the crankshaft by the piston and connecting rod arrangement. The tangential pressure inside an engine cylinder can be obtained by simply dividing the engine recurring torque by the piston area. The tangential pressure is normally decomposed into two components called gas tangential pressure and inertia tangential pressure. Then by the help of the Fourier series analysis, these parts can be further split into the form of harmonic components (sine and cosine functions) or harmonic coefficients. So basically, the resulting excitation of the engine combines the influence of the gas pressure and the inertial forces.

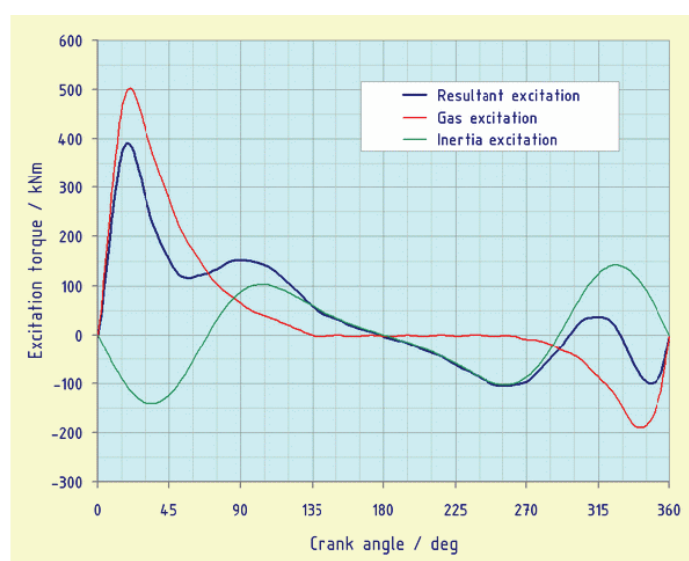


Figure 39: Excitation Torque in a Cylinder for a Typical Low Speed Diesel Engine for a Complete Revolution of Crankshaft (Gojko Magazinović, 2002)

Figure 39 illustrates an example of a typical engine excitation torque after considering the influence of gas pressure and inertial forces. The excitation of torsional vibrations are evaluated by the piston's stroke and mean effective pressure. For the ice based TVC analysis of the Container vessel, the engine manufacturer already provides the harmonic coefficients for the mean indicated pressure of the engine cylinder. So, by performing the Fourier series, the total gas tangential pressure and mass tangential pressure for a single cylinder can be calculated by using the harmonic coefficients values of the mean effective pressure.

5.5.2 Tangential Pressure Components

The harmonic coefficient values corresponding to the mean indicated pressure are given in the gas harmonics data provided by the engine supplier. By the given trigonometric Fourier series calculation using the coefficient values (1st order to 16th order), the gas tangential pressure for a single cylinder will be obtained.

$$P_{Gas}(\varphi) = a_0 + \sum_k a_k \cdot \cos(k \cdot \varphi) + \sum_k b_k \cdot \sin(k \cdot \varphi) \quad (39)$$

Where, a_k and b_k are the sine and cosine components of the mean pressure, k is the firing order and φ is the crank angle in degree. Figure 40 shows the obtained course of the gas tangential pressure.

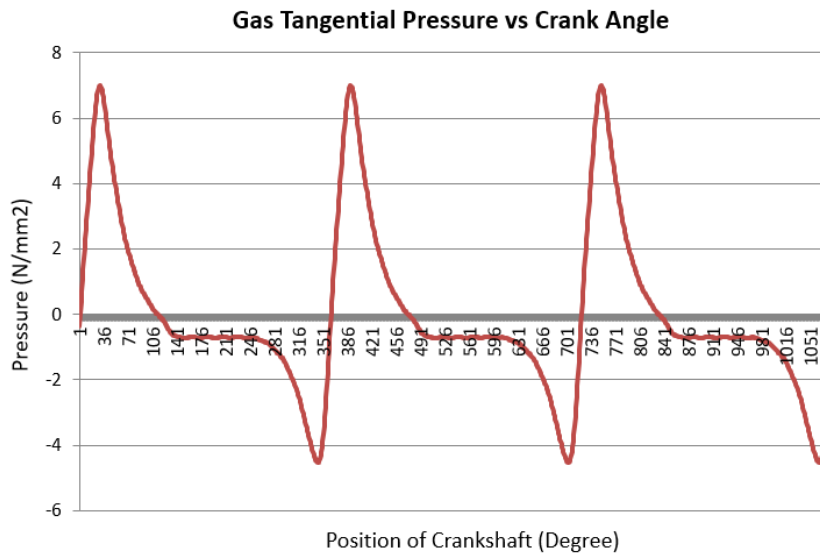


Figure 40: Gas Tangential Pressure for an Engine Cylinder During Crankshaft's Three Revolutions

Now the mass tangential pressure can be determined with the engine specifications data since the oscillating parts of the engine affects the engine torque. The fourier series used for the estimation of the mass tangential pressure can be expressed as follows:

$$P_{Mass}(\varphi) = \frac{m_{osc} \times r \times \omega^2}{A_p} \cdot \sum_k B_k \cdot \sin(k \cdot \varphi) \quad (40)$$

Where, m_{osc} is the mass of oscillating parts, r is the crank radius (*piston stroke/2*), ω is the angular frequency and A_p is the area of piston movement. According to the engine data, m_{osc} is already given and with the MCR speed of the engine, the angular frequency is calculated. The coefficients of B_k can be calculated in a defined periodic way and are only calculated for the first four firing orders ($k = 1, 2, 3, 4$) and this is because the value of the connecting rod ratio (λ) is raised to higher orders from 5th order onwards and the value of the oscillating masses will be negligible at higher orders of λ . Figure 41 depicts the course of inertia pressure for a single cylinder.

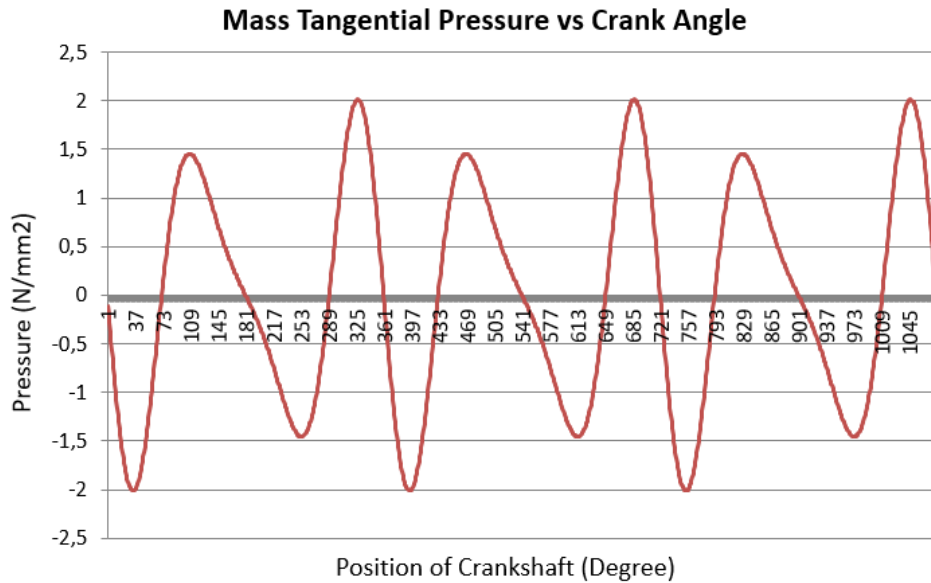


Figure 41: Mass Tangential Pressure for an Engine Cylinder During Crankshaft's Three Revolutions

Hence the total tangential pressure developing inside an engine cylinder can be obtained by the summation of both the tangential pressure component curves and the resulting graph is shown in Figure 42.

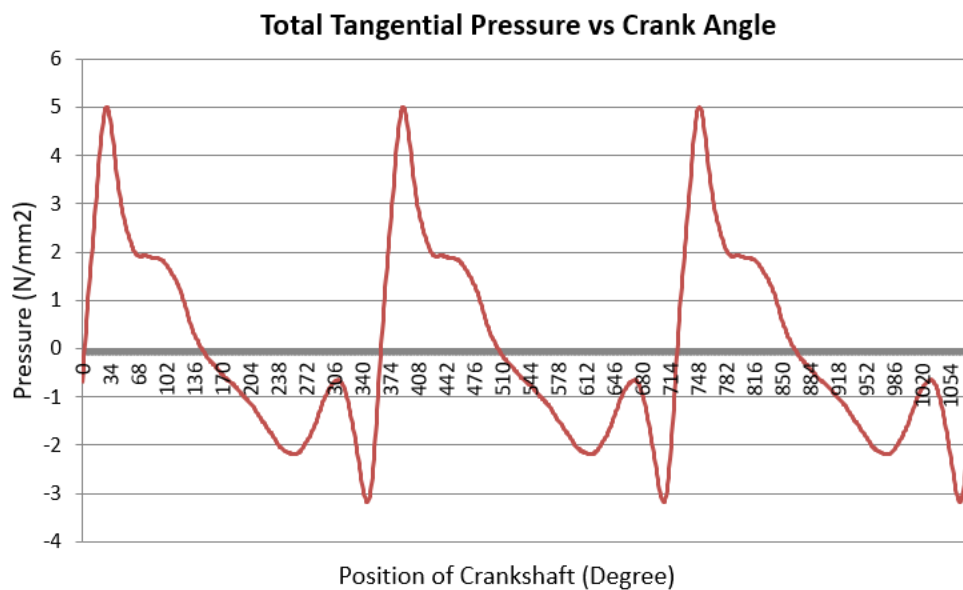


Figure 42: Total Tangential Pressure for an Engine Cylinder For Three Revolutions

Now by evaluating the tangential pressure coefficients, the engine moments can be derived. But the engine supplier has already provided the torque curves at the flywheel connection point for the three different engine operating conditions (MCR, Ice Bollard and 1st Blade Order) which are sufficient for the time-domain TVC calculations as per Finnish-Swedish Rules.

These curves are shown in Figures 43-45. The three engine torques corresponding to three different operating points used for the TVC analysis is illustrated below.

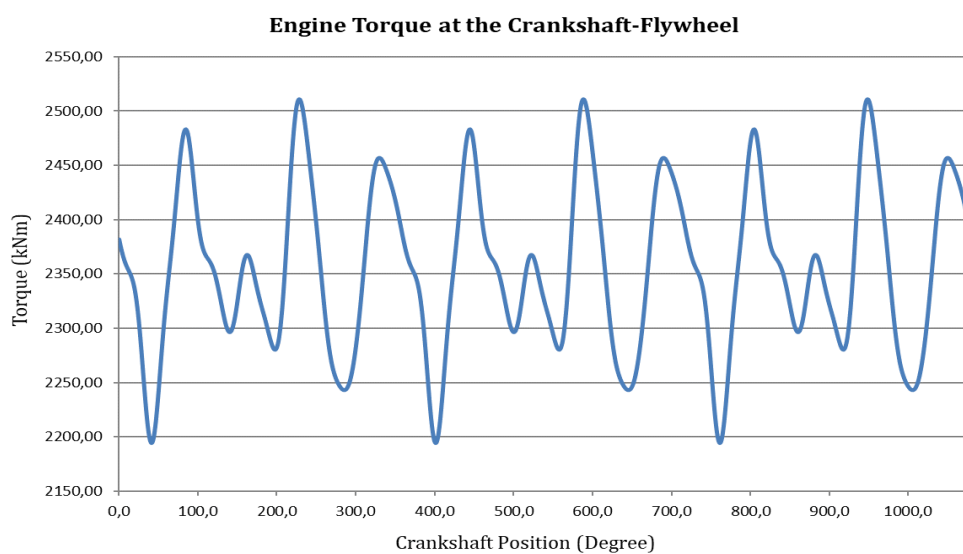


Figure 43: Engine Torque for the MCR Condition

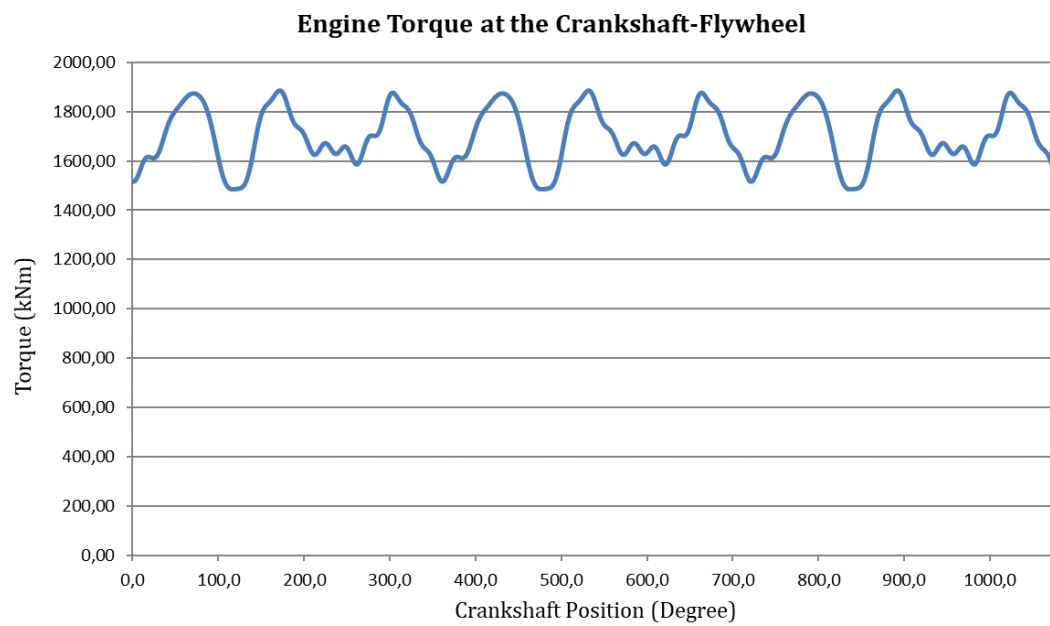


Figure 44: Engine Torque for the Ice Bollard Condition

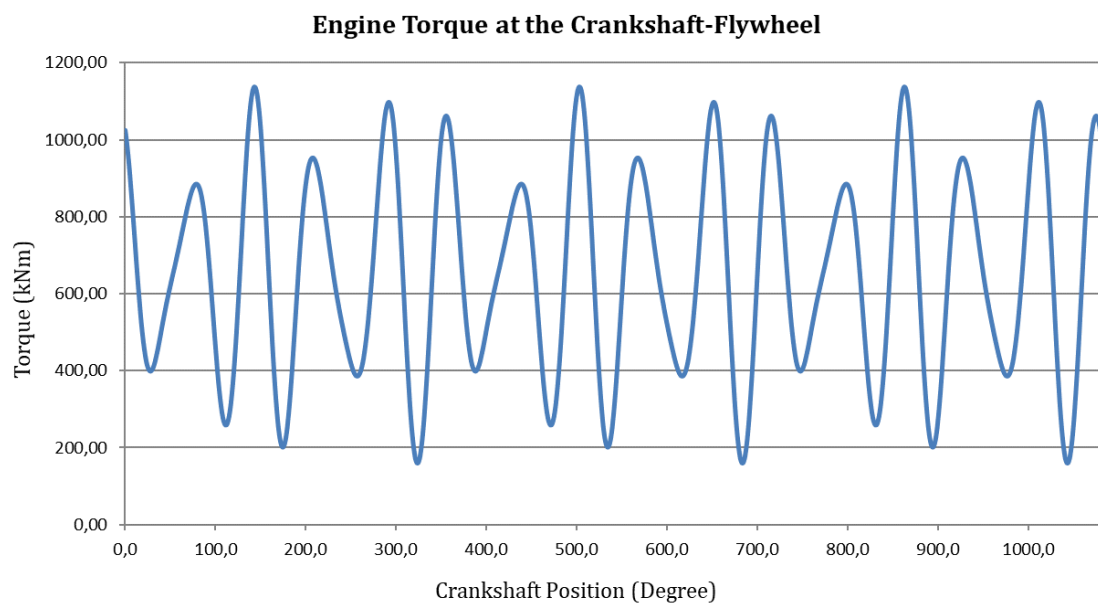


Figure 45: Engine Torque for the 1st Blade Order Condition

The aim of the work is to use these variable torque curves as the excitation engine torque applied at the flywheel side to get the load response from the shafting in the time domain analysis.

5.6 FE Model of the Propulsion System

The characteristics of the propulsion system in the numerical model is considered in different ways. The components including flywheel until the propeller hub are made as flexible elements whereas the propeller blades are idealized by point masses which are connected to the hub by rigid elements. This is made to ensure the influence of actual propeller blades during the simulation stage. In ice loading cases, the effect of ice torque is being affected at the propeller side and so the propeller blades are significant in this type of analysis. Hence mass elements that are modelled at exact coordinates of the blade tips acts as the blades of the propeller with the actual mass and mass moment of inertias. In the FE analysis, the maximum occurring torsional torque in the complete shafting is to be determined using this condensed model of the propulsion system. Figure 46 illustrates the developed geometry of the shafting system in ANSYS.

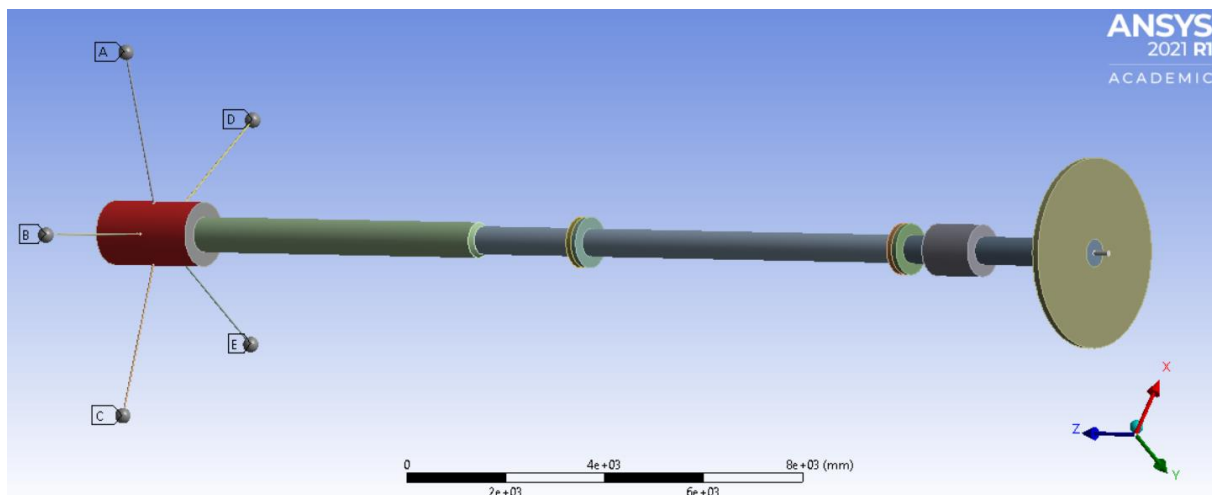


Figure 46: FE Model of the Shafting system

5.6.1 Application of Different Moments in the Model

In the FEM analysis, three different moments or torques are to be applied at different locations along the geometry. The oscillating moments or the engine moments are applied at the outer extruded part of the flywheel (representing the connection point between the crankshaft and flywheel). Figure 47 shows the position where the engine torque is applied in the shafting system.

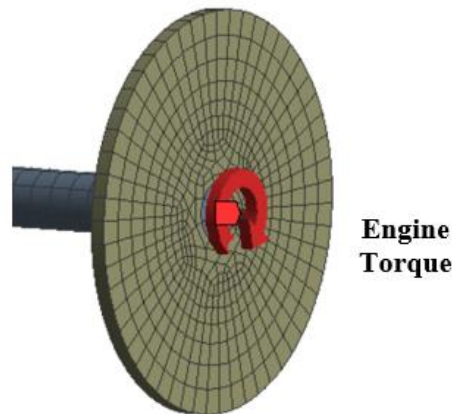


Figure 47: Position of Engine Torque Applying in the System

The propeller moment or hydrodynamic torque is assumed to be similar to the engine torque which is calculated according to the engine power and constant rotation speed of the propeller for different operating conditions. This moment is applied at the outer surface of the propeller hub. In the model, the connections between the single point masses (equivalent propeller blades) and the hub are made as rigid elements with respect to the propeller hub, hence in effect any load that is applied at the hub surface will be get transmitted to the shafting only after considering the additional effect of blade masses. In normal scenario, the ice torque is only getting in contact with the propeller blades, so to idealise that the ice loads are also made to be applied at the outer hub surface. This gives the counter effect of propeller moment and ice torque and the resulting coupled torque will be transmitted along the shaft. Figure 48 illustrates the position of the propeller and ice torques applied in the model.

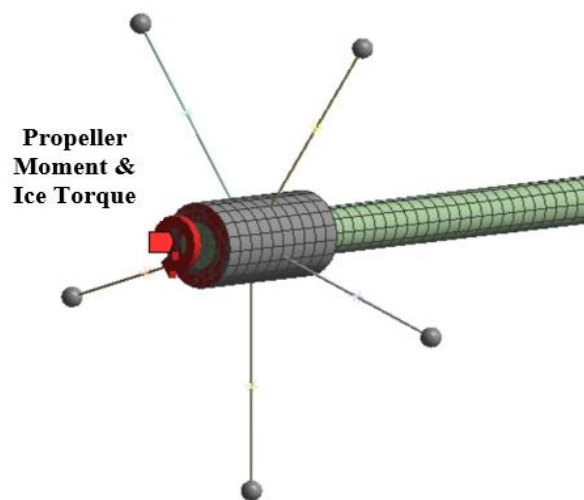


Figure 48: Position of Propeller Moment and Ice Torque Applying in the System

The propeller moment considered for the three different engine operating points are listed in the following Table 11.

Table 11: Propeller Torque for Different Engine Operating Points

Operating Condition	Engine Power (kW)	Engine Speed (rpm)	Propeller Torque (kNm)
MCR Condition	21700	88	2355
Ice Bolard Condition	13327	74.8	1701
1 st Blade Order	3160	46.2	652

5.6.2 Load Cases Considered for the Analysis

The finite element transient analysis has to be performed for all defined ice loads (four excitation loading cases) for the three different operating conditions of the engine. In addition to that an extra case which does not take the effect of ice impact has also been investigated for all the operating points as well. A summary of all the considered load cases in the TVC analysis of the propulsion shafting system is tabulated below.

Table 12: Load Cases Considered for Transient Analysis of the System

Engine Operating Condition	Ice Excitation Cases	Load Case
MCR Condition	Without Ice Load	A1
	Ice Torque 1	A2
	Ice Torque 2	A3
	Ice Torque 3	A4
	Ice Torque 4	A5
Ice Bollard Condition	Without Ice Load	B1
	Ice Torque 1	B2
	Ice Torque 2	B3
	Ice Torque 3	B4
	Ice Torque 4	B5
1 st Blade Order Condition	Without Ice Load	C1
	Ice Torque 1	C2
	Ice Torque 2	C3
	Ice Torque 3	C4
	Ice Torque 4	C5

5.7 Transient Analysis of the Propulsion System

To perform the time-domain calculation of the torsional response of the shafting system, a transient analysis has been conducted in ANSYS Mechanical software. The transient analysis is used in order to calculate the response of the system for time varying loads. The main procedure involved in this dynamic analysis is as follows:

- Geometry Modelling
- Assign Material Properties
- Specify Contacts and Initial Conditions
- Apply meshing
- Apply time settings, loads and supports
- Solve the model

The model used in the ice case simulation is different from the one used in the open water conditions. The model is updated with the influence of propeller blades at the hub side of the propulsion system. The material properties are similar to the one considered in the earlier stage. The material applied for every component is S355, a medium tensile strength grade steel. An automatic meshing method is chosen for the complete model with an element size of 150mm. The contacts between each part are assigned as bonded type of contacts which will enable the complete system to transmit the torque in the normal way. All the material properties regarding the S355 steel are already given in the Section 4.7.2. The meshed geometry used for the simulation is shown in Figure 49.

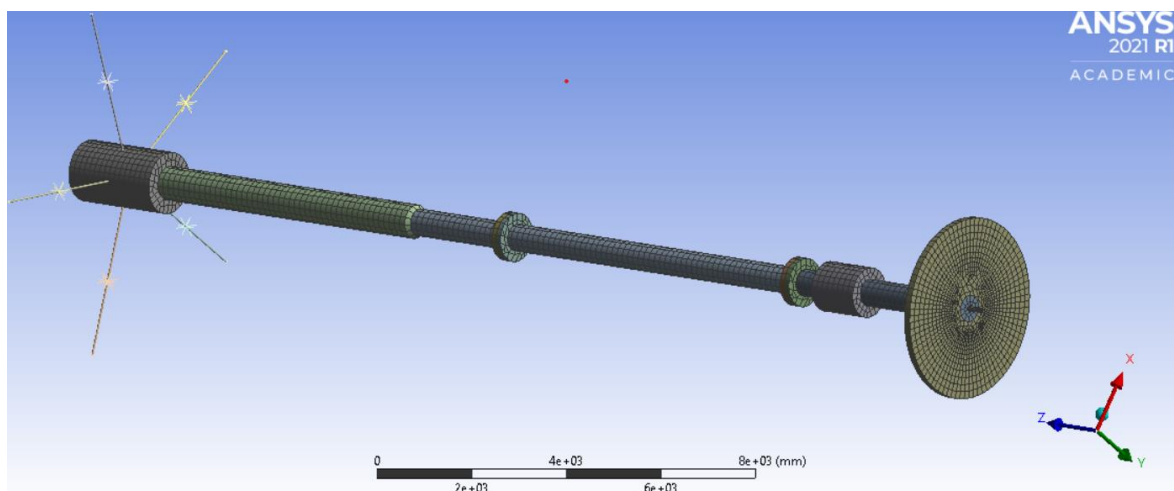


Figure 49: Meshed Geometry of the Shafting System

The time analysis settings used for the transient analysis is summarized in the below Table 13.

Table 13: Time Analysis Settings for the Transient Analysis

Solution method	Direct Transient
Number of transient time steps	1080
Time per step	1 sec
Overall structural damping coefficient	0.05

The propulsion system model with all the relevant moments acting at the different positions are assigned by considering some factors. The propeller moment is assumed to be equal to the engine torque and also to be constant for all the load steps throughout the analysis. Hence it is the engine torque and ice torques are the ones made to be the varying loads with respect to the time. In each analysis of the load cases mentioned in the Table 12, the oscillating engine torques described by the graphs shown in Figures 43-45 and the ice torque curves depicted in Figure 37 are used accordingly for each engine operating points.

5.8 Results and Observations for the TVC Analysis in Ice Conditions

The results of the transient analysis for each loadcase according to Table 12 is summarised in this section. The final occurring torque on the propulsion shafting is determined for each load cases for a duration of three complete revolutions of the crankshaft.

5.8.1 Results for Operating Point – MCR Condition

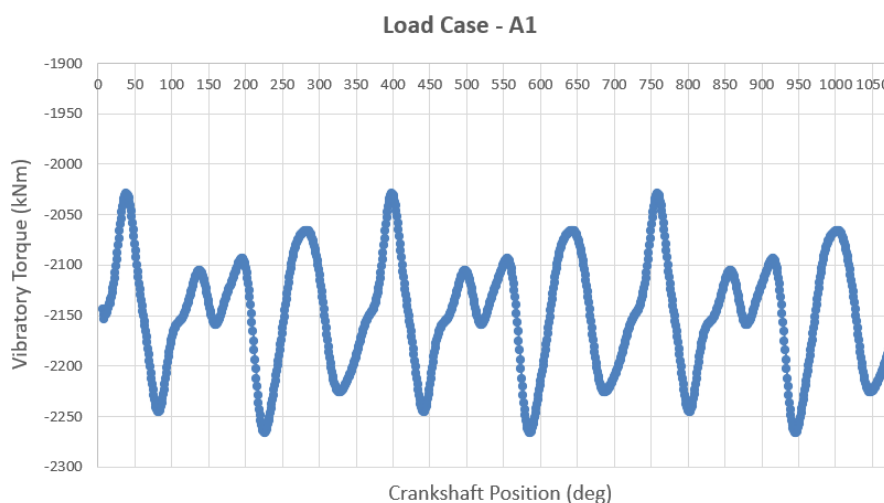


Figure 50: Torque Response in Propulsion Shaft for Loadcase A1 (No Ice Impact)

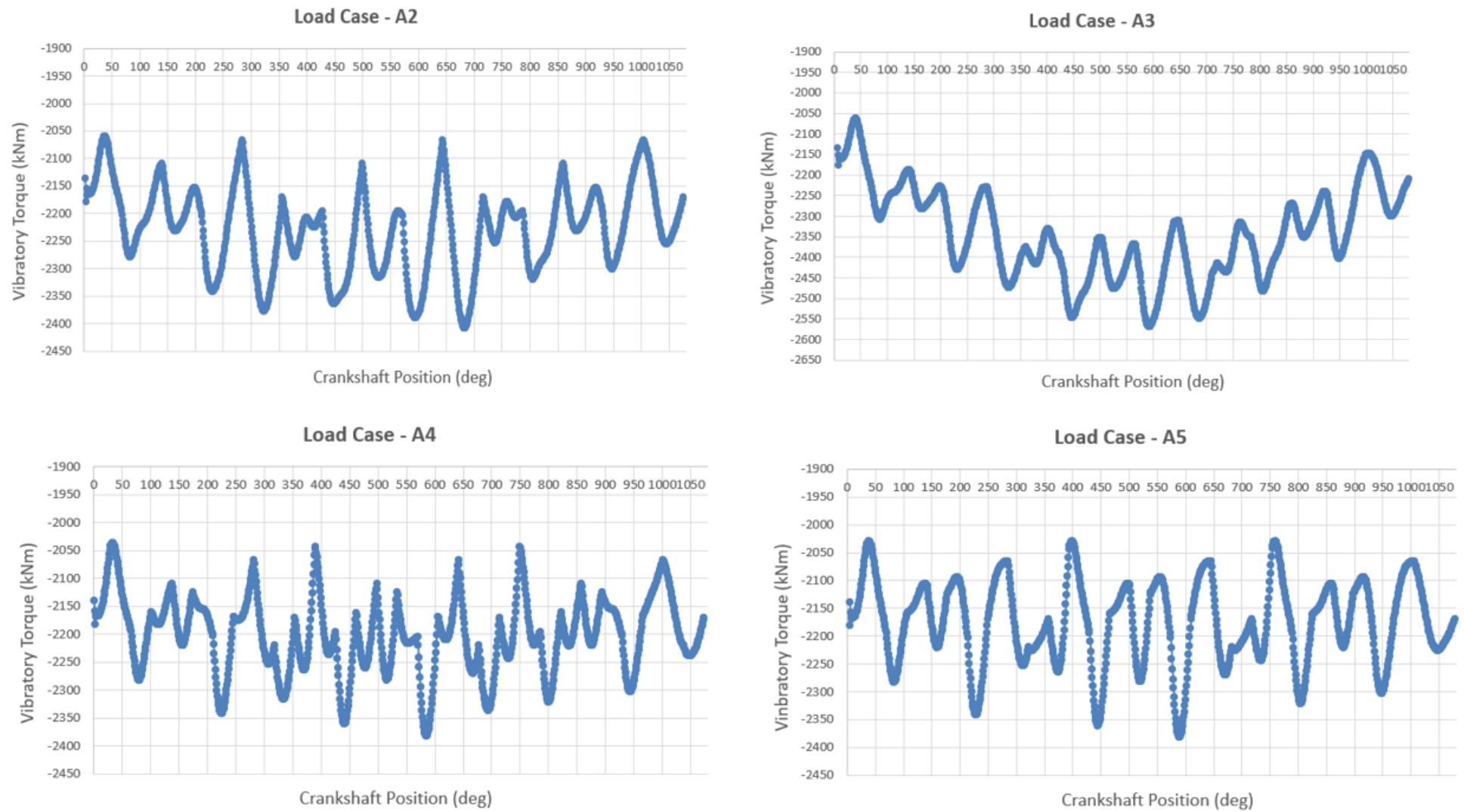


Figure 51: Torque Response in Propulsion Shaft for Load Cases A2 to A5 (MCR Point)

The observations made from the results obtained are:

- Load case A1 has only the effect of engine torque and propeller moment, and among that variable engine torsional moment component is the one causing excessive stresses.
- In all other load cases, the super imposing of propeller moment and ice torque enables the high stresses during the second revolution of the propeller blades (between 360° and 720°).
- For load case A2, the maximum torque on propulsion shaft is occurring at an angle of 687° . This shows that the blade impact sequence for a single blade to cut the ice block is high and the duration of impact is longer compared to other cases.
- In load cases A3, A4 and A5, the maximum stress is developing at an angle of blade rotation of 592° . This trend indicates that the torsional stresses on blades and shaft line are always maximum well before the propeller completes two complete revolutions.

5.8.2 Results for Operating Point – Ice Bollard Condition

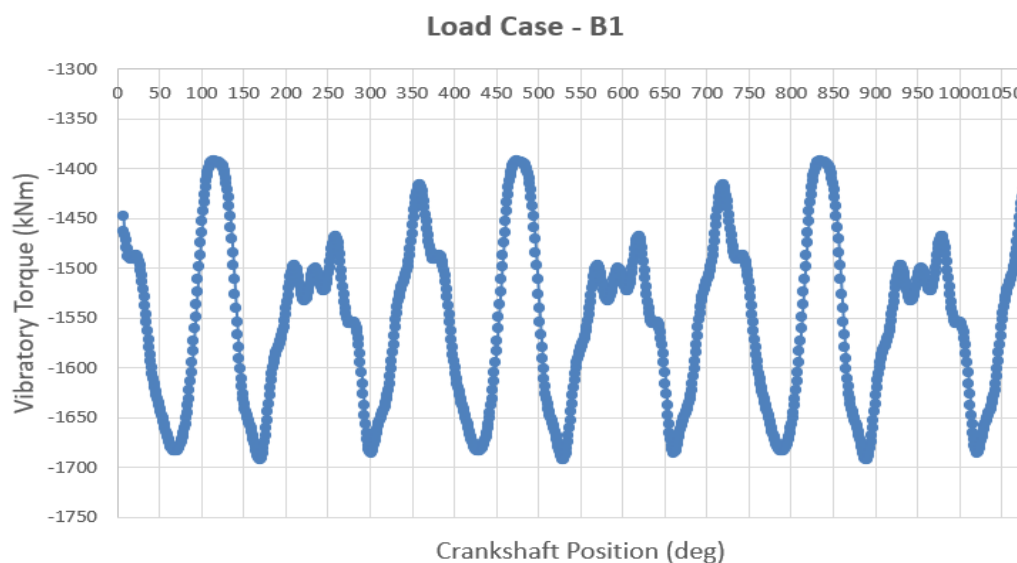


Figure 52: Torque Response in Propulsion Shaft for Loadcase B1 (No Ice Impact)

The interpretation of the results are:

- When the propeller operates at 85% of MCR speed (bollard condition), there can be many small peaks arising along the revolution, but the stresses are below the yield stress.
- The impact of ice blocks on propeller blades are relatively on moderate level at the starting and end of milling process compared to MCR condition, where it was smaller.
- The maximum torque is occurring for the load case B2, because of the long duration of the impact sequence of a single blade on a single thick ice block.

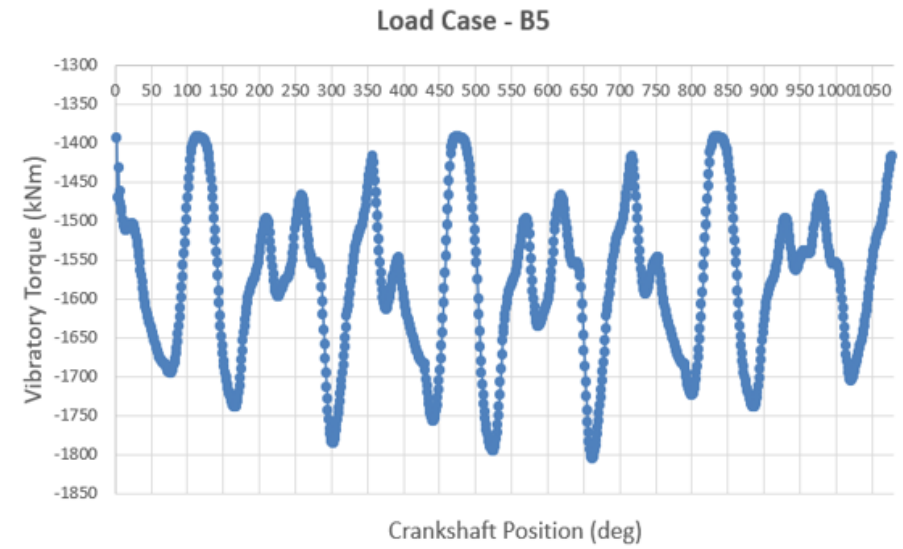
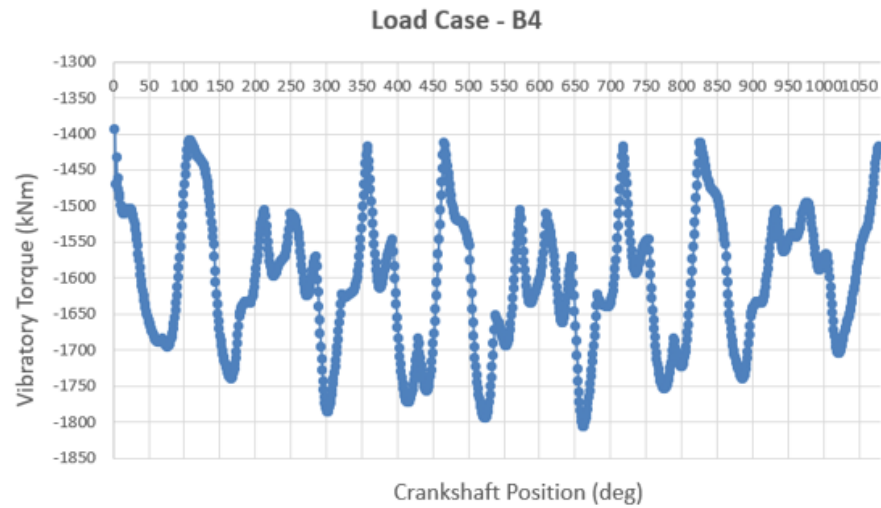
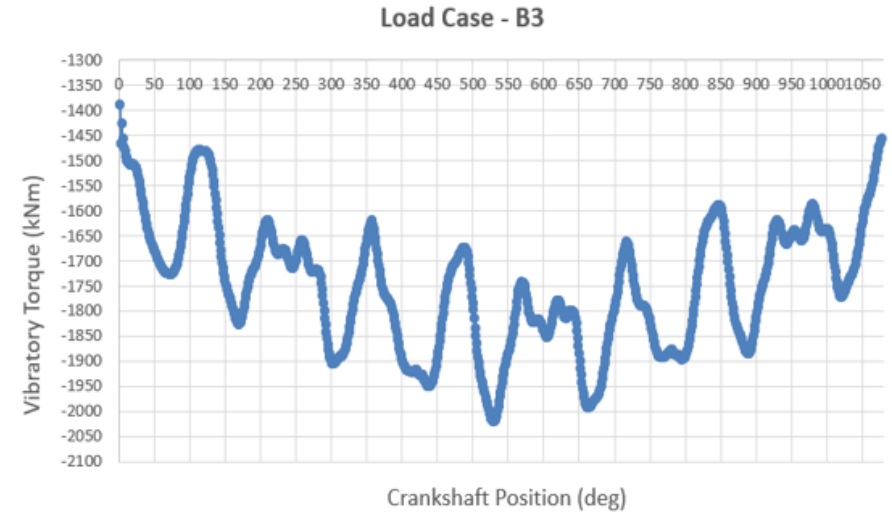
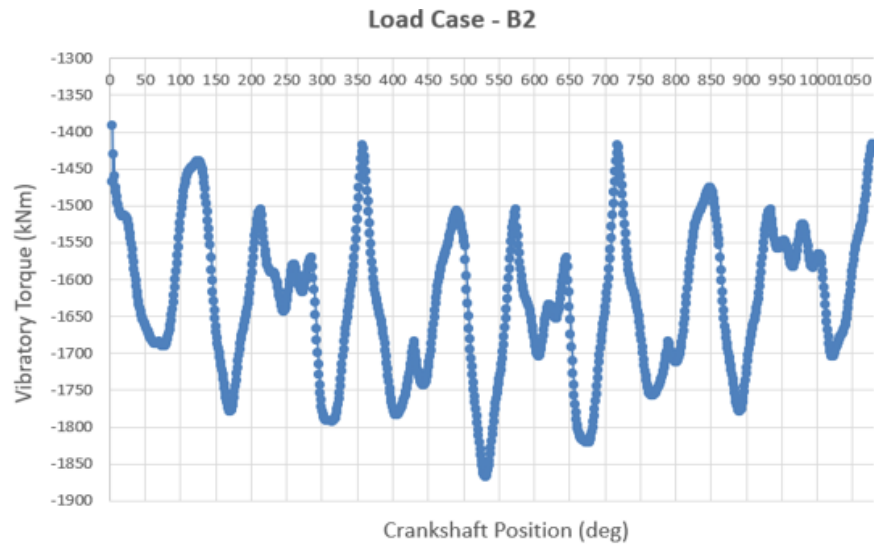


Figure 53: Torque Response in Propulsion Shaft for Load Cases B2 to B5 (Ice Bollard Point)

5.8.3 Results for Operating Point – 1st Blade Order Condition

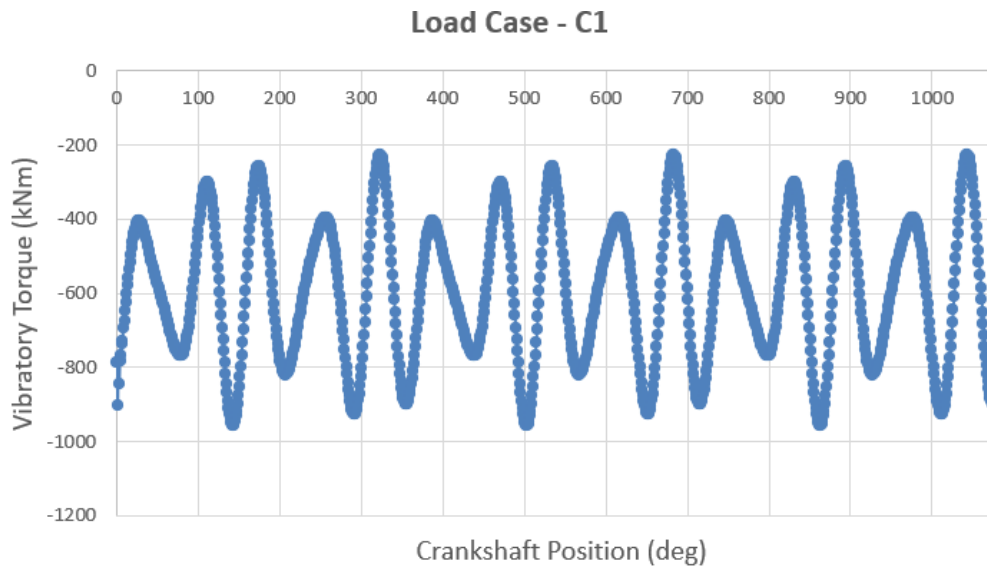


Figure 54: Torque Response in Propulsion Shaft for Loadcase C1 (No Ice Impact)

The observations after doing the time-domain analysis of the 1st blade order condition are:

- The torque response in the propulsion shafts is almost identical in all the curves, except the trend followed at the midway of milling process.
- Load Case C1 is only subjected to engine torque and propeller moment and the torque variation follows a harmonic nature throughout the duration of rotation of blades.
- For load cases C2 and C3, the maximum torque attained at 501° and 503° degrees of blade rotation respectively. As the engine is operating at very low speed (46 rpm), the rotation of propulsion shaft causing blade impacts will happen slowly.
- The maximum stress for load cases C4 and C5 occurred at 655° degrees, since in both these cases, the duration of single blade impact on ice blocks are same.
- Among all, the maximum shaft torque is exhibited for the load case C3, where the propeller blades are meant to be cutting two ice blocks (thinner in size) at a blade angle of 36° but causing double bladed impact sequence on the shafting.

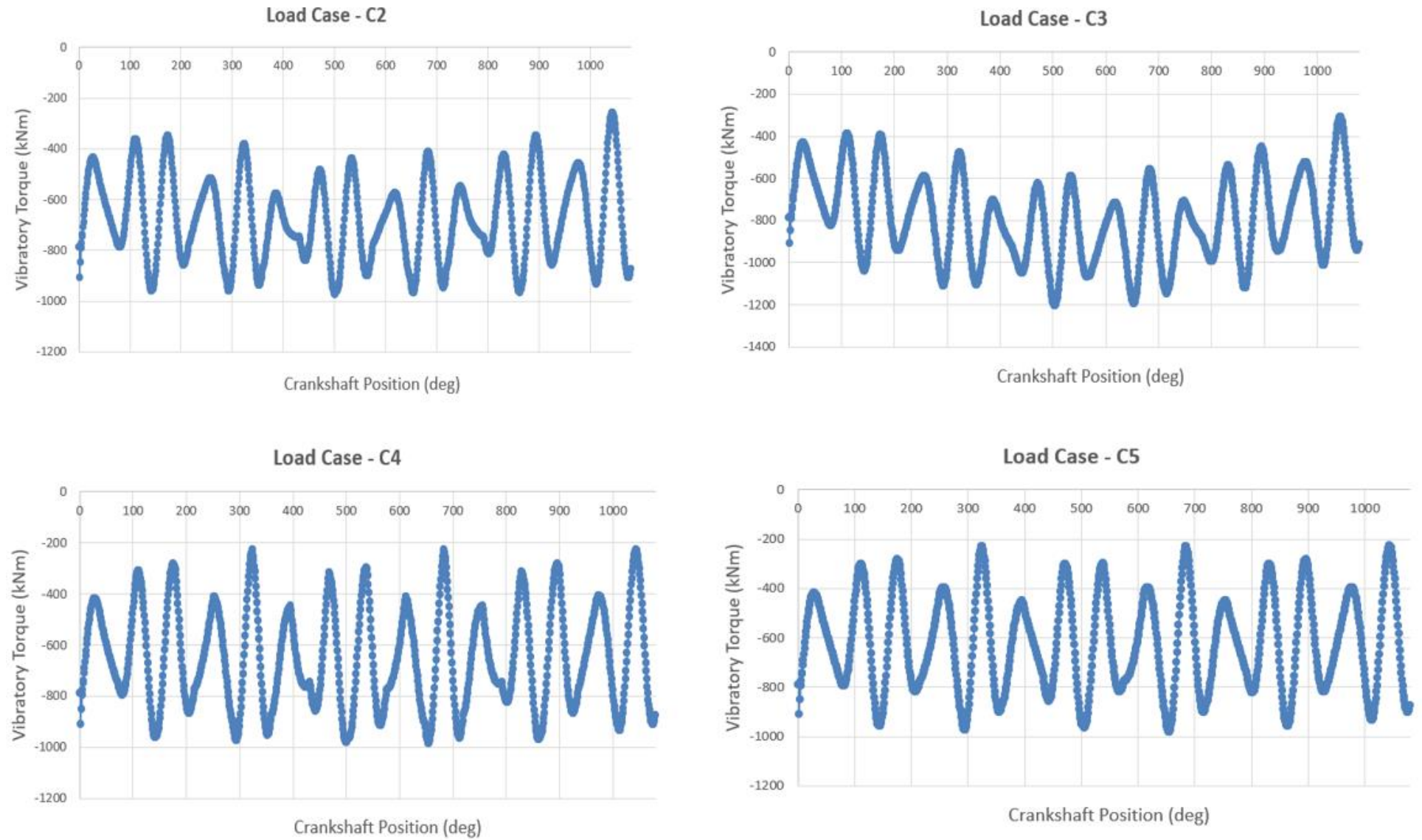


Figure 55: Torque Response in Propulsion Shaft for Load Cases C2 to C5 (1st Blade Order Point)

6. EVALUATION OF THE RESULTS

6.1 TVC Analysis for Without Ice Condition– Review of Result

The FEM procedure developed further to the implementation of the analytical calculation method has been done in frequency-domain approach to meets the needs of a typical Torsional Vibration Calculations (TVC) analysis in without ice conditions. The methods used in the analytical and numerical approaches are seem to be logical after comparing the results. The FE model prepared for the finite element simulation will help to analyse the results in the shafting design stage itself for the shaft designers and marine engineers.

The aim was to obtain comparable torsional natural frequencies of the FE model and also to investigate and confirm the critical frequency where the maximum stress is occurring. A proper 3D shaft structure model to the original shafting system helps to simulate and solve the FEM model very rapidly for global strength and stress analysis. After taking and comparing the TVC data of the shafting system with the old propeller, the natural frequencies obtained for the propulsion system with the new propeller are came to be almost similar. The results for both the cases are tabulated below in Table 14 and it can be seen that only the 1st vibratory natural mode will be seriously influenced by the propeller change or during the retrofitting process.

Table 14: Natural Frequencies of Two Propulsion Plants

Mode	CASE A (Original Propeller)	CASE B (New Propeller)
1 st Mode	3.32	3.85
2 nd Mode	18.61	18.62
3 rd Mode	35.62	35.62
4 th Mode	46.48	46.5

From the results of the TVC analysis of the propulsion system of the Container vessel adapted for the ice conditions, it is clear that the analytical results of the natural frequencies are seem to be almost similar to the numerical values. This shows the accuracy of the analytical approximation method (Holzer's Method) chosen for the analysis. Table 15 shows the comparison between the natural frequencies obtained in both cases.

Table 15: Comparison of Natural Frequencies of the Propulsion Unit With New Propeller

Mode	Holzer's Method (Hz)	FEM (Hz)	Difference (Hz)
1 st Mode	3.85	3.23	0.62
2 nd Mode	18.62	19.72	1.1

On comparing both the stress results between the existing propulsion system and the case with the new propulsion system, it shows that the main resonance, which usually signify as a system's critical speed, occurs when the system vibrates in phase with the 7th order excitation. All the calculated stresses are lying well below the permissible limits provided by the DNV for continuous operation. In all the shafts, the major resonance point is observed at the same location which usually indicates the main critical speed (barred speed range) of the system; and the range is $n = 30 - 36$ rpm. This major resonance occurs when the whole shaft vibrates in phase with the n^{th} order excitation. The power transmission system of the container vessel is run by a two-stroke engine, and so n will be equal to the number of engine cylinders.

Among all the shafts, the maximum stress is occurring for the intermediate shaft with a value of 50 N/mm^2 , but it is well below the yield stress of the shaft material. There can be some minor peaks or resonances in the system but their effect is negligible in the whole torsional vibration point of view. The location of major and minor resonance peaks in the torsional stress graph of the intermediate shaft are shown in Figure 56.

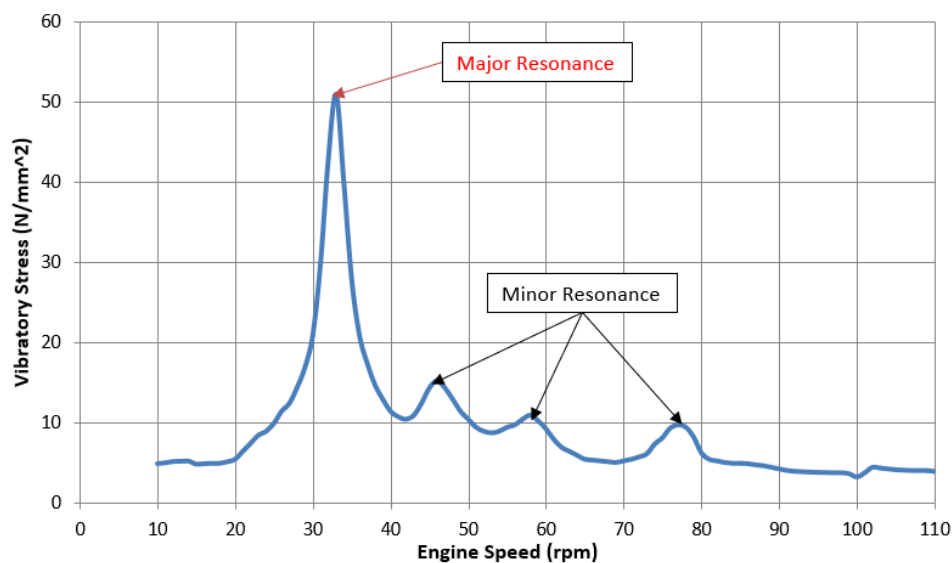


Figure 56: Occurrence of Different Resonances in Intermediate Shaft

In the mis-firing conditions (analysed when the cylinder 6 is not firing), the intermediate shaft and propeller shafts are showing higher stresses that exceeds the limit curves of continuous operation. This is occurring due to the 1st mode 3rd firing order resonance at a speed of 77 rpm. Hence it is advisable to maintain the maximum speed of the main engine to be limited to around 73 rpm to safeguard the engine and other components in the propulsion system.

6.2 TVC Analysis with Ice Condition – Review of Result

The Torsional Vibration Calculations (TVC) analysis for the ice condition is performed numerically in order to obtain the shaft torsional response due to the time varying loads (oscillating engine torque and the non-uniform ice torques) acting on the propulsion system. After conducting the time-domain TVC analysis of the marine power transmission system of the ice class 3000TEU Container vessel, the obtained results are summarised in the following Table 16.

Table 16: Summary of TVC Ice Case Results

Maximum Torque Occurring in the Shafting					
MCR (88 rpm)		Ice Bollard (75 rpm)		1 st Blade Order (46 rpm)	
Engine Torque (2355 kNm)		Engine Torque (1701 kNm)		Engine Torque (652 kNm)	
Load Case	Maximum Moment (kNm)	Load Case	Maximum Moment (kNm)	Load Case	Maximum Moment (kNm)
A1	2266	B1	1692	C1	958
A2	2408	B2	1867	C2	973
A3	2569	B3	2020	C3	1203
A4	2382	B4	1805	C4	983
A5	2382	B5	1805	C5	983

From the results, it is evident that a maximum torque of 2569 kNm can be expected in the propulsion shafting system during the load case A3 when the engine is operating at MCR condition. Hence it must be ensured that the design of the shafting system assembly with the new propeller (intended to operate in ice channel) is able to transmit this torque safely and also to prevent the causing of high shear stresses due to torsional oscillation leading to fatigue failure

of the shafts during continuous operation of the vessel. Then analysing the three engine operation points, the maximum moment is occurring for the third ice excitation load cases in every scenario. This shows the double impact of a single propeller blade on two ice blocks. The impacts can be smaller, but as a whole the double sequence of milling causes high stresses on the blades and the following hub and shaft lines.

6.3 Strength Pyramid of the Propulsion System

During the design stage of propulsion shafting in the ship building process, the strength of the propulsion line has to be designed based on the strength pyramid principle. The principle states that under any circumstances the loss or damage of the propeller blades shall not cause any major damage to the propeller and other intermediate shaft components. In other words, a strength pyramid chart shows the order of occurrences of the damage starting from the blades of the propeller to the shaft line. From the conducted TVC analysis, it is clear that all the vibratory stresses developing due to the torsional vibration of the propulsion shaft line is under the maximum permissible limits that a shaft can withstand. Hence, we can say that the propulsion system installed in the investigated Container vessel follows the design principle of pyramid strength which ensure that the damage of the shafting is not possible at the earlier stages of the continuous operation of the vessel. An arrangement of the main components in a typical strength pyramid is shown in Figure 57. The order of damage or loss of the parts should begin from top to bottom in the strength pyramid of a propulsion machinery.

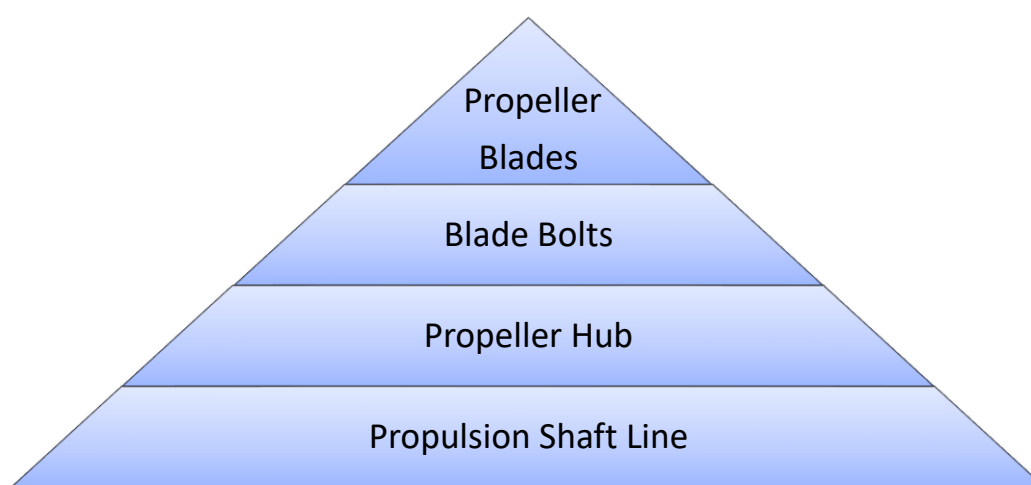


Figure 57: Strength Pyramid of the Ship Propulsion System

6.4 Possible Counter Measures for High Vibratory Stresses

There are some possible ways for treating the torsional vibration problems and minimizing the effects on the propulsion system of a ship in both design phase and operation stages. The primary aim should be to restrict the vibration levels well below the corresponding admissible limits. The methods used in the operation phase of the marine propulsion shafting are often expensive. To find a cost-effective installation of the shafting system, the vibration aspects have to be considered at the earliest stages of the design process with the best possible counter measures. Some of the feasible measures are discussed below:

- Torsional vibration can be restricted to an extent by the use of anti-vibration mountings and also by other flexible mounting systems. The anti-vibration mountings are fixed beneath a machine preventing the surrounding structure from the vibrating equipments.
- Viscous dampers are a great option to limit the overall vibrations and crankshaft stresses. These are primarily used to control engine vibrations and so they are located usually close to the anti-node of the crankshaft to give more effectiveness (Troy, F. P. and Charles, H., 2002). But this solution is very costly since a large torsional vibration damper itself costs a lot.
- Shaft material properties is another important parameter to play with to reduce the vibratory stresses. In place of widely used standard material like Carbon steel, the use of higher tensile strength grade steel can highly influence the downturn of vibratory stresses in the shafts.
- Selection of appropriate shaft dimensions can be used to keep vibration responses within the safe limits. Changing of the shaft diameters are done very often since it is the cheapest counter measure among all other options.
- Balancing of the system can control torsional vibration to a great extent, which means to add up an extra mass (usually tuning wheel) at the forward part of the propulsion line (mostly near to the free end of crankshaft) to steady out the unbalancing forces.

7. CONCLUSION

Through this thesis work, a new FEM based calculation procedure was developed for conducting the Torsional Vibration Calculations (TVC) for the marine power propulsion system of a Container vessel. It shows that by generating a proper 3D model of the shafting system at the early stage of the design phase, a good and effective finite element simulation can be conducted to analyse the torsional stresses developing in the structure with the help of any FEM software that are currently in use by industries. This will enable the design offices and shipyards to get an initial idea about the safety of the shafting line while designing and installing on board. The choice of selection of other related components like the sufficient number of blades for a propeller, the type of propeller, etc. can also be designed and modify according to this vibration analysis. For this thesis, the investigation was done for a system having slow speed diesel engine directly coupled with the propeller shaft which then drives the propeller using the main engine power.

The torsional vibration study has a very significant role in ship machinery design as these vibrations can cause small part cracks or excessive deflections in components which will result to the danger of fatigue failures when the system is vibrating at critical speeds near the natural frequency of the structure. The main constraints faced during the conducted torsional analysis is the lack of theoretical information regarding the analytical procedure for vibratory stress determination, precisely adding the effect of engine harmonics data in correlation with the torque excitation of the components. Another challenging constraint is the idealization of the numerical model of the structure as the model should exhibit the exact mechanical properties of the original system. Among the techniques used in this thesis, FEM analysis came to be the most convenient and effective way to analyse the system in a global manner by discretizing the structure and calculating the natural frequencies, mode shapes and torsional stresses.

For this thesis, the work is mainly divided into tasks to identify the major differences between the TVC of the propulsion system for a vessel operating in normal open water and the one navigating through ice conditions. After assessing the first scenario (open water conditions), the torsional stresses developing in each shaft segments have been calculated with respect to the engine speed. This was done in frequency domain approach as the analysis only focus on the oscillating engine forces that are harmonic in behaviour. The obtained results of vibratory

stresses in the shaft line are compared later with the permissible limits provided by the DNV rules and came to be in the safe side for the whole operating speed range. This whole analysis was done initially to get an overview about the critical natural frequencies of the system and also the barred speed range that should be avoided by the vessels in continuous operation.

In the second part of this thesis work the TVC analysis of the dynamic response of the propulsion system in ice is investigated. The whole process is formulated in a complete FEM manner, where all the external applied torques are calculated manually as a prerequisite. The FEM model has undergone a transient analysis subjected to three different moments acting on the propulsion line. As these excitation loads are varying according to time and also have a non-harmonic nature, the time-domain approach is the best suitable one and also been suggested by the DNV and Finnish-Swedish Ice Class Rules. Nevertheless, the torsional vibration in the time domain analysis requires much more computational time compared with the frequency domain calculations. After assessing this method, it is seen that the engine speed (conducted for three operating points) at which the ice impact is applied have a great influence on the torsional response. Among the conducted tests, the Maximum Continuous Rating (MCR) speed of the engine exhibits the maximum torque on the shaft line. Hence it is advisable to operate the vessel well below the MCR speed to safe guard all the propulsion related components.

Furthermore, the ice class simulation highlights the need to consider the influence of angular velocity of propulsion unit with the propeller design loads. In conclusion, the present study forms a basis for the detailed evaluation of the torque response due to engine design loads, ice excitations and hydrodynamic loads for ice going vessels.

8. FUTURE DEVELOPMENTS

The possible recommendations that can be carried out for the future developments in the torsional vibration calculation methods are:

- ❖ This thesis was based on the propulsion system of a directly coupled engine-propeller model. For a more detailed approach, the simulation shall be extended to include medium speed engines that are connected to the propeller shaft by a gear drive or electric motor arrangement.
- ❖ Also, the simulation stages can be extended further to identify the validity of the ice excitation loads that are actually based on rules by classification societies and also the reliability of the engine response.
- ❖ The dependency of the power transmission system on the hydrodynamic loads and ice contact loads are still not very realistic and it can be investigated further.
- ❖ Moreover, the damping methods that can be provided for the propeller during propeller-ice interaction has to be clearly identified, as only water damping is considered for the normal TVC calculation.

9. ACKNOWLEDGMENTS

First of all, I would like to express my sincere gratitude to my thesis supervisor Dipl.-Ing. Jörn Klüss, Head of Design Department and to Dipl.-Ing. Abinav Viswanath Subramanian of Mecklenburger Metallguss GmbH (MMG), for their continuous support, professional and insightful assistance, and giving me a complete guidance throughout this thesis work by conducting meetings with external specialists, providing sufficient data and information, arranging all the facilities needed which made life easier. I was able to learn a lot from this short period of internship and I once again extend my gratitude to the entire team of MMG for giving me this opportunity to carry out and complete my master's thesis in the company, despite of the global pandemic situation.

I would also like to convey my acknowledgement to Dr.-Ing. Patrick Kaeding, Dr.-Ing. Thomas Lindemann and Dipl.-Ing. Gunnar Kistner, my supervisors in the University of Rostock, Germany for helping and guiding me through every stage of the internship by giving valuable insights and knowledge about the thesis work and also other curriculum related matters. I would also like to specially mention and express my gratitude for conducting online sessions throughout the internship period to assess the progress and development of the work.

Also, I convey my profound gratitude to Prof. Philippe Rigo (University of Liege), Christine Reynders and all the people who are working behind the EMship+ M120 program. I thank you for giving me the opportunity to join this program and the technical knowledge that I gained in this academic path will be a great asset in my career. I would also like to thank my course mates Mr. Sooraj Revikumar Pillai Anila and Mr. Abdul Fatah Karyaparamban for their assistance and valuable help in this master's program.

Finally, I express my special thanks to my parents and siblings for motivating and providing me with continuous support and morale throughout this program. I also convey my gratitude to all the professors who taught us about Naval Architecture and Ship Structures during this course period and a special mention to all my beloved batch mates in the EMship+ M120 3rd Cohort for the encouragement and appreciation showed in these times.

Kevin Lal

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APPENDIX A1 - FORCED TORSIONAL VIBRATION CALCULATION METHOD

- Torsional stress calculation (open water condition) method for the critical speed (33 rpm) for the first order torsional natural frequency (24.19 Hz) and 7th firing order.
- Sin and Cos components of excitation force are obtained from gas harmonics data (T237600).

System No.	Components	Mass Moment of Inertia, J	Torsional Stiffness, k	Absolute Damping	Relative Damping	Crank Angle	External Moment		Factor for Excitation	Torsional Stress Calculation				
							Sin- Component	Cos- Component		Relative Amplitude, ϕ	Torque, T	Shaft Diameter, d		Shear Stress, τ
		(kgm ²)	(MNm)	Damping Coeff. V or Archer Factor	Non Dimensional Damping Factor, $\tan \epsilon$		(Nm)	(Nm)		(mrad)	(kNm)	Outer (mm)	Inner (mm)	(N/mm ²)
1	MOMENT COMPENSATOR	5995,00	3759,0		0,010		0	0		13,759	48,3	896,0	150	0,34
2	CYLINDER 1	30557,00	2865,0	58,8	0,010	0,00	-20591	35912	100	13,746	282,1	896,0	247	2,01
3	CYLINDER 2	31231,00	2865,0	58,8	0,010	102,86	-20604	35905	100	13,649	520,4	896,0	150	3,69
4	CYLINDER 3	30557,00	2865,0	58,8	0,010	257,14	-20579	35919	100	13,472	750,3	896,0	247	5,34
5	CYLINDER 4	30666,00	2865,0	58,8	0,010	205,71	-20572	35922	100	13,216	976,4	896,0	226	6,94
6	CYLINDER 5	31231,00	2865,0	58,8	0,010	154,28	-20566	35926	100	12,885	1200,6	896,0	150	8,51
7	CYLINDER 6	30557,00	2865,0	58,8	0,010	308,57	-20585	35915	100	12,478	1412,3	896,0	247	10,06
8	CYLINDER 7	31231,00	3477,0	58,8	0,010	51,43	-20598	35908	100	12,000	1619,7	896,0	150	11,48
9	THRUST BEARING	13180,00	4525,0		0,010		0	0		11,551	1705,5	896,0	150	12,08
10	FLYWHEEL	15889,10	329,0	100,0	0,010		0	0		11,186	1805,7	570,0		49,66
11	GENERATOR	3589,80	1155,9		0,010		0	0		6,014	1816,9	570,0		49,97
12	SHAFT FLANGE 1	515,00	107,2				0	0		4,665	1818,0	570,0		50,00
13	SHAFT FLANGE 2	1262,80	151,4				0	0		13,471	1808,3	715,0		25,20
14	PROPELLER	121508,00		9,1			0	0		25,280	0,0			

Figure 58: Material Data, Damping Coefficients and Excitation Moments for TVC Calculation

System No.	1. Without External Force			
	Sin Component		Cos Component	
	Relative Amplitude, ϕ	Torque, T	Relative Amplitude, ϕ	Torque, T
	(rad)	(Nm)	(rad)	(Nm)
	1,0000	0,0000	0,0000	0,0000
1	1,0000	3508087	0,0000	0
2	0,9991	21372409	0,0000	-303648
3	0,9916	39494524	0,0002	-608376
4	0,9778	56979219	0,0005	-896075
5	0,9579	74169646	0,0011	-1169552
6	0,9321	91204007	0,0017	-1427833
7	0,9002	107301920	0,0025	-1656260
8	0,8628	123070914	0,0035	-1860680
9	0,8274	129452312	0,0044	-1826923
10	0,7988	136879924	0,0051	-1854086
11	0,3829	137684165	0,0149	-1822867
12	0,2638	137763654	0,0176	-1817554
13	-1,0213	137008934	0,0346	-1791998
14	-1,9263	407457	0,0464	16559633
	M Rest (c)=	407457	M Rest (d)=	16559633

2. With External Force			
Sin Component		Cos Component	
Relative Amplitude, ϕ	Torque, T	Relative Amplitude, ϕ	Torque, T
(rad)	(Nm)	(rad)	(Nm)
0,0000	0,0000	0,0000	0,0000
0,0000	0	0,0000	0
0,0000	-20594	0,0000	59793
0,0000	-41088	0,0000	119194
0,0000	-61307	-0,0001	177867
0,0000	-81160	-0,0001	235416
0,0001	-100512	-0,0002	291411
0,0001	-119344	-0,0003	345643
0,0001	-137456	-0,0004	397520
0,0002	-136052	-0,0005	393315
0,0002	-134146	-0,0006	387415
0,0006	-132871	-0,0018	383605
0,0007	-132654	-0,0021	382958
0,0020	-131209	-0,0057	378731
0,0028	5274	-0,0082	-227849
M Rest (g)=	5274	M Rest (h)=	-227849

3. Combined Effect				Final Result	
Sin Component		Cos Component		Relative Amplitude, ϕ	Torque, T
Relative Amplitude, ϕ	Torque, T	Relative Amplitude, ϕ	Torque, T	(mrad)	(kNm)
(rad)	(Nm)	(rad)	(Nm)		
0,0137	0,0000	0,0007	0,0000		
0,0137	48212	0,0007	2304	13,7588	48,27
0,0137	273329	0,0007	69654	13,7460	282,06
0,0136	502090	0,0006	136767	13,6494	520,38
0,0135	722356	0,0006	202967	13,4718	750,33
0,0132	938933	0,0005	268046	13,2163	976,44
0,0129	1153855	0,0004	331677	12,8846	1200,58
0,0125	1356410	0,0003	393340	12,4777	1412,29
0,0120	1555147	0,0002	452762	12,0003	1619,71
0,0116	1644230	0,0001	453211	11,5505	1705,55
0,0112	1748232	0,0000	451815	11,1861	1805,67
0,0059	1760540	-0,0014	448962	6,0145	1816,88
0,0043	1761845	-0,0017	448440	4,6654	1818,02
-0,0121	1752902	-0,0059	444069	13,4708	1808,28
-0,0237	0	-0,0088	0	25,2797	0,00
M Rest=	0,0	M Rest=	0,0		

Determining Initial Sin and Cos values for Combined Effect		
Free Vibration	Forced Vibration	Initial Values
407457 [=c]	5274 [=g]	0,01374
16559633 [=d]	-227849 [=h]	0,00066

Figure 59: Effect of Free and Forced Excitation in TVC

APPENDIX A2 – PERMISSIBLE TORSIONAL VIBRATION STRESS LIMITS

• Crankshaft

Technical data for limit curve

GL Rules 2008, I - Part 1; Chapter 2: Section 16, Pg:16-1

Type of Shaft :	Crank shaft		
Tensile Strength of shaft material	R _m	600	N/mm ²
Material Factor	C _w	42,222222	-
Nominal Speed	n ₀	88	rpm
Speed	n	10	rpm
Speed Ratio	$\lambda (n/n_0)$	0,1136364	
Shaft Diameter	d	820	mm
Size Factor	c _D	0,5930637	-
Form Factor	c _K	1	-
Torsional Vibration Stress	τ_1	27	N/mm ²

(For all intermediate shafts made of forged, low alloy carbon steel, R_m = 600 N/mm²)

(Form factor of intermediate shaft with Flanges = 1.0)

(Continuous Operation for Two Stroke Engine)

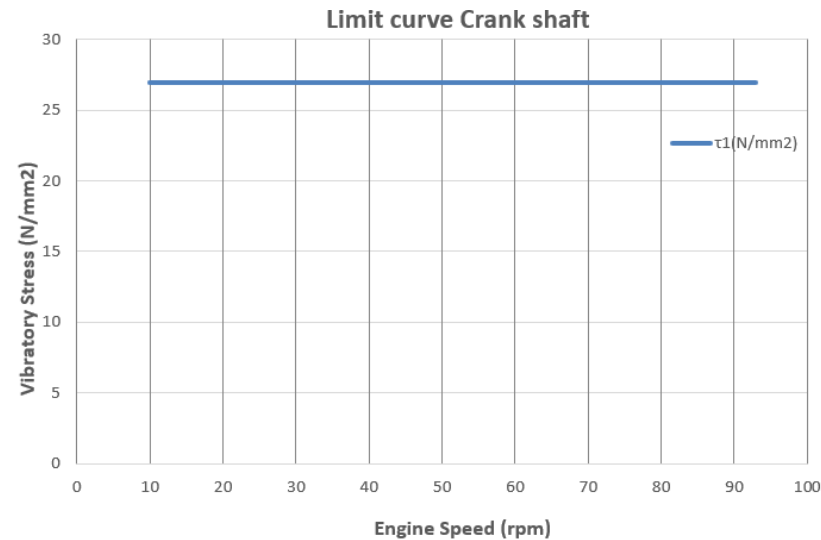


Figure 60: Limit Curve for Crankshaft

- Intermediate Shaft**

Technical data for limit curves

GL Rules 2008, I - Part 1; Chapter 2: Section 16, Pg:16-1

Type of Shaft :	Intermediate Shaft			
Tensile Strength of shaft material	R _m	600	N/mm ²	(For all intermediate shafts made of forged, low alloy carbon steel, R _m = 600 N/mm ²)
Material Factor	C _w	42,2222222		
Nominal Engine Speed	n ₀	88	rpm	
Engine Speed	n	10	rpm	
Speed Ratio	λ (n/n ₀)	0,11363636		
Diameter of the shaft	d	570	mm	
Size Factor	c _D	0,61140144		
Form Factor	c _k	1		(Form factor of intermediate shaft with Flanges = 1.0)
Torsional Vibration Stress	τ ₁ (λ<0.9)	76,7774799	N/mm ²	
Torsional Vibration Stress	τ ₁ (0.9 ≤ λ ≤ 1.05)	35,624324	N/mm ²	
Torsional vibration Stress based on Speed ratio, τ ₁		76,77747986	N/mm ²	(Continuous Operation)
Torsional Vibrational Stress	τ ₂	130,5217158	N/mm ²	(Transient operation)

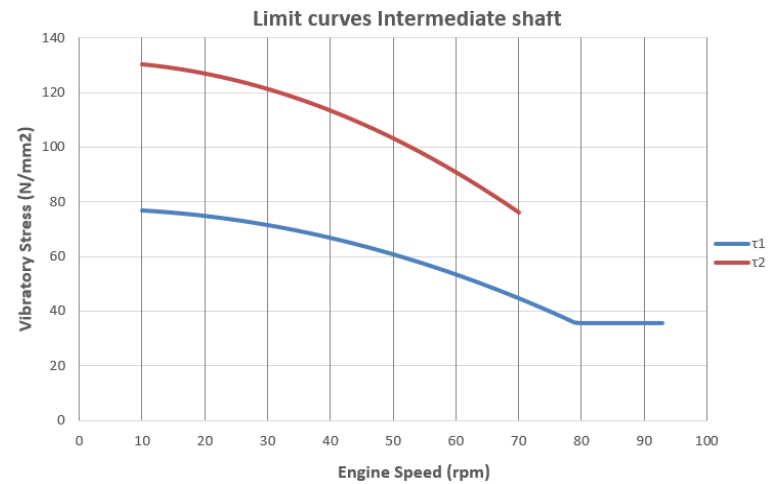


Figure 61: Limit Curve for Intermediate Shaft

- Propeller Shaft**

Technical data for limit curves

GL Rules 2008, I - Part 1; Chapter 2: Section 16, Pg:16-1

Type of Shaft :	Propeller Shaft			
Tensile Strength of shaft material	R _m	600	N/mm ²	(For propeller shafts in general, R _m = 600 N/mm ²)
Material Factor	C _w	42,2222222		
Nominal Engine Speed	n ₀	88	rpm	
Engine Speed	n	10	rpm	
Speed Ratio	λ (n/n ₀)	0,11363636		
Diameter of the shaft	d	715	mm	
Size Factor	c _D	0,59981686		
Form Factor	c _K	0,55		(Form factor of propeller shaft with keyless propeller fit within the aft = 0.55)
Torsional Vibration Stress	τ ₁ (λ<0.9)	41,4275024	N/mm ²	
Torsional Vibration Stress	τ ₁ (0.9 ≤ λ ≤ 1.05)	19,2221309	N/mm ²	
Torsional vibration Stress based on Speed ratio, τ ₁		41,4275024	N/mm ²	(Continuous Operation)
Torsional Vibrational Stress	τ ₂	94,96341582	N/mm ²	(Transient operation)

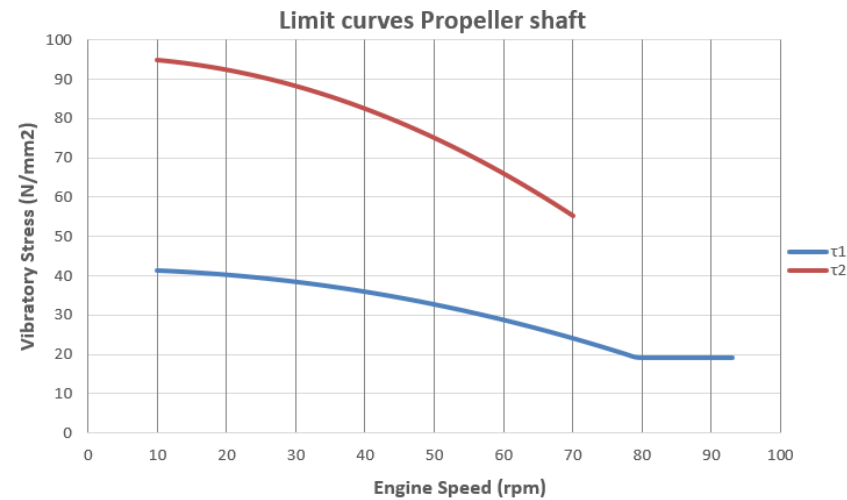


Figure 62: Limit Curve for Propeller Shaft