Master thesis : Study on the Vibration of Corrugated Cargo Tank Bulkheads

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Study on the Vibration of Corrugated Cargo Tank Bulkheads

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by

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Master Thesis

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ABSTRACT

Cargo tankers corrugated bulkheads are constantly subjected to excited vibration due to the propulsion system components, such as engines, shafts and gear boxes, and severe vibration can be observed if the bulkheads vibrate in resonance with the excitation frequency. Added to it, the influence of the cargo density and the filling level at the cargo tank can lead to a considerable variation of the natural frequencies, shifting them closer to the excitation frequency and leading the system to a dangerous resonance state.

In addition, vibration induced by the propulsion system is known for an extremely high number of load cycles in a short period of time. In a state of resonance, the effects of such cycles combined with huge amplitudes may negatively affect the design fatigue life, leading to fatigue induced cracks.

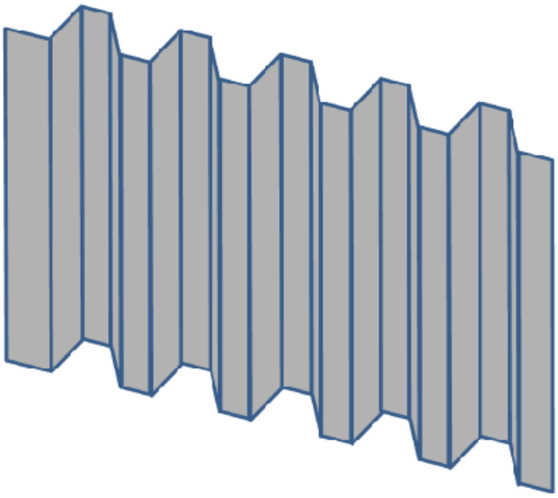
In the course of this study, two main questions are supposed to be answered. First, how to design a proper corrugated cargo tank bulkhead in order to minimise resonance risks due to the excitation frequencies generated by the propulsion system? And second, to evaluate the vibration level and the stress range at which fatigue damage is likely to occur. For that purpose, parametric Finite Elements models are developed in order to analyze how the natural frequencies vary according to the variation of dimension parameters and cargo filling levels. In addition, the model is refined at the stress concentration areas for a fatigue strength analysis in order to determine the vibration levels and the stress range for which the fatigue damage is likely to occur and finally to evaluate the fatigue life of the bulkhead in the considered conditions.

Keywords: Vibration, Corrugated Bulkheads, Fatigue

1 CHAPTER 1 - INTRODUCTION

1.1 Background and Motivation

Corrugated bulkheads are generally used by small or medium sized ships as the boundaries between adjacent cargo tanks. Usually they are seen more frequently in product or chemical tankers (**TSCF**, 2011). The corrugated shape, as seen in Figure 1a represents a preferred structural solution, when compared to a flat bulkhead, because it generally presents a lower mass (**Ko**, **et al**, 2016) than stiffened flat plates, since the corrugations eliminate the necessity of the use of stiffeners. In addition, they provide easier maintenance and cleaning, which is desirable for tankers, and also reduces corrosion problems occurrence (**TSCF**, 2011).



(a) General shape of a corrugated bulkhead (**Ko**, **et al**, 2016).

(b) Corrugated bulkhead in a ship (**Özgüç**, 2017)

Figure 1: Corrugated bulkheads examples.

As any other ship structural component, the bulkheads are also subject to vibration in a natural frequency, which is a function of its geometry, material properties and boundary conditions. Vibration of ship parts, in extreme cases, can impair well-being, efficiency and health condition of people on board, lead to damages to the ship structure and its cargo, and, it can even poses a risk to the safety of the vessel (**Germanischer Loyd**, 2001).

Excited vibration due to the propulsion system components, such as propellers, engines, shafts, gear boxes and other equipment, with an excitation frequency similar to the structure´s natural frequency, may lead to a state of resonance and, consequently, increase risk of fatigue damage of the ship structure. The resonance phenomenon is known by an extremely high response amplitude to an external excitation, which can lead to appearance of cracks, that can propagate and ultimately cause structural failure (**Rao**, 2018). In the case of tankers bulkheads, this condition is even more worrisome, because fluid contamination, loss of cargo or even accidents can be caused depending on the cargo material and on the extent of the damage.

In addition, vibration induced by the propulsion system is characterized for extremely high number of load cycles in a short period of time, which, in a state of resonance, is combined with severe high amplitudes. The effects of such load conditions may negatively affect the design fatigue life, leading to fatigue induced cracks occurrence and a decrease in the structure lifespan.

Therefore, proper vibration analysis during structural design stage are a key step in ship design, once the bulkheads geometry must guarantee that the natural frequencies are far enough from the propulsion´s excitation frequency, thus minimizing the risks of resonance occurrence and maintaining a proper fatigue life.

1.2 Objectives and Aims

Considering the resonance problems introduced in the previous section, the main objectives for this thesis are:

- Develop a parametric Finite Elements model of a section of a typical cargo tanker ship, which allows a proper vibration parametric analysis, but also a reliable stress assessment for fatigue evaluation;
- Calculate the bulkheads natural frequencies considering tank filling levels and design parameters variation, through a parametric finite elements simulation in ANSYS Workbench.
- Discuss of the impact of the design parameters variation on natural vibration characteristics, compare the results with full scale measurements and, if possible, select new geometries which satisfactorily move the bulkheads' natural frequencies away from the excitation frequency;
- Determine the corresponding stress range at the boundaries, through an analysis with a refined FE-model, for which fatigue damage is likely to occur and evaluate the fatigue life of the bulkheads under the considered conditions.

1.3 Report Structure

The present executive summary is divided in three main chapters:

- **Chapter 1 - Introduction**: This present chapter sets a background and discuss the motivations behind the conducted study, as well establishing the objectives and goals.
- **Chapter 2 - Vibration Analysis**: This chapter starts with a brief overview about vibration and resonance theory for later proceed to the description of the parametric

modelling features and procedures, materials, variables and mesh parameters. Following, the details of the simulation, such as boundary conditions and parameters range are explained. Finally, the model is validated by comparing it to the measurements results and the results are presented and discussed.

- **Chapter 3 - Fatigue Evaluation**: Chapter 3 begins with a discussion about the different approaches used for fatigue assessment and the selection of the applied approach. Then the concepts behind S-N curves and FAT classes are explained, culminating with the determination of the appropriate parameters. The mesh is refined in the surroundings of the welds and the nominal stress range is determined, then allowing the estimation of the fatigue life of the structure.
- **Chapter 4 - Conclusion**: Further discussion over the procedures and results are made in Chapter 4, leading to a conclusion over the work and recommendations for further developments.

Throughout the next chapters, the work steps are discussed and the obtained results are presented.

2 CHAPTER 2 - VIBRATION ANALYSIS

2.1 Theory review

2.1.1 Free and Forced Vibration

Any movement that is repeated after an interval of time is called vibration, and it involves the alternate transformation of potential energy into kinetic energy and the other way around (**Rao**, 2018). Examples of oscillatory system can be a pendulum or a mass-string system, both seen in Figures 2 and 3.

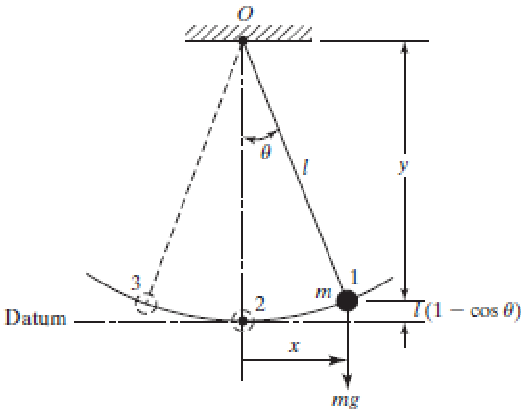


Figure 2: Pendulum (**Rao**, 2018).

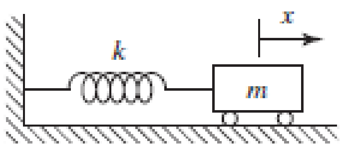


Figure 3: Mass-string system (**Rao**, 2018).

A system that oscillates only under initial disturbance without external force application, such as a pendulum for example, is said to undergo free vibration. To describe the movement of a mass-string system undergoing free vibration, with a mass m, connected to a string with constant k and displaced by a distance x, the Newton's 2^{nd} Law of motion gives the following equation (**Rao**, 2018).

$$F = m\frac{d^2x}{dt^2} = m\ddot{x} \tag{1}$$

The force F can be due to weight elevation, as for a pendulum, or due to a spring compression as in this example. If the mass is translated by a distance x from its initial position, the force generated by the spring is kx. Then, with the application of the

D'Alambert's Principle for undamped vibration, the equation of motion can be written as equation below (**Rao**, 2018).

$$m\ddot{x} + kx = 0 \tag{2}$$

In this context, when a system is left to freely vibrate after a disturbance, without external interference, it vibrates in a frequency known as natural frequency (ω_n), which can be obtain as described in equation 3 (**Rao**, 2018).

$$\omega_n = \sqrt{\frac{k}{m}} \tag{3}$$

Sometimes, the oscillatory movement is not only generated by an initial disturbance, but for the application of an external force. If a force F_e acts on the system, the Newton's 2^{nd} Law gets a new term, therefore the movement can now be described by the following equation (**Rao**, 2018).

$$m\ddot{x} + kx = F_e \tag{4}$$

The applied force can be either a scalar or an harmonic force, which also apply loads to the system with a specific frequency. It is in the case of harmonic excitation forces that the system can experience a worrisome phenomenon called resonance.

2.1.2 Resonance

Harmonic excitation forces can be described by sinusoidal, cosine or even exponential functions, as seen below, where F_0 is the amplitude of the function, ϕ is the phase angle and ω is the frequency of the harmonic excitation (**Rao**, 2018).

$$F_e = F_0 sin(\omega t + \phi) \tag{5}$$

$$F_e = F_0 cos(\omega t + \phi) \tag{6}$$

$$F_e = F_0 e^{i(\omega t + \phi)} \tag{7}$$

The excitation frequency ω is independent from the system characteristics and, when the natural frequency of vibration of a structure has a value similar to the frequency of the external excitation frequency, the phenomenon known as resonance can occur (**Rao**, 2018). As an example, if a system is submitted to a cosine excitation force, the equation of motion is described by the free vibration equation in addition to a cosine function.

$$m\ddot{x} + kx = F_0 cos(\omega t) \tag{8}$$

The solution is split between homogeneous and particular solutions, which are the solutions of equation 2 and function F_e respectively. The point of interest is the particular solution, which assume the form described below.

$$x_p(t) = Xcos(\omega t) \tag{9}$$

$$X = \frac{F_0}{k - m\omega^2} = \frac{\delta_{st}}{1 - (\frac{\omega}{\omega_n})^2} \tag{10}$$

Where X is the amplitude response and $\delta_{st} = F_0/k$, known as static deflection, denotes the deflection of the mass m when submitted to a force F_0 (**Rao**, 2018). When excitation and natural frequencies coincide their values, the term ω/ω_n tends to 1, leading the amplitude X to infinite values and, consequently, increasing indefinitely the system response to the excitation.

In Figures 4 and 5, one can observe in different ways how the phenomenon of resonance works. First, the magnification factor is a way to express the response amplitude, by relating the dynamic and the static amplitude and it is described as the following expression.

$$\frac{X}{\delta_{st}} = \frac{1}{1 - (\frac{\omega}{\omega_n})^2} \tag{11}$$

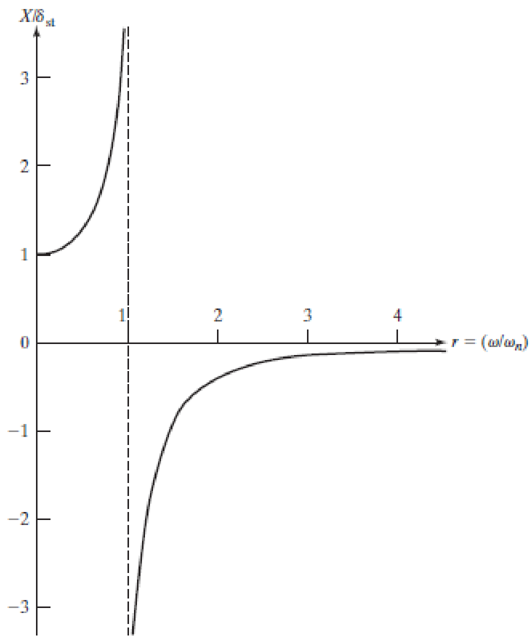


Figure 4: Magnification factor (**Rao**, 2018).

As seen in Figure 4 the magnification factor presents a clear asymptotic behavior when the excitation frequency and the natural frequency coincide. This behavior leads to a system response in resonance that increases after each cycle, as it is shown in Figure 5.

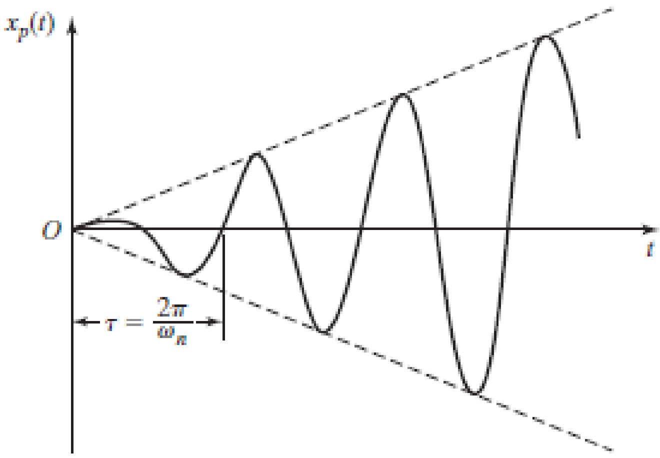


Figure 5: Resonance response (**Rao**, 2018).

Throughout the history there are many cases of structures that suffered the effects of excessive vibration and some even came to failure due to resonance. Perhaps the best known case is the Tacoma Narrows bridge, seen in Figure 6, which was built in the state of Washington in the United States and collapsed in 1940 due to excessive vibration as a result of severe winds (**ASCE**, 2021).

Figure 6: The collapse of Tacoma Narrows bridge in 1940 (**ASCE**, 2021).

2.2 Parametric Modeling

A parametric model of each studied bulkhead was developed in order to ensure an automatic update of the design parameters and provide a solid database for the vibration analysis. The modeling work was done in Design Modeler environment of ANSYS Workbench software, due to its automatic processing and better graphics displays.

The bulkheads were modeled as shell elements by extrusion and divided in three parts, following the thickness patterns discussed in section **??**. The profile dimensions of corrugation depth, width, top flange and bottom flange were set as simulation parameters, as well as the bulkheads plating thicknesses.

In the next subsections, the modeling steps are discussed.

2.2.1 Surrounding Ship Structure

In order to simulate the effects of the surrounding structure on the bulkheads natural frequencies, some structural elements of the ship, such as hull plating, deck plating, double bottom, double side and floor plates were modeled and connected to the bulkheads edges.

The surrounding structure serves as boundary conditions for the bulkheads during the simulation, but it is also important for the stress assessment, which will be relevant for the future fatigue evaluations.

The dimensions and thicknesses of these plating elements were set following technical drawings of typical ships, while the extent of the hull body was defined as mid length of the cargo tank length, both aft and forewards.

2.2.2 Hydrodynamic Mass

When a body is accelerated in a fluid field, the inertia of the fluid opposes the movement, generating an equivalent virtual mass added to the body´s actual mass (**Lopresto**, **et al**, 2017). Therefore, the expression that governs an excited vibration motion, would look like the following equation.

$$([M_b + M_a])\ddot{u} + [C]\dot{u} + [K]u = F_e \tag{12}$$

Where

- $[M_b]$ is the body´s own mass matrix;
- $[M_a]$ is the added mass due to the surrounding fluid movement;
- $[C]$ is the damping matrix;
- $[K]$ is the striffness matrix;
- F_e is the excitation force;
- u is the studied degree of freedom;

The hydrodynamic mass represents a key factor in the design of structures that operates totally or partially submerged in fluids, such as pipes, risers, anchor chains, pontoons, ships and, as expected, tankers bulkheads. Usually, the effects of the hydrodynamic mass decrease considerably the natural frequencies of the submerged structure, while the mode shapes remain not far from the same when compared to a no-fluid situation (**Presas, et al**, 2017).

In order to simulate the effects of the hydrodynamic mass on ANSYS, a Modal Acoustic analysis is performed, instead of a simple Modal Analysis. A solid following the bulkhead shape and touching its surface is modeled at each side of the bulkhead to simulate the fluid, while the extent of the vertical extrusion is also set as a parameter, once it represents the fluid level.

The fluid is, then, assigned as an acoustic body, sea water is defined as the material and the solid-fluid interface is placed in the connections between the faces of the bulkhead and the fluid.

2.2.3 Material Properties

The assigned material to the ship structure, including hull and bulkheads is ANSYS default structural steel, which is an isotropic elastic material and its main properties are summarized in Table 1. Regarding the fluid, as mentioned before, is defined as sea water, which relevant properties are shown in Table 2 below.

Table 1: Structural steel properties.

Density [kg/m^3]	7850
Young´s Modulus [kPa]	2,00e+8
Shear Modulus [kPa]	7,69e+7
Poisson´s ratio	0,3

Table 2: Sea water properties.

Density [kg/m^3]	1025
Speed of Sound [m/s]	1482,1
Viscosity [N.s/m^2]	0,001003

2.2.4 Mesh

The mesh of finite elements was divided in two regions. The first region is made up by the ship structure and the fluid body. Since the natural frequencies and the mode shapes of these elements are not the main interest of this study, a quadrilateral element sized 0,2 m is assigned to them. The bulkhead vibration, however, is the main object of the analysis, so they represent the second region, with a finer mesh of 0,1 m elements.

2.3 Simulation

Once the models of both bulkheads are ready, the structural and acoustic regions are defined and the mesh is set, one can proceed with the simulation. However, some specifications must be done in advance, such as the boundary conditions and the range of variation of the design parameters, which will lead to the definition of the design points.

2.3.1 Boundary Conditions

Since the focus of the study is the modal analysis of the bulkhead, some displacement constraints have to be set to the hull structure.

For example, the plates in the deck, bottom and double bottom have fixed degree of freedom regarding displacement in Z-axis, while the side, double side and hopper tank plating are fixed in Y direction. Finally, the floor plates under the bulkheads have the displacement in X-axis constrained. That way, the components of the surrounding ship structure are prevented from vibrating, allowing the analysis to focus solely on the bulkheads movements.

In addition, the free edges of the hull structure are assigned with simply supported boundary condition, in order to simulate their continuity with the rest of the ship structure.

Regarding the bulkheads, the only free boundary around their structure, is their inner part. In a real ship, this region represents the crossing between longitudinal and transversal bulkheads and, to simulate this kind of boundary constraint, it was applied a simply supported condition as well. Thus, the displacement in this edge is fixed, but the rotation is still free. The other edges are already constrained by the connections between bulkheads and hull structure.

Finally, a pressure of 0 Pa is applied in the free surface of the fluid body to set a boundary condition to the whole body of fluid.

2.3.2 Parameters Range

In order to summarize and simplify the comprehension throughout the rest of the report, the group of independent design parameters receive a letter code each one and they are explained below.

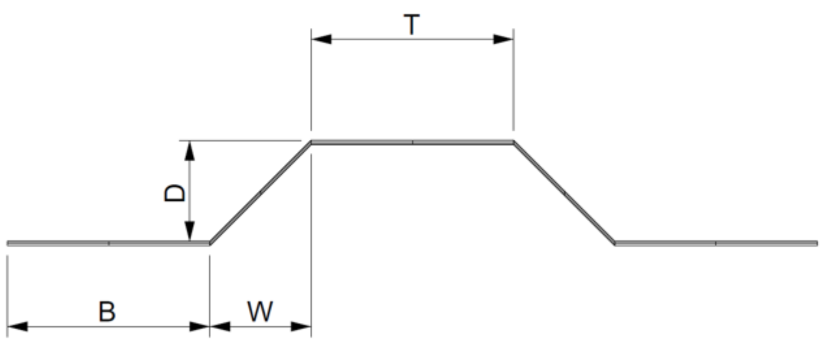


Figure 7: Corrugation parameters.

- T - Length of corrugation top flange;

- B - Length of corrugation bottom flange;
- D - Corrugation depth;
- W - Corrugation width;
- $U1$ - Upper bulkhead thickness;
- F - Fluid level;

Since there are normally different thicknesses in the same bulkhead, it was decided that the pattern between the thicknesses in a bulkhead would be kept the same through the simulation. Only the thickness of the upper center part is varied, while the other ones are set as dependent parameters and they vary according to the input assigned value of U1, maintaining the same intervals between them.

Regarding the tank filling level, its variation greatly influences the natural frequencies results. It means that the effects of the other parameters variation in the natural frequency results can not be properly compared if they are obtained in different fluid levels, therefore the filling level is not considered in all parameters combination. Instead, all simulations are performed with constant filling level of 95% in each bulkhead, the same as the original measurements so the appropriate comparisons can be carried out. The fluid level influence, instead, is only evaluated regarding the original geometries, varying from 10 to 90% in intervals of 10% and an additional 95% level for comparison purposes.

3 CHAPTER 3 - FATIGUE EVALUATION

When designing a ship, some limit states have to be taken into consideration to ensure structural resistance, among them the fatigue state. In fact, structures that are subjected to cyclical loads may fail even if the applied load produces a considerably lower stress when compared to the yield stress and it happens due to the fatigue

Fatigue occurrence can reduce the life cycle of a structure when it initiates cracks that are constantly propagating until leading to a failure (**Barsoum**, 2020). Therefore fatigue evaluation are carried out during design stages in order to avoid or mitigate fatigue related failures, which could lead to unexpected repair or even unwanted events, such as accidents (**Fricke**, 2017).

In this Chapter, a preliminary fatigue evaluation is carried out taking as object of study the original geometry of Bulkhead A. The approach and the analysis features are explained in the next subsections.

3.1 Approach

To assess the fatigue life cycle of a ship structure, different approaches can be applied depending on their efficiency and applicability. Since the bulkheads issue configures a problem in welded joints, the following approaches can be implemented (**DNV**, 2021):

- Nominal stress S-N curves, also known as FAT classes;
- Hot spot stress S-N curve;
- Weld notch stress S-N curve.

For this evaluation, the selected approach is the nominal stress S-N curves. The nominal stress is defined as the stress that takes into account macro-geometric effects, but excludes the stress concentration due to structural discontinuities and the presence of welds (**DNV**, 2021). In these cases, when nominal stresses are well defined, the joint classification and corresponding FAT classes takes into account the local stress concentrations due to the joints and the welding profile. However, if the joint is located in a region with high stress concentration, a geometry dependent stress concentration factor K is applied to take into consideration this effect, as it can be seen in Figure 8.

Since this assessment is meant to be a preliminary evaluation of the fatigue life cycle of the bulkhead, only the nominal stress is considered in the following analysis. Nominal stress is generally obtained by using coarse or fine mesh Finite Elements analysis or using analytical calculation based on beam theory, including effective breadth of flanges (**DNV**, 2021).

In this evaluation, the nominal stress is obtained by a refined Finite Elements analysis, considering a limit state of 45 mm/s speed in the middle of the bulkhead´s plate, defined by classification rules for excited vibration (**DNV**, 2020).

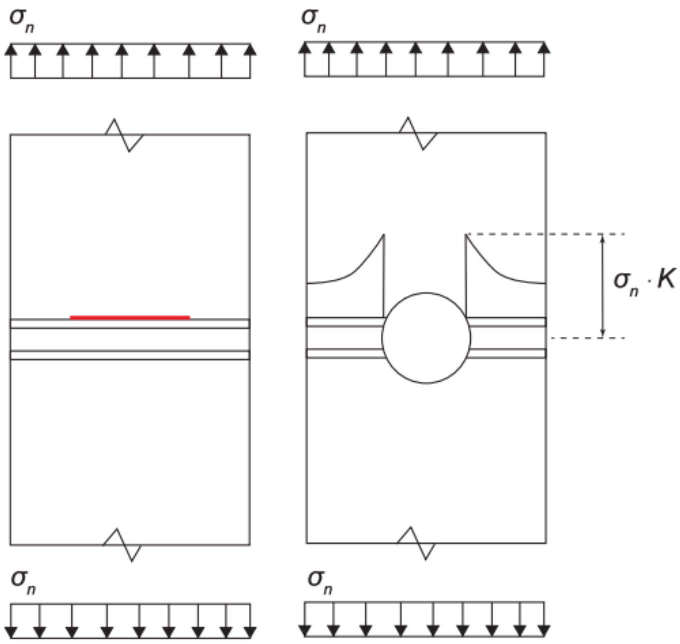


Figure 8: Nominal stress, σ_n, and local stress, $\sigma_{loc} = \sigma_n \cdot$ K (**DNV**, 2021).

3.2 FAT Class

In order to determine the proper parameters of the S-N curves, it is necessary, in a first moment, to define the appropriate FAT class, which is an important parameter in the nominal stress approach and takes into account local the stress concentration (**DNV**, 2021).

The FAT class is determined according to the selected welding detail. For the bulkhead problem, it is considered that the welding joints are cruciform welds with full penetration, as seen in Figure 9, which leads to a FAT class of 71 N/mm² to be used during this evaluation.

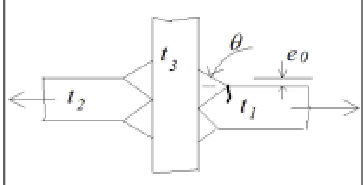
No.	*Geometry*	*Description of joint*	*K factor*	*FAT N/ mm^2*
1	 $e = \frac{t_1}{2} + e_0 - \frac{t_2}{2}$ $t_1 \leq t_2$	Cruciform or tee-joint K-butt welds with full penetration or defined incomplete root penetration. The effective weld thickness may be assumed as the thickness of the abutting plate t_1 minus *f*, where *f* is the incomplete root penetration of $0.2t_1$ with a maximum of 3 mm, which should be balanced by equally sized double fillet welds on each side. $e_0 \leq 0.3t_1$, $K_{mα} = 1.0^{4)}$, $K_{me} = 1.45^{3)}$, θ = 45°$^{1)}$ Cruciform joint: Tee-joint:	 1.27 1.13	 71 80

Figure 9: FAT Class determination (**DNV**, 2021).

3.3 S-N Curves

S-N curves, which are obtained from fatigue tests, are used to analyze the fatigue capability of welded structures. For practical fatigue design, structures can be divided into several classes, each one with a corresponding S-N curve, depending on their welded details and base material (**DNV**, 2021). The basic S-N curve is given by the following expression.

$$logN = logK_2 - mlog\Delta\sigma \tag{13}$$

Where,

- N = predicted number of cycles to failure;
- $\Delta\sigma$ = stress range, in N/mm^2;
- m = negative inverse slope of S-N curve;
- logK_2 = logK_1 - 2δ
  - K_2 = constant of design S-N curve, representing 97,5% probability of survival;
  - K_1 = constant of mean S-N curve, representing 50% probability of survival;
  - δ = standard deviation of log N = 0,2.

The S-N curves used for this analysis can be seen in Figure 10, while their detailed parameters are shown in Table **??** in Appendix **??**. The FAT Class of 71 N/mm² selected in the last section leads to the curve F in Figure 10, which parameters are summarized in Table 3.

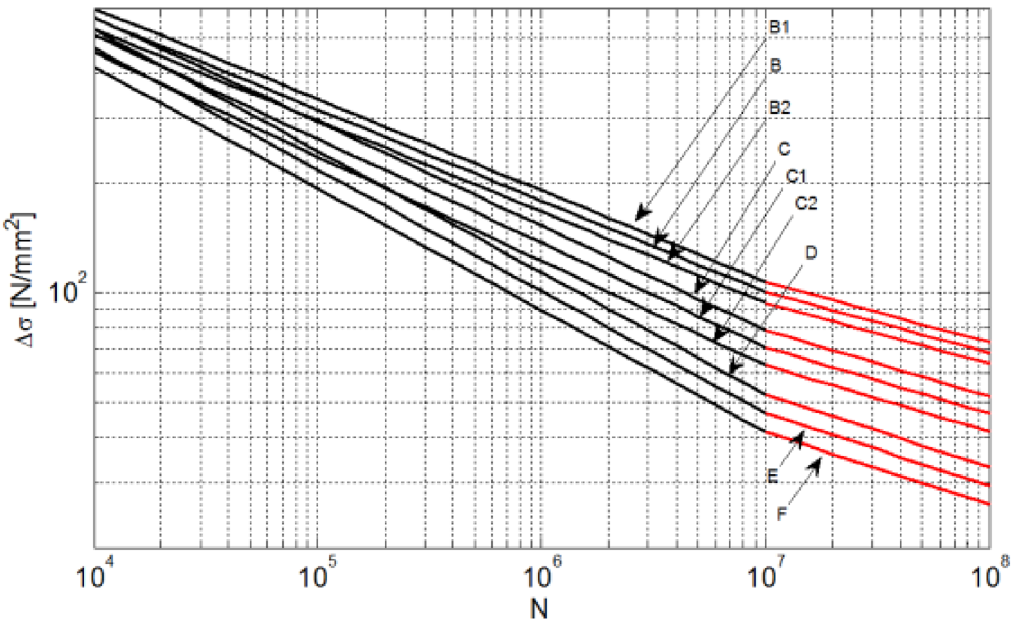


Figure 10: S-N curves (**DNV**, 2021).

Table 3: S-N Curves parameters for FAT = 71 N/mm².

S-N Curve	Reference stress at $2 \cdot 10^6$ cycles (FAT)	Reference stress at 10^7 cycles (knuckle) $\Delta\sigma_q$	Structural stress concentration embedded in the detail and taken at $2 \cdot 10^6$ (10^7) cycles	$N \leq 10^7$		$N > 10^7$	
	N/mm²	N/mm²	-	$\log K_2$	m	$\log K_2$	m
F	71	41,52	1,27 (1,27)	11,855	3	15,091	5

3.4 Stress Range

As mentioned before, the selected approach takes into consideration the nominal stress, which excludes discontinuities and stress concentration. In order to determine it, a new modal analysis was performed in the model of the original Bulkhead A geometry, but now with a finer mesh of 0,05 m elements in the region around the bottom welds.

Since the analysis provide an eigenmode stress, it has no physical meaning and some assumptions have to be made to proceed with the analysis. With the eigenmode displacement, one can approximate the nodal speed by following the expression below.

$$V = 2\pi \cdot \delta \cdot f \tag{14}$$

Where the term $\omega = 2\pi \cdot f$ is the angular frequency which is multiplied by the nodal displacement to determine the node linear velocity generated by the natural vibration (**Cummings**, **et al**, 2006).

In this scenario, it is now relevant to determine the nominal stress in the plate field, which will be used for the fatigue evaluation. The analysis of the welds between the bulkhead and the hopper tank plate show some regions of stress concentration. Therefore, a wider region around the stress concentration regions is considered to obtain the nominal stress.

Considering the limit state in which the velocity in the middle of the plate is 45 mm/s, a quadratic proportional conversion is made in order to estimate the nominal stress in the given limit state.

3.5 Evaluations

With the parameters defined in Table 3 and the stress range estimated, the equation 13 can be solved to determine the number of cycles to failure. Later, considering that the bulkheads operate in corrosive environment, a factor of 0,5 is applied to simulate the reduction in the structure´s lifespan due to the corrosion

Taking into consideration the overall number of cycles until failure and the operation frequency, the failure would be reached after approximately 944 days, or 31,5 months. It means that cracks would start to appear in the welds after approximately 2,5 years of uninterrupted operation in resonance conditions.

It is important to mention that this is a preliminary evaluation and that deeper and more detailed analysis are necessary to better determine the fatigue failure occurrence. However, this first analysis already indicates that regular repairs would be necessary is operated over longer periods of time in resonance condition.

4 CONCLUSION

The present project was carried out in a context of excited vibration of corrugated bulkheads in cargo tanks, an issue of major relevance in ship structural design, since it encompasses a multidisciplinary background with tasks related to modal analysis, structural strength, hydrodynamics and fatigue assessment. The main goal of the project was to carry out a parametric analysis of typical corrugated bulkheads by analyzing their natural frequencies variation according to changes in the design parameters, as well as to selected more appropriate geometries that could avoid the resonance occurrence. In addition, a preliminary fatigue assessment had to be carried out to evaluate the fatigue life of such bulkheads under resonance conditions.

The study started with the set of a context and a brief review over free and forced vibration, as well as the resonance phenomenon, in order to understand the basis of the problem. Following, a parametric study of the object was carried out, passing through all steps, since modelling, hydrodynamic mass, mesh features and boundary conditions. The developed finite elements model was properly validated, taking as reference real measurements at a typical bulkhead and, after the parameters range was defined, the simulation could be carried on.

The effects of the design parameters on the eigenfrequencies were analyzed both quantitatively, though the Tables in Annex, and graphically, by the means of the curves that were plotted. It was able to see the effects of the thickness variation on the natural frequencies and identify that the same parameters, varying in the same range, can generate different results depending on the bulkhead geometry. Finishing the modal analysis, new geometries could be selected for both studied bulkheads, improving their performance when submitted to the excitation frequency and avoiding the occurrence of resonance.

Next, the study went through the fatigue scope by providing a brief explanation regarding the different available approaches to assess the fatigue strength of a structure, culminating with the selection of the Nominal stress S-N curves approach to proceed with the evaluation. Therefore, an appropriate FAT class was set, the stress range was defined after a refinement in the finite elements model mesh and considering a limit state speed of 45 mm/s.

The fatigue evaluation lead to a conclusion that the approximate fatigue life of the Bulkhead A, under the given circumstances of permanent operation in resonance condition, would be of less than three years.

After analyzing the work that was done, it is possible to say that the study delivered the answers to its initial questions, but there is a considerable space for improvement in further works. First of all, the analysis was conducted in ANSYS Workbench, which provided good quality graphic tools and friendly modelling and simulation environments. However, some unknown errors during simulation forced the reduction of some parameters range, which impacted negatively in final outcomes. Carry out the simulations one more time, but in ANSYS APDL, could provide another level of validation and even work around the simulation issues.

There is also space for improvement in the fatigue evaluation, because some simplifications were done in the way, since it was a preliminary assessment. The selected approach used nominal stresses to determine the stress range, which ignores stress concentration regions. The application of more detailed approaches with more accurate stress range would provide a clearer overview regarding the fatigue life of the bulkheads.

In summary, a multidisciplinary study was conducted regarding the vibration of cargo tank corrugated bulkheads, providing the answers and meeting the expectations of its initial objectives.

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