Assessment of Lifting Criteria for Crane on a Heavy Lift Ship ‘load check and fatigue life calculation’

Udit Sood

Master Thesis

presented in partial fulfillment
of the requirements for the double degree:
“Advanced Master in Naval Architecture” conferred by University of Liege
"Master of Sciences in Applied Mechanics, specialization in Hydrodynamics, Energetics and Propulsion”
conferred by Ecole Centrale de Nantes

developed at University of Rostock
in the framework of the

“EMSHIP”
Erasmus Mundus Master Course
in “Integrated Advanced Ship Design”

Ref. 159652-1-2009-1-BE-ERA MUNDUS-EMMC

Supervisors: **Prof. Dr.Eng./Hiroshima Univ. Patrick Kaeding,** University of Rostock

Dr.-Ing. Thomas Lindemann, University of Rostock

Internship Supervisor: **Mr. Helge Rathje,** SAL Heavy Lift GmbH**,** Hamburg, Germany

Reviewer: **Prof. Maciej Taczala,** University of Szczecin

Rostock, February 2017

ABSTRACT

‘Assessment of lifting criteria for cranes on a heavy lift ship'

By **Udit Sood**

A Greater demand of energy on land has led to an increase in the offshore drilling and energy harvesting activities. The shipping industry has shown a rise in demand for the specialised vessels called the heavy lift ships. These ships are designed to transport goods to the offshore sites for construction and maintenance and also heavy cargo from one place to another by sea. The heavy cargo loading and offloading operations are carried out using powerful cranes present on such vessels. The crane capacities determine the cargo consignments which need to be handled. They form a vital machinery on such type of vessels. When we compare cargo operations with the onshore operation, these are much more complicated both in terms of the cargo handling and transportation.

The work scope of this thesis comprises of review of standard lifting criteria for shipboard cranes provided by DNV-GL classification society. The load on the crane is assessed during the cargo operations. The stresses experienced at different locations are found out using finite element modelling. This data is then processed for fatigue analysis.

The comparative study of crane structure and operation rules created by Lloyds Register and DNV-GL classification society is also carried out. The best-suited rules for the company business are analysed.

Finally, a report summing up all the results, findings is prepared. Some valuable optimisation procedures are suggested which would serve for a better structural and fatigue life for the new coming model for such type of crane.

Keywords: DNV-GL, Lloyds Register, offloading, offshore, cargo operations, finite element modelling, optimisation, fatigue life.

ACKNOWDEDGEMENTS

I would like to express my sincere regards for my company supervisor *Mr Helge Rathje*, Senior Project Engineer at SAL Heavy Lift GmbH. For giving me the opportunity to work on a company project. And providing me with all possible assistance in terms of knowledge and resources during the internship.

I am grateful to my University of Rostock Professor *Dr.-Ing Patrick Keeding* and *Dr.-Ing Thomas Lindemann* ,for giving me a valuable chance to work on an industrial internship and providing me timely assistance in terms of knowledge for the sound understanding of my thesis. Without their essential guidance I would not have succeeded in my endeavour.

Besides my mentor I had been assisted by some of the professional and highly qualified engineers from SAL Heavy Lift GmbH. I would express my deep gratitude to *Mr. Arthur Stuckert*, Project Engineer, *Mr Jonathan Schmidt*, Senior Project Engineer, *Mr Ralf Tschritter*, Senior Project Engineer, *Mr Ben Konrad*, Project Engineer, *Mr. Bin Wang*, Project Engineer and *Mr Tilo Klappenbach*, Project Engineer who had guided me during my internship.

It was a great experience working with the highly qualified and professional staff of the SAL Heavy Lift GmbH .I would also like to thank Chief design engineer *Mr. Marco Doelling* from TTS crane for giving me the opportunity to work on the design of their crane model.

I would like to express my regards towards DNV-GL and Lloyds Register personals who had guided me with the rule and guidelines.

In the end, I would like to thank my family members my parents and my brother who had supported me in terms of moral support throughout my thesis work.

This thesis is developed in the frame of the European Masters Course in Integrated Advance Ship Design named EMSHIP for the European Education in Advance Ship Design.

Ref.:159652-1-2009-1-BE-ERA MUNDUS-EMMC.

TABLE OF CONTENTS

TABLE OF FIGURES

TABLE OF EQUATIONS

$$Crane\ Loads\ 1 = F_d\left(L_g + F_h(L_1 + L_{h1}) + L_{h2} + L_{h3}\right) \quad (1)$$

$$Crane\ Loads\ 2 = F_d\left(L_g + F_h(L_1 + L_{h1}) + L_{h2} + L_{h3}\right) + L_w \quad (2)$$

$$\Psi_{See} = 1 + \frac{v_r}{9.81}\sqrt{\frac{C_s}{L_{Nsee}}} \geq \Psi \quad (3)$$

$$\phi_1 = \phi_q = H_{1/3}\ [°] \quad (4)$$

$$b_r = \omega^2 xr(\mathrm{m}/sec^2) \quad (5)$$

$$\mathrm{F_{sa}} = (\mathrm{Fa\ x}\Delta \mathrm{L_N}) \quad (6)$$

$$\mathrm{Fa} = 1 - (1 + \sigma)\frac{\Delta \mathrm{L_N}}{\mathrm{L_N}} \quad (7)$$

$$\mathrm{L_W} = \mathrm{q.c_f.Aw}\ [\mathrm{N}] \quad (8)$$

$$q = \frac{v^2}{1.6}\ N/m^2 \quad (9)$$

$$\mathrm{v} = 44\mathrm{x}\left(\frac{\mathrm{h_L}}{10}\right)^{0.15} \leq 50\left(\frac{\mathrm{m}}{\mathrm{Sec}}\right) \quad (10)$$

$$\mathrm{V_R} = 0.5\mathrm{xV_L} + \sqrt{\mathrm{V_{in}^2} + \mathrm{V_t^2}} \quad (11)$$

$$\mathrm{V_H} = 0.1\left(\mathrm{H_{sig}} + 1\right) \quad (12)$$

$$\text{Lateral force} = \frac{\mathrm{W}}{100}\mathrm{x}\left(2.5 + 0.1 + \mathrm{rxn} + \mathrm{H_{sign}}\right) \quad (13)$$

$$\text{Radial force} = \frac{\mathrm{W}}{1000}\mathrm{xn^2xr} \quad (14)$$

$$\text{Radial force} = \frac{2.5 + 1.5\mathrm{xH_{sign}}}{\mathrm{H_w} + \mathrm{L_{vert}}}\mathrm{x}(\mathrm{Wx}\Psi) \quad (15)$$

$$\mathrm{D_g} = \frac{\mathrm{n_g}}{\mathrm{N_g}} \quad (16)$$

$$\mathrm{N_g} = \mathrm{AxS_g^{-m}} \quad (17)$$

$$\mathrm{D_{RC}} \leq \mathrm{D_{RFC}} \leq \mathrm{D_{LCC}} \leq \mathrm{D_{PC}} \quad (18)$$

$$\mathrm{s} = \frac{\mathrm{s_2} - \mathrm{s_1}}{2} \quad (19)$$

$$\mathrm{s} = \frac{\mathrm{s_2} + \mathrm{s_1}}{2} \quad (20)$$

$$\sigma_\mathrm{m} = \text{Mean stress}\left(\left(\frac{\sigma_{\max}}{2} + \frac{\sigma_{\min}}{2}\right)\right)(\mathrm{N/mm^2}) \quad (21)$$

$$\frac{\sigma_a}{\sigma_{ar}} + \left(\frac{\sigma_m}{\sigma_u}\right)^2 = 1 \quad (22)$$

$$\frac{\sigma_a}{\sigma_{ar}} = 1 - \left(\frac{\sigma_m}{R_m}\right)^2 \tag{23}$$

$$N_{cal} = N_k * \left(\frac{\sigma_{ar}}{\sigma_{af}}\right)^k \tag{24}$$

$$\frac{\sigma_a}{\sigma_{ar}} + \left(\frac{\sigma_m}{\sigma_u}\right) = 1 \tag{25}$$

$$\sigma_{ar} = \sqrt{\sigma_{max}\sigma_a} \tag{26}$$

$$\sigma_{ar} = \sigma_{max}\sqrt{\frac{1-R}{2}} \tag{27}$$

$$\sigma_{ar} = \sqrt{\sigma_{max}\epsilon_a} \tag{28}$$

$$D = \sum_{i=1}^{n_b} \frac{n_i}{N_i} \tag{29}$$

$$\Delta\sigma \leq 1.5 f_y \tag{30}$$

$$\Delta\tau \leq \frac{1.5\, f_y}{\sqrt{3}} \tag{31}$$

$$\frac{\Upsilon_{Ff}.\Delta\sigma_{E,2}}{\Delta\sigma_C/\Upsilon_{Mf}} \leq 1.0 \tag{32}$$

$$\frac{\Upsilon_{Ff}.\Delta\tau_{E,2}}{\Delta\tau_C/\Upsilon_{Mf}} \leq 1.0 \tag{33}$$

$$\left(\frac{\Upsilon_{Ff}.\Delta\sigma_{E,2}}{\Delta\sigma_C/\Upsilon_{Mf}}\right)^3 + \left(\frac{\Upsilon_{Ff}.\Delta\tau_{E,2}}{\Delta\tau_C/\Upsilon_{Mf}}\right)^3 \leq 1.0 \tag{34}$$

$$N_R = 2x10^6 \frac{(\Delta\sigma_C/\Upsilon_{Mf})\hat{}m}{\Delta\sigma_{ch}^m} \tag{35}$$

$$2x10^6 \frac{(90/1.15)\hat{}3}{192\hat{}3} = 103515 \text{ cycles} \tag{36}$$

$$\lambda = \frac{1}{2x10^6} x \sum_1^8 \left(\frac{\Delta\sigma_{ch}}{\max\Delta\sigma}^m x\, n_E\right)^{\frac{1}{m}} \tag{37}$$

$$N = \frac{A}{S^m} \tag{38}$$

$$\text{Log } N = \log A - m\log S \tag{39}$$

LIST OF SYMBOLS

RFEM	RFEM Finite Element Software Package
DNV-GL	Det Nortskey Veritas
MATLAB	MATLAB Software Package
Lloyd’s register	Lloyds Register
AutoCAD	AutoCAD Design Software
CLAME	Code For Lifting Appliances in Marine Environment, Jan 2016
LR	Lloyd’s Register
FEA	Finite Element Analysis
DIN	Deutsche Industry-Norm
(t by t)	Tip stress and strain based method for stress calculation
SWL	Safe Working Loads
IACS	International Association of classification Society
Orca Flex	Orca Flex Hydrodynamic calculation Software
Octopus	Octopus Hydrodynamics Load Calculation Software
CAD	Computer-Aided Design
SAL Heavy Lift	Schifffahrtskontor Altes Land GmbH & Co. KG
RStab	Stability calculation finite element software
SN curves	Stress Number of Cycle curve
SWT	Smith, Watson and the Topper method
M.V Lona	Motor Vessel Lona
Miller Rule	Damage calculation method
RFAT	Add on module for Fatigue calculation
EN	European Norms
S-355	Steel class S-355
2-D	2-Dimessional
3-D	3-Dimessional

LIST OF TABLES

DECLARATION OF AUTHORSHIP

I declare that this thesis and the work presented in it are my own and have been generated by me as the result of my own original research.

Where I have consulted the published work of others, this is always clearly attributed.

Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely my own work.

I have acknowledged all main sources of help.

Where the thesis is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.

This thesis contains no material that has been submitted previously, in whole or in part, for the award of any other academic degree or diploma.

I cede copyright of the thesis in favour of the University of Rostock..

Date:13.01.2017 *Signature:*

1. INTRODUCTION

1.1 Preface

A greater demand of energy on land has led to an increase in the offshore drilling and energy harvesting activities. The shipping industry has shown a rise in demand for the specialised vessels called the heavy lift ships. These ships are designed to transport goods to the offshore sites for construction and maintenance and also heavy cargo from one place to another by sea. The heavy cargo loading and offloading operations are carried out using powerful cranes present on such vessels. The crane capacities determine the cargo consignments which need to be handled. They form a vital machinery on such type of vessels. When we compare cargo operations with the onshore operation, these are much more complicated both in terms of the cargo handling and transportation. The heavy cargo loading and offloading operations are carried out using powerful cranes present on such vessels. The crane capacities determine the cargo consignments which need to be handled and form vital machinery on heavy lift vessels. When we compare cargo operations with the offshore operation, these are much more complicated both in terms of the internal load handling within the installation and the sea-lift operations of a supply vessel. The reason is that during high sea operations experiences wave induced motions. The boom tip of the crane will have wave induced motion components in both the horizontal and vertical directions. The magnitude of these motions may vary with the sea state and wave heading angle.

Lifting operations on a heavy lift ship are much more demanding for the crane components than similar operations being carried out on a fixed offshore platform.

The topic -‘Assessment of lifting criteria for cranes on a heavy lift ship’ relates to the crane on the heavy lift ship which is one of the most essential machinery, for the vessel trade and business. Special care needs to be taken in its design, maintenance and operation. Shipping companies owning heavy lift ships have special designers, operators and the handler for the equipment. In order to ease the structural design and inspection during long operation of the crane on such vessel this thesis topic has been chosen.

The study starts with accessing the structural strength. This is achieved by modelling using an Auto Cad and Inventor software.

The second task comprised of the determination of the loads acting on the structural members and determines the load cases. It is seen from the rules that the loads acting on the structure are a summation of all the live loads, wind loads, ship accelerations, cargo loads and dead loads and other physical parameters such as the heel and the trim effects.

After completion of the calculations the loading conditions are imposed on the finite element model and it is simulated. The calculations are done based on the analytical formulation given by DNV-GL Rules for the lifting appliances [REF 09].

The summation of these load cases is carried out by the MATLAB code which compares the stresses at an individual nodal point and calculates the damage using the Miner's Rule.

The final part of the thesis comprises of the comparison of the rule requirements for the structural components of the crane. In my thesis, I have made comparisons between the classification society DNV-GL and Lloyd's register [Appendix B]. Different amplification factors used during the design calculations are compared.

Together all these calculations, results and observations are summed up to form a compiled master thesis to be submitted for university evaluation.

1.2 Scope of study

The title of the thesis is "**Assessment of lifting criteria for cranes on a heavy lift ship**". The work scope is divided into sections and has been explained in the following sequential order:

1) Study of the crane structure and review of standard lifting criteria for shipboard cranes provided by DNV-GL and Lloyds register classification society:

This part of the project comprises of defining the crane structural components and then compiling major rules designed by the classification society for the lifting of heavy. The safety requirements and the limits for heavy lift ship will be assessed. The criteria followed during and before lifting in order to calculate the load on the deck as well as on crane boom and the crane housing will be summed up to find the safest means to carry out the cargo operation.

2) Classification of measured crane load of different operating condition and modelling in AutoCAD and Inventor in order to obtain actual dimensions of the crane.

The crane is subjected to different load cases and crane positions this will be assessed in terms of the crane boom angles. These parameters are determined to find out the appropriate boom angle with the reverent load case.

The target is to calculate stress at different loading conditions on the crane structure using finite element software. The crane boom and the housing are subjected to different stresses at different loading conditions. These stresses for eight loading conditions will be found out. Calculation of the fatigue on the structure of the crane using the damage criteria.

3) MATLAB software is used for reading all the stress tensor values obtained from the finite element software to find the fatigue on the nodal points of the crane structure. A report is prepared summing up all the results, findings and the suggestions. The thesis work is prepared with the approval of the operator, manufacturer and the classification society.

In the end the optimization procedures are suggested for operation and manufacturing of cranes.

2. CRANE STRUCTURAL COMPONENTS

The crane is divided into many structural components and special safety considerations need to be taken for each of them prior to estimate the lifting loads:

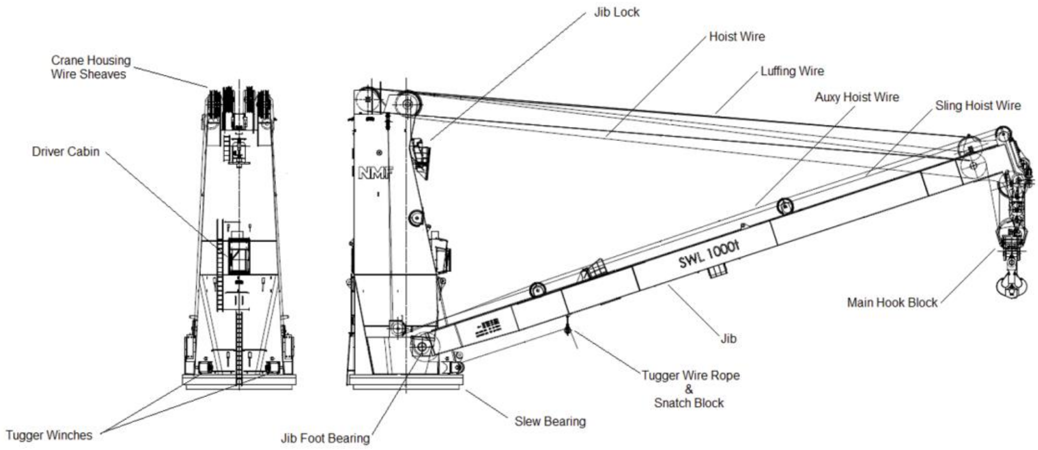


Figure 1: This is the 2-Dimension view of the shipboard crane ref:[01]

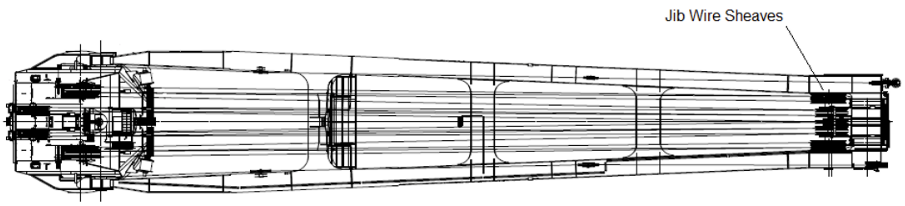


Figure 2: This is the top view of the boom structural member of the crane ref: [01]

The crane as a complete machinery can be divided into different parts, which makes it easy for the analysis and study of its gigantic structure.

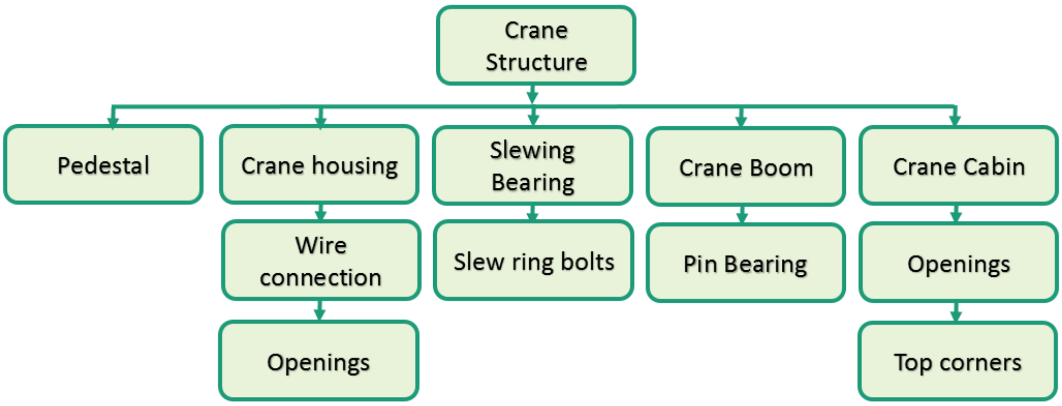


Figure 3: The structural components contributing crane structure

2.1 Crane Jib:

The jib is constructed in the shape of a twin box girder which is connected by three cross braces and a boom tip has a sheave block.

The jib is attached to the crane housing by two foot bearing. The connection is made by a pin which rests on spherical bushing. The crane jib is one of the heaviest structural components of the crane (152 tons) refer: [01]. It bears the compressive loads of the cargo and is supported by the wires on the other end. The jib being long (37 meters) and suspended shows the maximum deflection at its tip. It is the structural member transferring the major load of the cargo to the crane housing. The safety and the fatigue requirements for the crane boom are important as it possesses a major risk for the failure due to its large outreach and cantilever type behaviour as seen from the finite element results.

2.2 Crane slew bearing and the system:

The rotation of the housing about its axis is carried out using a slew bearing which allows the crane to rotate 360 degrees about its axis. The sealing is done using lip seals which prevent entry of unwanted material into the oil lubricated chamber. refer :[01]

Slew bearing is grease lubricated and need to be lubricated periodically using a grease gun. The slew ring bolts clamp the crane housing stationary part of the bearing to the deck housing.

The crane jib is kept in a locked position using two lock pin when the ship is moving and the crane operation is not required.

The slewing operation is carried out using fifteen slew drive assembly having an axial variable displacement pump and a planetary gearbox. [01]

2.3 Crane Cab:

This is a steel chamber with windows for the operator to sit. The crane cab is an extension of the crane housing and is connected to the structure by a cut out window in the crane structure. It forms the place of the clear view of the operator of the crane.

2.4 Structural Failure:

Types of the failure the crane can experience during its lifetime:

1. Structural failure.
2. Loss of sealing and weather protection.
3. Bearing damage.
4. Damage by contact.

In this assignment, we concentrate on the failure of the structural components of the crane.

Causes which may lead to structural failure of the crane and its components:

a. The operating condition which may cause the greatest impact to the structural integrity of the crane is the load operations being carried out when the vessel is excessively heeled or trimmed due to improper ballasting.

b. Inadvertent contact of the crane jib with another structure.

Criticality of damage to various parts of the crane: refer: [01]

a. Crane structural pedestal:

	Failure effects						
Failure mode	**Equipment**	**On System**	**Operation**	**Early detection**	**Probability of occurrence**	**Severity**	**Criticality**
Structural failure	Damage	Loss of integrity	Stopped	Inspection	Low	Major	Low

b. Crane Housing:

	Failure effects						
Failure mode	**Equipment**	**On System**	**Operation**	**Early detection**	**Probability of occurrence**	**Severity**	**Criticality**
Structural failure	Damage	Loss of integrity	Stopped	Inspection and strain gauge readings.	Low	Major	Low

c. Slewing System:

	Failure effects						
Failure mode	**Equipment**	**On System**	**Operation**	**Early detection**	**Probability of occurrence**	**Severity**	**Criticality**
Structural failure	Damage	Loss of integrity	Stopped	Inspection and strain gauge readings.	Low	Major	Low
Bearing Failure	Damage	Loss of integrity	Stopped	Inspection	Mild	Major	Medium
Bolt failure	Damage	Loss of securing	Potentially damaging	Inspection	Mild	Critical	Low
Slew gear failure	Damage	Rotation compro mised	Stopped	Inspection	Mild	Major	Medium

d. Crane jib:

	Failure effects						
Failure mode	**Equipment**	**On System**	**Operati on**	**Early detection**	**Probability of occurrence**	**Severity**	**Criticality**
Structural failure	Damage	Loss of integrity	Stopped	Inspection.	Low	Major	Low
Bearing Failure	Damage	Luffing compro mised	Stopped	Inspection	Mild	Major	Medium

3. COMBINATION OF STRUCTURAL PARTS TO FORM COMPLETE CRANE.

In order to reduce the production cost dynamic design calculation need to be carefully carried out. In order to do so computer based software such as CAD and finite element methods is usually used. In my case we I have made use of company license finite element software of RFEM to determine the loads at the various components and then use these calculations for the structural design.

This part of the design is the most critical and stressful. As the weakest link determines the overall performance of the machinery. Over or under calculations can prove harmful for the design of the component. And as it is very difficult to consider the crane as a whole and the dynamic performance is difficult to calculate. The residual and design imperfections are always a mystery in the design phase. As the dimensions of the crane structure are large, so the buckling and elastic deformations also need to be considered. The impact loads and the inertial loads also complicate the problem. All these factors create a big difference in the static design model and the actual working conditions. Other loads which may create nonlinear response are wind loads, Heat, cold and the wave induced effects on the crane. It is quite difficult to describe the force on the crane accurately as the loads change with respect to time. The traditional design methods cannot meet up with design requirements of modern offshore crane and high end motion analysing software's such as an RFEM nonlinear calculation need to be employed which are expensive and require skilled operators to work on the software.

Many researches are being carried out in this field to meet up with the problems faced with the earlier designs. In order to ensure that the crane bears with the harsh working experience due to the alternating loading, wind force and the wave affects the classification society has set up certain design requirements and proposed a factor of safety for the design of the crane parts.

This rule states that the structural components of the crane the dynamic amplification factor Ψ (factor of safety) shall not be taken less than: [REF: DNVGL 0378, Section 10, Page 100, Edition May 2016]. [02]

- Ψ =1.3 for lifting loads for W up to 2500KN
- Ψ =1.5 –W/12500 for 2500Kn<W<5000KN
- Ψ =1.1 for W>5000KN

Where W is the maximum working load and Ψ is termed as the dynamic amplification factor.

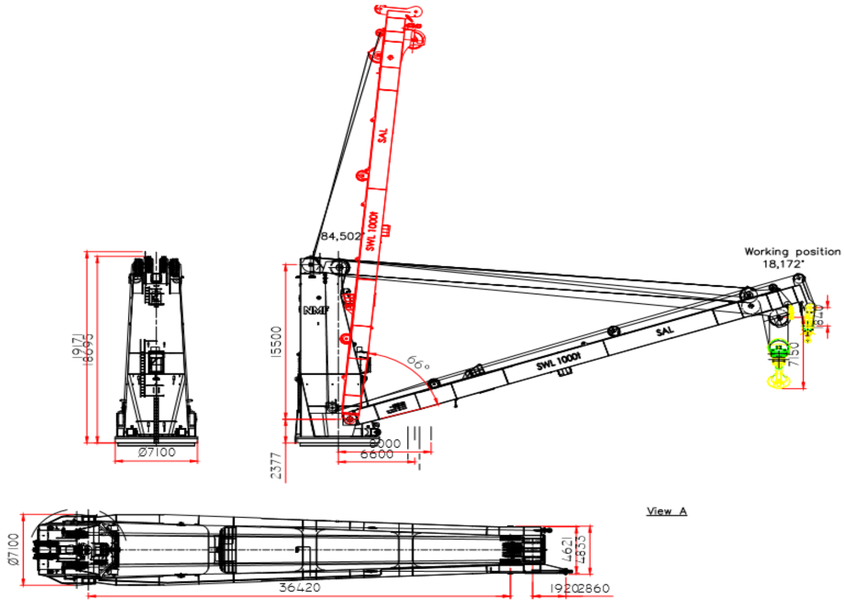


Figure 4: A gernalized dimensioning of the crane and its essential components refer: [01]

4. STUDY OF THE CLASSIFICATION RULES FOR LIFTING APPLIANCES

To carry out cargo operations the vessels need to comply with the rules specified by classification society or the European Union rule requirements. The classification society chosen for the assignment is DNV-GL. And the certificate provided after inspection is in compliance with DNV GL guidelines which is valid for the shipboard crane operations carried by on-board ship operating under safe working conditions.

In this thesis the following rules will be used in order to carry out the required calculations.

- DNV-2.22 lifting appliances
- DNVGL-RU-0050 cranes on crane vessels.
- DNVGL CG 0378,Offshore & Platform Lifting.
- Code For Lifting Appliances in Marine Environment (Lloyd’s Register)

The guidelines are given by the two major classification societies that are DNV-GL and the Lloyds Register regarding the construction, maintenance and the operations of the inland and the offshore cranes. They mainly concern the study of the static loads, allowable stresses and fatigue loading on the crane during its sea and offshore cargo handling operations.

The study is created after discussions with the classification representative’s senior specialist marine team leader “Dr. In Enno Alberts” and joint meeting held with “Mr Marco Hartmann” a lead specialist for lifting appliances Lloyd office in Hamburg.

The main areas covered in this part of the topic are the rules followed in the heavy lift cargo operation during the harbour lift. As the vessel is in sheltered water, there is little effect of the wave on the lifting operation. While on the other hand the cargo loads on the boom tip create significant deflection and create cyclic loading on the crane structure, which can pose a threat of fatigue due to continuous cargo handling. In the sailing conditions the boom is in the rested position or is in supported condition. It is usually locked with the boom tip facing upwards and lock pin inserted for additional support and restricting motion of the boom due to sea conditions. This load case is not much of a fatigue problem for the crane. The European rule specifies that the stresses which are below 50% of the yield stress of the material do not contribute for the fatigue loading on the crane structure. Hence the sea going condition with boom locked is not much of fatigue problem.

In order to analyse the loads on the crane we consider the static and the dynamic loading factors which are considered crucial during the design of the crane structure. These conditions are given below: refer: [02]

- On board and off board lifts.

- Side loads, and the wind effects.
- Vertical dynamic loads
- Heal or trim of the ship during the cargo operations
- Allowable stress of the structural components
- Special consideration for the crane foundations, slewing bearing and the pedestal
- Storing position of the crane boom during non-crane operation condition subjected to wave conditions at the sea

These conditions are monitored by keeping a close check on the given parameters.

- Vertical velocity of the vessel during operation
- Hoist velocity
- Loading conditions for the crane structure

5. LLOYDS REGISTER RULES AND GUIDELINES FOR LIFTING APPLIANCES

The factors which have a drastic influence on the life of the crane and its structure have been specially controlled in terms of the force limitation. The rules and guidelines are founded under the heading of "Code For Lifting Appliances in Marine Environment, Jan 2016" under Lloyds Register guidelines. The detailed specific rules can be found in under the heading of "Cranes and submersible Lifting Appliances" gives us a brief algorithm formula of the major loads which are kept in considerations during the design phase of the crane. These loads are characterized into four different cases: refer: [04]

5.1 Case 1

For the crane operating without wind, the design is to be considered with respect to the combination of the dead load, live load horizontal forces and the dynamic forces due to the crane movements. Given by:

$$\boldsymbol{Crane\ Loads} = \boldsymbol{F_d}\left(\boldsymbol{L_g} + \boldsymbol{F_h}(\boldsymbol{L_1} + \boldsymbol{L_{h1}}) + \boldsymbol{L_{h2}} + \boldsymbol{L_{h3}}\right) \tag{1}$$

Where:

- F_d=Duty factor
- L_g=dead load
- F_h=Live load
- L_l=Hoisting factor
- L_{h1}=Horizontal component of the live load due to the heel and trim.
- L_{h2}=The next most unfavourable horizontal load.
- L_{h3}= The horizontal component of the dead load due to the heel and trim.

5.2 Case 2

In this case the wind loads are also considered in the calculations.

$$\boldsymbol{Crane\ Loads} = \boldsymbol{F_d}\left(\boldsymbol{L_g} + \boldsymbol{F_h}(\boldsymbol{L_1} + \boldsymbol{L_{h1}}) + \boldsymbol{L_{h2}} + \boldsymbol{L_{h3}}\right) + \boldsymbol{L_w} \tag{2}$$

Where:

- L_w=The most unfavourable wind load. Refer: [04]

Rest all the parameters are same.

5.3 Case 3

In this case "the crane is considered in the stowed position when subjected to the forces subjected from the accelerations due to the ship's motion and the static inclination, together with the wind force. The effects of the anchor locks and lashings are to be taken into consideration. Refer: [04]

5.4 Case 4

In this case the “canes are to be considered into the exceptional load conditions” which are: Refer: [04]

a) Coming into contact with buffers.

b) Failure of the hoist wire or sudden release of the load for the cranes with the counterweight.

c) Test loading.

The Lloyds guidelines are the ones which have recently been updated and the Jan 2016 issue. The guidelines given under “Fatigue design assessment application and notification “given by the LR represent the application, responsibilities and the Fatigue design assessment procedures for the ship structure. The crane foundation is analysed for the fatigue, and it is at the discretion of the manufacturer to go for fatigue analysis for cranes structure operating on the heavy lift ship. It is mostly seen that as the cranes on such ships have very few loading and the discharge operations during their lifetime the load cycles experienced during its life are quite low (below 20,000) hence the fatigue does not really pose a topic of much of a concern for crane structure of heavy lift ships. Reference is again made to Chapter 4, Section 2.3.4 of the CLAME. Ref: [05] Makes the decision how much of a concern fatigue shall be for the design and operation of a crane depends on the intended use of the crane and the way it is stowed on board. For cranes which are rarely used, such as some shipboard cranes which are turned away from the quayside while in harbour and where the mobile harbour cranes are utilized to unload the vessel, fatigue might be less of a concern as for jib cranes in a trans-shipper compared to the container or the feeder ship crane which are heavily utilized. The classification society is really concerned about the storage position of the boom when the ship is in its sea voyage or at anchor with no cargo operations taking place. The storage of the boom resting freely or with the two booms tied up to each other in a freely hanging position have proved to be with future studies a case of major fatigue crack initialization case. It is also found that the boom locked on the housing with its boom tip upwards poses a concern for fatigue crack on the structure of the housing. Hence, even though the class does not give direct rules for the calculation of fatigue for the structure at lower stress range (below the endurance limits and stresses below 10% of the maximum stress), but still they ensure that these practices which have proved to be a major concern for the fatigue crack initialization are discouraged on board.

The FEA that is “Federation European De La Manutention” ref: [06] which is an organization operating in the European Union have specified rules for building crane and other lifting beam.

In their rules under the heading of "Rules for the design of hoisting Appliances" in the chapter "DIN 15018-1" ref: [08] specifies the structural requirements for the building of the crane based on fatigue calculations. These design criteria are followed by the manufacturer during its design phase. It also specifies the load cases, calculation, verification and analysis of the structure and the ropes used in the cranes for the hoisting and the lowering of the crane boom. Ref: [08]. The regulation brings out the significance of the duty factor which has been incorporated for the structural design criteria given in the Lloyds rules. These rules are kept as the benchmark for the design rules for fatigue by LR. These rules are specified under the heading of "Lifting appliances in a marine environment". The definition of the "duty factor" can be found under the heading of the design standards such as F.E.M. 1.001. ref: [06] the guidelines given by LR "duty factor" is defined as the "amplifying coefficient". It states that for pure fatigue calculations the duty factor may be set to 1,0. This also that the low cycle fatigue may need to be taken into consideration for the determination of the fatigue failure of the structural components of the crane having low duty cycles. Ref: [05]

While the guidelines provided under "EN-3.1.9 under DIN 4132" EN rules provides the guidelines for the fatigue assessment of the crane structure. These rules are not directly incorporated in the classification society, but they are used as the benchmark for all the calculations considered essential. The EN 3-1-9 states that in case the wind load increase the mean stress by creating a constant load on the structural components and hence increasing the chances of the fatigue failure of the component.Ref: [10] On the other hand the rule also points out that the effect of the wind is usually negligible and can be neglected for the calculations. It depends on the case whether the wind load has an influence in our case or not. But in order to be on the safer side, I have incorporated wind effect on the longer cross section of the boom structure which will give a bigger margin for the safety and the operational convenience of operator.

From the above we can conclude that F.E.M. 1.001 Ref: [06] rules state that it requires the operational load case without wind can be considered while EN-3.1.9 ref: [10] series of standards appears to be more widely accepted and followed.

The Lloyds register has three levels of fatigue design assessments:

Level 1: considers the welding details in comparison with a detail design guide;

Level 2: considers ship specific voyages (sea areas) and calculates stress concentration in way of bottom and side longitudinal intersections with transverse structures using a special software.

Level 3: makes use of the full ship motion analysis in connection with a fine mesh finite element investigation which results in a fatigue live prediction.

From this it is seen that the lifting analysis can be done by taking the wave height and the period or on the other hand by considering the ship specific motion analysis. The limitation is that the rule needs to be implemented fully, by either method and there should be no mixing of the criteria.

The allowable stresses calculated using these load factors and the rule requirement formulae given above are then tested for the yielding and fatigue of the component.

In this rule it is found that no mention of the swell is incorporated. It is usually assumed that in order to keep the structure safe maximum swell condition are given by the manufacturer.

This calculation for the fatigue is done in the design phase and need to be followed by the operators using the class rules and guidelines

Snatch loads are a major concern during the operation of the cranes. The snatch load is the situation when the hoisting speed of the crane is low and the wave created on at sea results the vessel to rise and, during this scenario ship hull can accidentally touch the suspended cargo and give it a reaction upward force. This reaction force will create fluctuating loads on the boom tip and may cause severe damage to the structure. This effect does not directly incorporate in the classification rules for the heavy lift ships. Snatch loads are, in general, to be avoided and safety checks are incorporated during the cargo handling conditions in order to prevent such situations. The dynamic factor for general purpose offshore cranes is (using the simplified method) is derived by taking into account the various velocities which are present during a lift, e.g. minimum and the actual hoisting speed, boom tip velocity and load supporting deck velocity. The prescribed minimum hoisting speed is kept as a reference, the risk of snatch loads occurring during the operations are kept under check. But in the heavy lift vessels due to the large weights of the cargo the hoist and the lowering speeds are really low. In order to prevent the lifted cargo touching the deck or transferring the loads to some structure, it is ensured that the sea conditions are fairly calm. The restricting conditions of the sea during the cargo operations are given by the manufacturer.

Figure 5: Ship 183 Type [Ref: www. http://sal-heavylift.com/]

Table 1: Ship crane specifications on ship 183 type [Ref: http://sal-heavylift.com/]

Cranes	**2 x 1,000 tons SWL,** **Combinable up to 2,000 tons**
Capacity	**1,000 MT @ 16m outreach** **800 MT @ 25m outreach** **500MT @ 38m outreach**
Slewing	**360 degree with hydraulic motor drive**
Luffing	**18.17 degree to 84.35 degree**
Hoisting	**Maximum boom tip height of 37.3 meters**
Operating	**5.4 degree inclination(max)** **(5 degree Heel and 2 degree trim.)**
Wind Speed	**20m/sec (Fatigue calculation)**

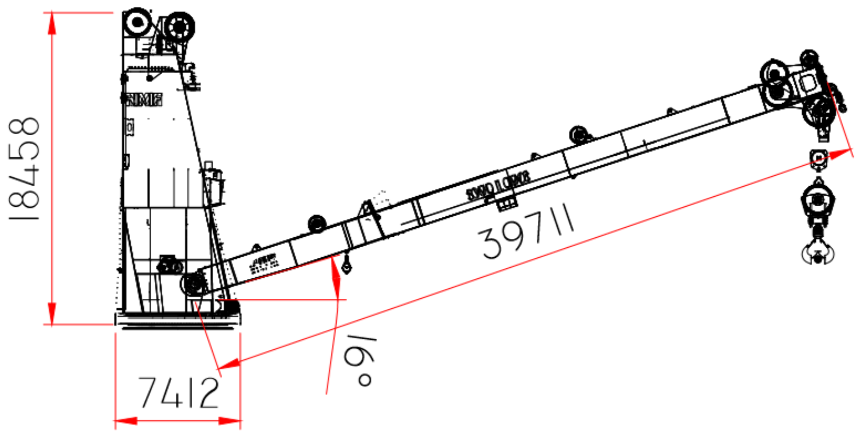


Figure 6: Basic dimentions of the crane and its angle of refrence

The engineered lift calculations eliminate the risk of snatch loads to occur during open sea condition lifts of shipboard cranes the clearance distance between the cargo lift and the structure is defined. Special formula and the rule requirements are incorporated in order to ensure these parameters.

It was searched in the rules that whether it was possible to lift loads by crane beyond the SWL. It was found out that it is not allowed. Only under special conditions on the joint approval of the flag state, port authority, classification and the manufacturer a load lift of 110% of SWL is allowed. But as we can see these are for special cases and the operation are deemed to be accepted for the situations which are specified. But in the normal operations the lift loads in no case should exceed the SWL capacity of the crane.

One of the major concerns of the crane manufacturers is to rate their crane with Safe Working Loads after the class inspections. It was searched in the rules, whether it was possible to get the class approval for the cranes for higher safe working loads for the crane for a similar model which had been allotted a lower lifting capacity by a different classification society. And as my comparison was mainly concerned with the classification society DNVGL and LR and both are members of IACS (International Association of Classification Society) I received an answer that was a “No”. As both of these classification societies are under IACS (International Association of classification society) the design factors set by them have an almost equal value. It is seen that the crane classed by one classification society when having a given SWL will have mostly the same SWL, when its similar model is tested by another class. But this value would never exceed the one which was given before. And if the manufacturer plans to change the class, first of all it requires a special reason which should be acceptable by IACS for the change of class. And still other classification society cannot rate SWL higher than the previous defined. This system gives uniformity in the inspection criteria and ensures that the safety is always ensured between all class inspections to a similar level. This similarity in the design requirements also extends to the design rules and the factors which will be discussed in the later sections.

6. DNVGL GUIDELINES FOR CRANE STRUCTURE

Classification society DNVGL provides special rule requirements for the classification of the heavy lift crane operating on the ships. It is seen that comparing the rules small differences are found among them. The major field of our concern will be design factors calculated with the different structural strength calculation procedures.

The structural analysis of the crane is important during the design stage as well as during its operating life. In order to obtain the stresses at various components of the crane the DNVGL rules and guidelines for the design and calculation of the loads are followed. Ref: [17]. This gives us the basic overview of the loads which we will experience and need to employ in order to get the actual load calculations and deflections on my finite element model.

The main field of the concern is the structural components of the crane. These structural components are constructed keeping in mind certain failure criteria and the loadings which may be harmful for its long working. These criteria need to be looked upon during the design phase of the crane structure:

- Design Structural Loads.
- Fatigue analysis.

The loads carrying capacity of the crane are defined as design load of the cranes. These loads determine the static and the dynamic loads the crane is designed to handle in the normal running conditions in its day to day business. The subcategories of the loads are described as.

1. Regular loads.
2. Irregular loads
3. Special loads.

6.1 Regular loads:

These regular loads are further subdivided into subcategories.

- Dead loads.
- Hoist loads.
- Dynamic forces due to the drive system.
    - Vertical dynamic force due to the lifting of loads.
    - Vertical dynamic force due to the suspended loads.
    - Horizontal driving force due to the lifting of loads.
    - Horizontal driving force due to suspending of the loads.
- Dynamic forces generated by the ship motion.
    - Vertical dynamic force.

  - Horizontal dynamic force due to inclination of the crane base.
  - Horizontal dynamic force due to acceleration of the crane basis.
- Diagonal pull loads due to the cargo runner deflection angles. Ref: [17]
  - Harbour operation.
  - Sea operations.
- Partial dropping off the useful loads during normal operations.
- Tie down forces on the cargo hooks.

The regular loads are the operating loads which the crane experiences during the majority of its life span. These loads are the usual loads which the crane manufacturer tests for 25 years of the crane operating life. It is further categorized into:

- Dead loads (L_E): Dead loads are the weights of all the fixed and mobile components of loading gear and loose gear permanently present during the operation. Dead loads are calculated by multiplying the mass by the acceleration of gravity g = 9.81 m/s². This is accounted as the structural self-weight in the finite element modal analysis.
- Test load (L_P): The test load (L_{P}dyn) of loading gear is the test load which is to be raised, lowered and braked by motor during the test using the drives (dynamic test).This load is 110% of the safe loads. This will be explained more clearly in the test certificate given below. This test is carried on the periodical basis and need to be logged in for the class surveys. Ref: [17]

Table 2: Load limits for inspection

Nominal loads (L_{Ne})	Test loads (L_{Pdyn})
1. Up to 20 tons	SWL +25%
2. 20 t to 50 tons	SWL +5t
3. Over 50 tons	SWL +10%

- Hoist load (L_H): Regarding the crane dimensioning, the nominal load which the crane handles are termed as the hoist loads. Sea operation: $L_H = (L_E + L_N)$

Where L_N which is a nominal load at sea can be calculated by the formulae of hoist load coefficient Ψ.

$$\Psi_{See} = 1 + \frac{v_r}{9.81}\sqrt{\frac{c_s}{L_{NSee}}} \geq \Psi \qquad (3)$$

Where:

- Ψ_{See} = hoist load coefficient for sea operation
- v_r =The relative speed between load and hook in the course of lifting the load [m/s]
- c_s = crane stiffness [kN/m]

- L_{Nsee} =Nominal load at sea [t]
- Ψ = hoist load coefficient for harbor operation
- Ψ_{See} = hoist load coefficient for sea operation

Ψ is also called as the dynamic amplification factor. This factor need to be multiplied with the lifted loads and it gives the real SWL of the crane. The loads combined with the self-weight of the crane gives the total loaded condition of the crane. This dynamic amplification factor is induced in the crane operation period due to the sudden breaking of the cargos and the sudden stopping effects of the crane operations which result in loads much greater than the actual weight of the cargoes to be applied on the boom head. So the lifting load calculations are carried out after multiplying the actual load weight with this coefficient. Now the question will arise that how do we calculate the nominal sea load L_{Nsee}which is one of the unknowns in the formulae. This is presently not required because we consider the ship to be operating in sheltered waters. But during the sea voyage this is an important criterion, but as the crane is without loads in that situation the stresses are below 50% of yield strength and hence does not contribute to fatigue failure of the component.

➢ Dynamic forces due to drive systems:

The dynamic forces of the system consist of the horizontal and the vertical forces which can be calculated from the formulae given in the equation 1. These calculations require stiffness which can be calculated by the instructions given. It is found out by multiplying the self-weight with the dynamic amplification factor that is 5% of the self-weight for the boom to find out the actual loading due to the dynamic conditions (As provided in the section above under Dynamic amplification factor)

The horizontal dynamic force angle can be calculated by the equation given below. Ref: [15]

$$\Phi_1 = \Phi_q = H_{1/3} \ [°] \tag{4}$$

Where:

- Φ_1: Deflection angle in longitudinal direction of the crane boom
- Φ_q: Deflection angle in transverse direction of the crane boom
- $H_{1/3}$: Significant wave height [m]

This factor and the weights are combined with the physical parameters such as the list and the trim factors to find the loads. Ref: [15]

For such operations the maximum trim angle (β) allowed is +/-2^o and the heel angle (α) allowed is +/-5^o. Horizontal forces due to the suspended loads are equal to:

$$b_r = \omega^2 xr (\text{m}/sec^2) \tag{5}$$

Where:

- ω: $(\pi x n)$/ 30 = Angular speed [1/s]=(0.0314s^{-1})
- r : Rotating/slewing radius [m]=38mts
- v : $\omega \cdot$ r = Circumferential speed [m/s]=1.1938m/sec
- n : r.p.m. [1/min]

➢ Dynamic forces due to the ship motion:

The calculation is to include the following influences: Ref: [15]

- Vertical and horizontal motions of the cargo deck
- Motion behaviour, on which the crane is mounted
- Structure weight of the crane
- Hydrodynamic properties of a floating or submerged load (neglected as in sheltered waters)
- Influence of anchoring systems
- Environmental conditions (Wind conditions considered)

➢ Diagonal pull loads due to cargo runner:

➢ The pull forces created by the crane should not exceed the requirements which states that: For such operations the maximum trim angle (β) allowed is +/-2^{o} and the heel angle (α) allowed is +/-5^{o}.

➢ Partial drop off of the load:

When a part of the useful load L_N is dropped during normal operation, in such cases a dropping factor f_a is considered.

$$F_{sa} = (f_a \, x \Delta L_N \quad) \tag{6}$$

$$Fa = 1 - (1 + \sigma)\frac{\Delta L_N}{L_N} \tag{7}$$

- ΔL_N:Change in load.
- L_N: Useful load (max 1000tons)
- σ: 0.5 for slow load dropping
  : 1.0 for fast load dropping

These are usually the instantaneous conditions occurring very few times in the crane structure. They are considered for the structural design phase and usually neglected in the fatigue calculations as they do not occur significant number of times during the crane life.

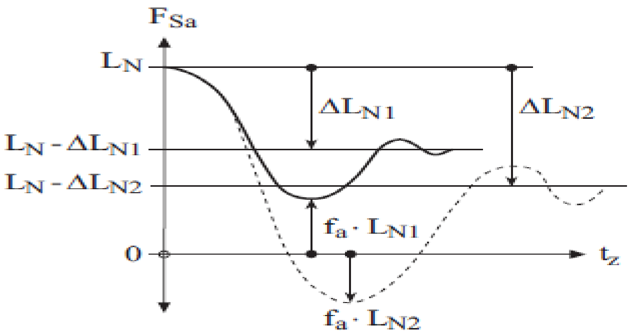


Figure 7: Determing the dropping factor f_a. Ref: [19]

6.2 Irregular loads.

These loads are further subdivided into subcategories: Ref: [17]

1. Wind load.
2. Snow and ice load.
3. Temperature load

6.2.1 WIND LOAD:

The total wind load acting on a crane structure is the sum of the wind loads acting on its various structural components. Ref: [15]

$$L_W = q.c_f.A_W \text{ [N]} \tag{8}$$

$$q = \frac{v^2}{1.6}\ N/m^2 \text{(Dynamic pressure)} \tag{9}$$

Where:

- v : Wind speed [m/s]
- cf : Form coefficient [-]
- Aw: wind area [m^2]

The maximum wind speed calculation for the loading and port operations are as provided below:

Table 3: Wind speed for various operations

Operating Mode	Wind speed at operation	Wind speed out of operation
1. Ship loading using its gear in harbour condition.	20m/s	50m/s
2. Ship loading using its gear in sea condition.	25m/s	50m.s
3. Offshore loading gear	25m/s	63m/s

But for the fatigue calculation the wind speeds are taken to be 20m/sec for the port operating conditions. So I have considered this wind as additional load on the structure.

The formula for the calculation of the wind velocity is: Ref: [84]

$$v = 44x\left(\frac{h_L}{10}\right)^{0.15} \leq 50\left(\frac{m}{Sec}\right) \tag{10}$$

Where:

- h_L = height of the center of area of the crane boom above waterline [m]

6.2.2 SNOW AND ICE LOAD:

A general ice accretion of 3cm thickness may be assumed for all parts of the construction which are exposed to the weather conditions. The specific weight of the ice is assumed to be 700 kg/m3. Ref: [18]

The specific weight of snow is assumed to be 200 kg/m3. In our case we do not consider the snow and the ice loads as the structure is designed in a form which cannot hold ice on it.

6.2.3 TEMPERATURE LOAD:

The temperature loads are to be considered in the calculated strength analyses. The European Rules state that in no case the temperature of the components to exceed above 150 degree Celsius as it has detrimental effect on the life of the structure. I have made a study of the reduction of the number of cycles due to the increase of the temperature of the structure. This will be discussed in the mean stress calculation section. Ref: [18]

6.3 Special loads.

The special loads are loads which are experienced during certain conditions and operations. Such loads may be of small interval, but involves larger forces so they form a critical part of the load determination and need to be observed during the crane design phase.

- Dynamic load testing.
- Buffering forces.
- Loads due to the safety system.
- Tear off the hoist loads.

Dynamic load testing are calculated using the given load

Table 4: Load limits for various loads

Nominal loads (L_{Ne})	Test loads (L_{Pdyn})
Up to 20 tons	SWL +25%
20 t to 50 tons	SWL +5t
Over 50 tons	SWL +10%

- Buffering forces: The impact force on the buffer is to be determined from the buffer characteristic and - in order to take into consideration the dynamic effect of the buffering force it need to be multiplied with the following factors fp:

fp: 1.25 for buffer with linear characteristic.

- Loads due to the safety system: The loads LS due to safety systems, such as e.g.
    - AOPS (Automatic Overload Protection System)
    - MOPS (Manual Overload Protection System)
    - ELRS (Emergency Load Release System)
- Tear off the hoist loads: These loads are for when in the catastrophic case of the hoist load is torn off, this result in fa = - 1, this can be seen from the *fig 7.*

The problems experienced by the crane structural parts in the earlier working models are usually analysed. This also forms only intermittent loads and not seen to occur significant times during the life of the crane operation.

From the above study we see that the maximum loads the crane can bear is defined by the maximum safe working loads (Other cases which exceed the safe working load do not have significant occurrence and hence are neglected). And this SWL loads are added with the structure weight, with the dynamic amplification amplitude for simulating motions and the physical and environmental conditions (Wind, list and trim).

These all finally contribute to the maximum loading a crane will withstand during its operating cycles.

7. FINITE ELEMENT MODELLING

The finite element analysis is a powerful tool for the structural analysis and analysis of 2-D or 3D models using the building elements such as members, plates, shell, solid or contact element.
The structure can be tested for the applied loads and also for the calculation of the linear and nonlinear load effects. This greatly reduces the long iterative calculations earlier required in such cases. The finite element provides a means of calculations for the deformations, internal forces, support forces, reaction forces and solid contact stresses. Further analysis for the stability and fatigue calculations are also carried out. These features of the software enable us to get quicker results for the effects on the members when they are subjected to the loading conditions.
The situation and the load conditions to be analysed are to be determined by the user and need to be accurately set in order to get the real state results for our calculations.

7.1 Element Description

The software provides feature to incorporate many types of the elements in its design in order to calculate the stresses. The building blocks can consist of the beam for the supports, shell element for major structural plating's and rib elements for the smaller supporting framework. These combine to form the finite element model. In my case I have used the shell elements for the analysis of the stresses representing the plate. It enables me to see the local buckling and other local phenomenon occurring at the sites of the connections and the welds and give us a detailed overview of the stresses. While the beams are used for the supporting members for the internal framework mostly comprising of the flat bar elements.

7.2 Material Properties

The material used for the structure is steel S-355 (E=210000N/mm^2). The material is chosen because the actual crane is made of this material type. And for the analysis point of view we use linear elastic and isotropic material. As the structure is made of the steel it follows the elastic nature and hence it falls into this category.

7.3 Modelling of the crane

In order to model the crane the following steps need to be followed to carry out the analysis.
Prior to the modelling the crane dimensions and the certain parameters such as the height, width, dimensions of plates and the plate thicknesses are required. Then the view of the crane in the top and front side views is modelled in AutoCAD. The dimensions which are missing are located in the manufacturer structural drawing or in the design manual. Some of the parameters are missing the company is contacted for the specific requirement and the data. The AutoCAD model needs

to be a perfect replica of the actual crane in order to obtain the correct values of the finite element model. In my assignment the crane boom and the crane housing are separated into two different components. This enables us to give simplicity the calculation and rotation of the model. Once the model is ready in AutoCAD it helps us give a picture of the crane how it looks like in the actual case scenario. In my model the crane boom and the housing parts are separated and are dimensioned with the parameters of all the curves and the corners in the right dimensions.

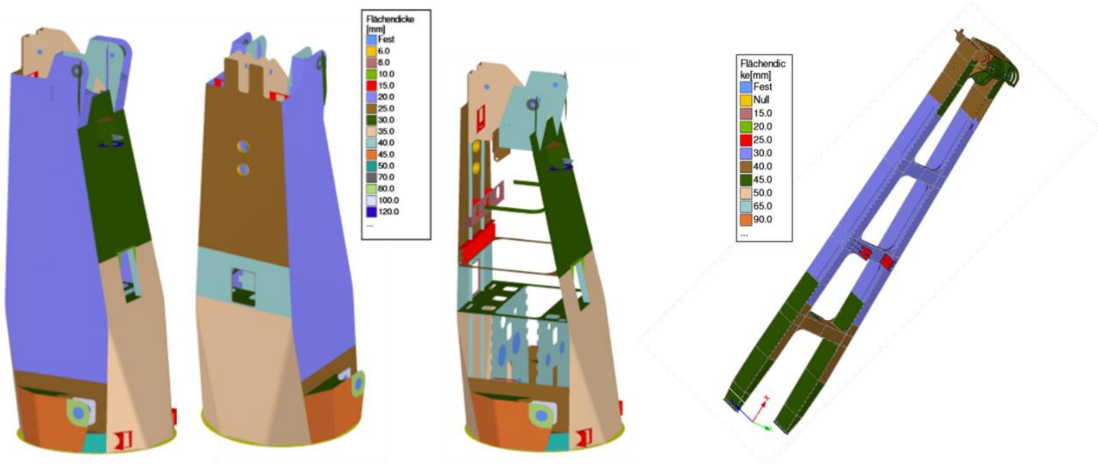


Figure 8: Crane structure as per the manufacturer drawings

The areas of the sharp edges and the rounded corners are specifically made with true dimensions as these are the parts which can be the stress concentration points and would be a matter of concern regarding fatigue.

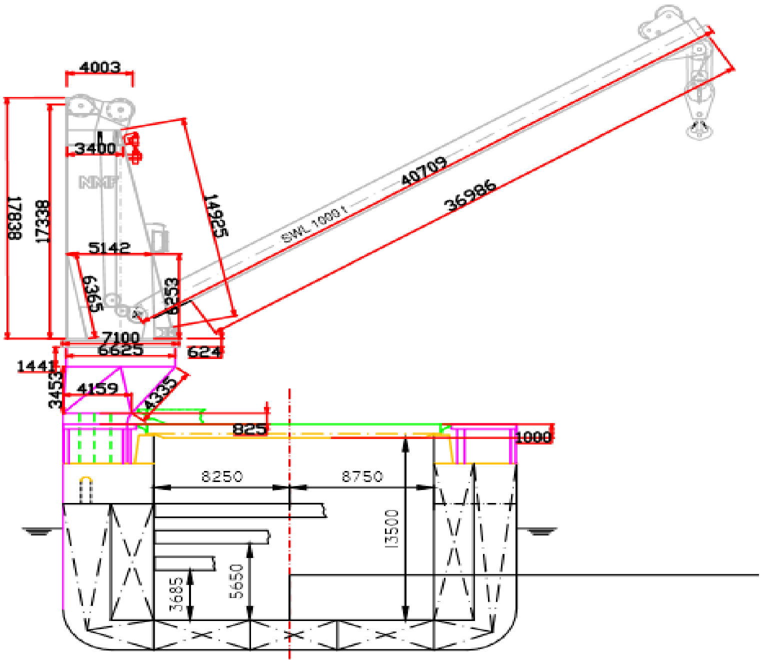


Figure 9: Front view of the crane consisting of the essential dimensions

This model would be beneficial for the company to present to the client who wishes to analyse the crane present on the ship. And also useful for me to further design the finite element model. The windows and the cut-outs are made in the model as they were of main concern in terms or stress points. The modelling of the back wall and the front operator chamber cut out is especially important for calculating the structural strength of the housing. Similarly the crane boom is quite a large member and is supposed to be elastic due to the material properties of the steel used for its construction. This boom member needs to be specifically analysed as it is supporting the loads on one side and is hinged using the bearing pin on the other. Similarly the boom tip is also being supported by the wires. The wires prevent too much of flexing of the boom and depending on the stiffness of the wires the boom tip will show deformations on application of the loads. This nature of the boom has to be checked for the deflections in the finite element model needs to be compared with the manufacturer given deflections (this is done by comparing model with the deflections of the boom finite element model of the manufacturer). As the crane is a copyright of most of the data about the structure was kept confidential. But SAL Heavy Lift being its business client had the opportunity to get the structural details lacking by the word of mouth from the chief design engineer of the crane manufacturing company.

The essential structural drawings were obtained and the position of the brackets and the stiffeners were placed on the crane after determining the stress concentration points at the structure and were later approved by the manufacturer by presenting my finite element model to their chief engineer. I had the opportunity to visit the manufacturer in order to get stiffening and other information which was not mentioned in the company drawings. These stiffeners and the brackets in the crane housing and the boom element were essential to calculate the structural strength of the component and needed to be modelled in the exact and accurate way. If any of these structural members were missing it would lead to calculation of different stresses in the finite element model from the real case scenario, so in order to get the real results of the structure the crane housing and the boom are accurately modelled and all the small and tiny steel parts are tried to be modelled.

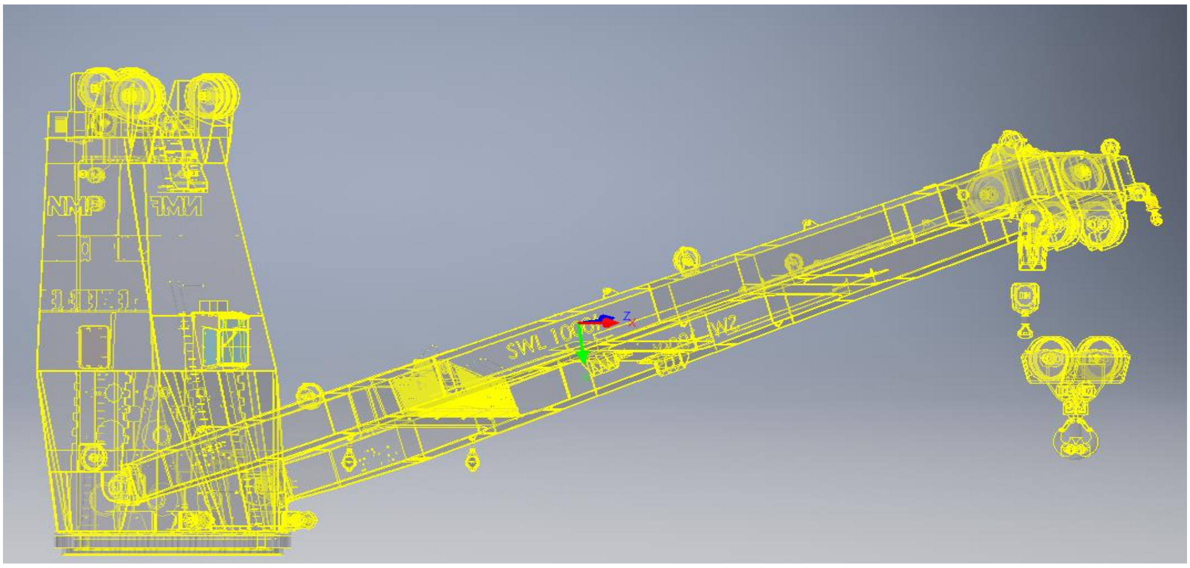


Figure 10: Plate and stiffening arrangement of crane structure

The edges of the boom are specially kept into consideration as they can be areas of the stress concentration points so they need to be rounded off. The curvature used on the actual boom was analysed and the same dimensions are given to the model. This enables us to locate the actual areas which would act as the points of the stress concentration. This part forms our first stage of the project. The pictures and the dimensions obtained from collecting all the structural drawings and filling the vacant spaces with the information given by the manufacturer helped me model the crane in drawing software. Once the first step of modelling in the AutoCAD and the inventor software is done, then we get a clear picture of the machinery how it actually looks. Now using the reference of the AutoCAD model the crane and the crane housing needs to be modelled. In order to do so it is not very simple. This model is now sent to the second stage, which is modelling of the crane and the boom in the RFEM software. This is finite element software and requires a plate and the shell elements to model the machinery.

The central portion of the boom consists of the connecting surface between the two boom arms. This is modelled and the two boom arms are then connected, it is ensured that the nodes match arms and members are well connected to share the loads which are suspended on the boom tip.

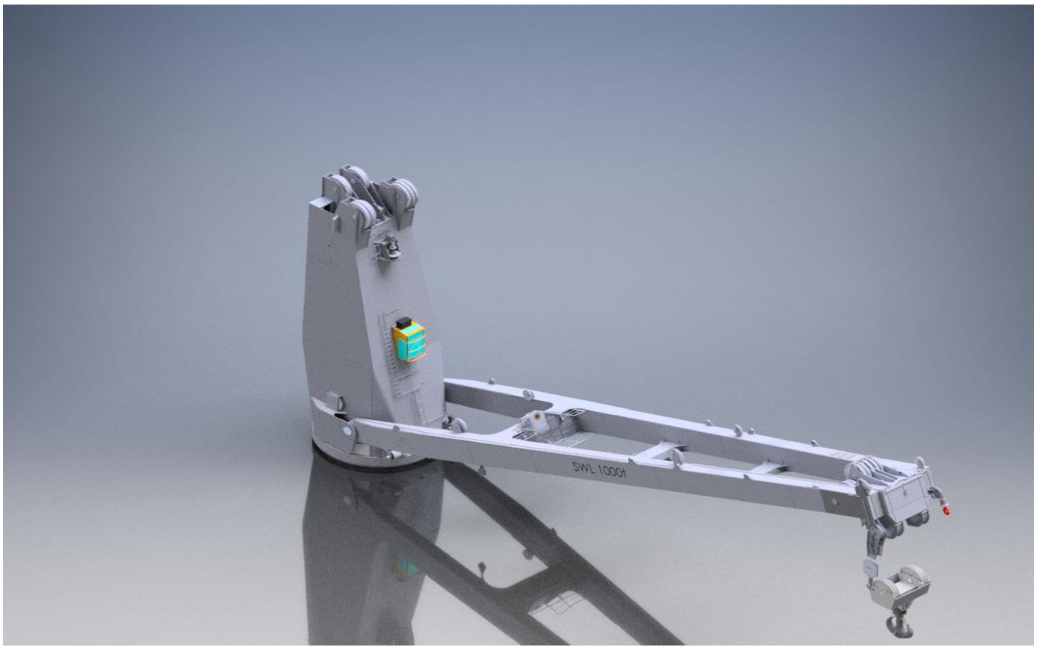


Figure 12: 3-D model of the crane

These two references and as well as the manufacturer's drawings are consulted to finally prepare a finite element model of the crane boom. The final model of the boom is presented in the drawing given below for further reference.

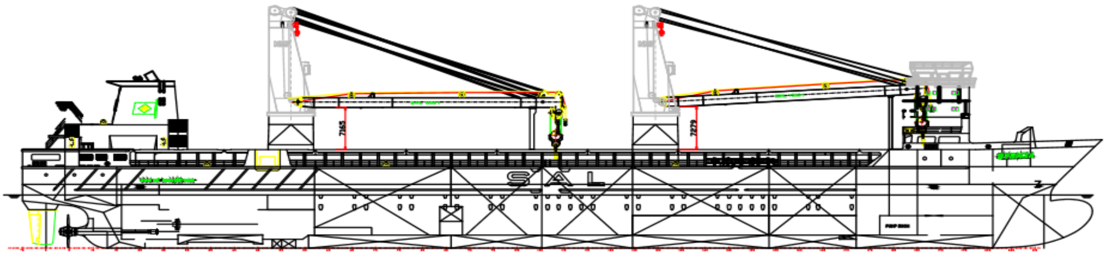


Figure 13: Position of the cranes on actual ship structure

[Ref: www.http://sal-heavylift.com/]

It is seen that for the reduction of the weight the plate thicknesses vary according to the areas which bear more stresses compared to the areas of less stress. It is seen that the surfaces are curved at most of the parts so in such places the quad angles are used while the areas of nodes lying in the same plane the flat surfaces are used. The crane housing has various cut outs for the back window and the operator's cabin. These are also modelled as they can contribute to stress concentration parts in the housing as it is seen from the model that there are very few stiffeners and most of the load of the crane boom and the crane wires are borne by the walls of the structure. These walls are made of steel of the grade S-355. So keeping all these factors into consideration the crane house is modelled. The areas of the maximum concern are seen to be the place where the boom attaches to the housing and the crane wires which are placed on the top and create a moment in the housing as well. These two weights and moments are imposed on the housing to form the net effect on the structure. The structural strength of the surfaces should be sufficient to bear the loads. This strength and the deflections of the structure determines how much load the crane can lift. As in our case as the crane is gigantic and is meant to lift the loads as large as 1000tons and the boom span extends to 37 meters. It is seen that the structural members are quite strong and provides a firm and stable supports for the loads.

The modelling of the housing is done very precisely in the finite element software. The number of the nodes and the point of application of the loads are specially kept in consideration as per the real case scenario. As per the design it is seen that the housing being strong and firm it does not face much of the stress problem. And only the areas of the concern are the top end, which supports the rope loads and the area of the corners and the notches for the cut-out of the crane operator and the back window for the inspection. These are large, cut outs and in the surface of the load bearing wall of the crane pose to be a matter of concern. It is seen that these parts after being modelled are ready to be imposed by the forces the crane is supposed to face during its normal loading conditions. This location on the housing are being analysed for the deflection and the stresses the areas of the concern are determined and are marked as the hot spots in the crane. Special care needs to be given to high stress areas every time crane is inspected during surveys for and crack initialization. These hot spots are the places which require more advance inspection methods to be employed.

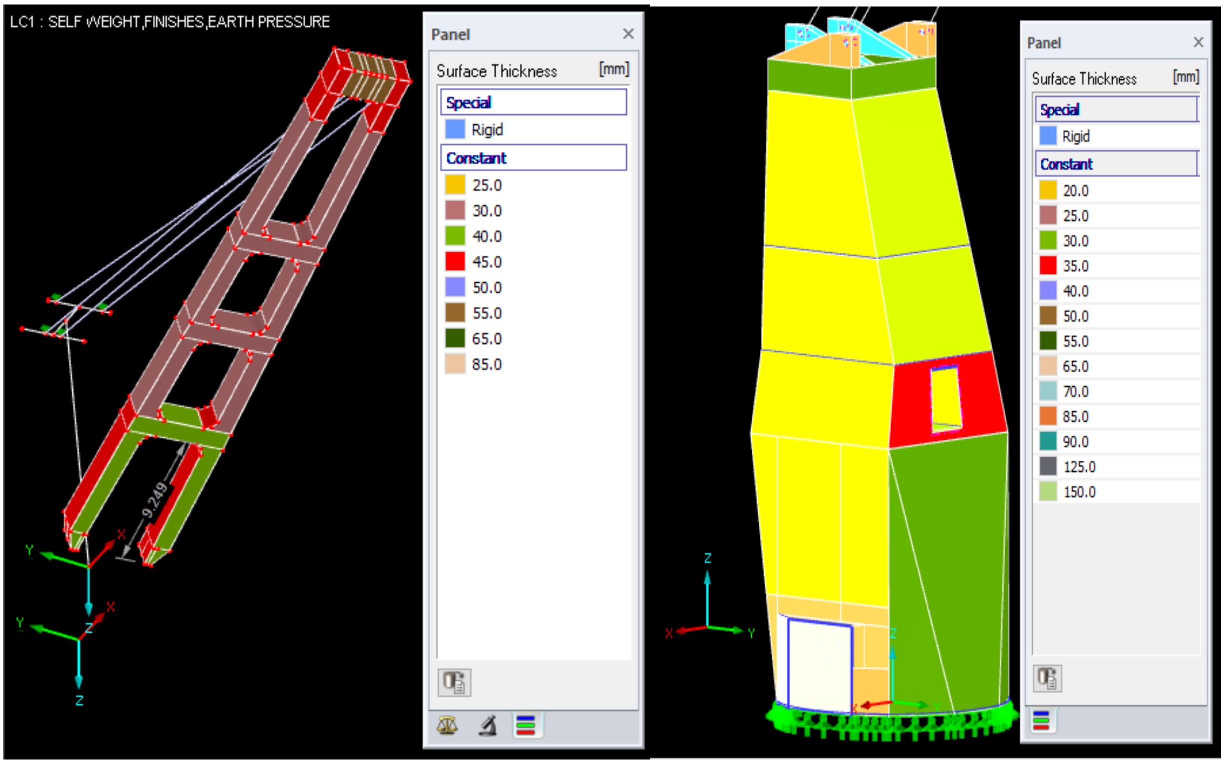


Figure 14: Various plate thicknesses for diffrent structural components

8. BOUNDARY CONDITION:

Now dealing with all the parameters it is seen that the boundary conditions for the problem set are fixed. These boundary conditions are:

1. **Boundary conditions for the fixation of the housing**: The housing of the crane is fixed in x, y and z coordinates but as the housing is sitting on a bearing which makes it freely movable in x, y and z rotational coordinates. Hence it is seen that the moments are kept free. The housing is free to rotate and change its support conditions in all the coordinate system. These changed geometrical conditions can be viewed by the figure given below.

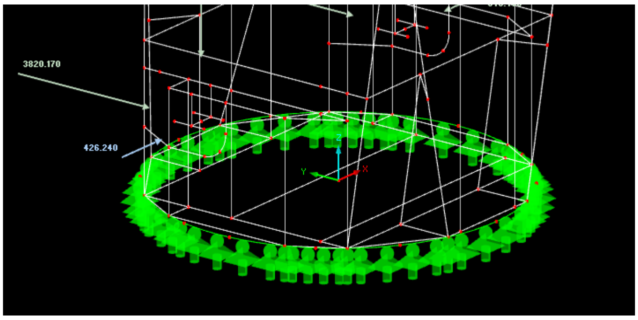


Figure 15: The boundary conditions for the housing support condition

2. **Fixation of the boom**: The boom is considered to be a large beam which is free to show deflections. The boom is the actual load transferring member of the crane and it needs to be analysed in detail. Hence the supporting conditions for the boom member are crucial for the analysis results. The boom tip is supported by the support wires which take a part of the boom structural load and the other part of the load is taken by the boom base. The boom tip wires are of specific stiffness and the young’s modulus which is present on the actual crane. While the base of the crane is hinged with freedom to move as a hinged joint in y axis. The support conditions can be viewed in the figure given below. The support wires as it is supported on the sheaves they have freedom of rotation motion in the X and Y direction, hence it is kept free in these axes. While the other motions are restricted. Same is the condition with the boom support as it is resting on the pin bearing it has freedom of rotation in the X and Y direction while the Z direction is held up because of the wire supports restricting its motion.

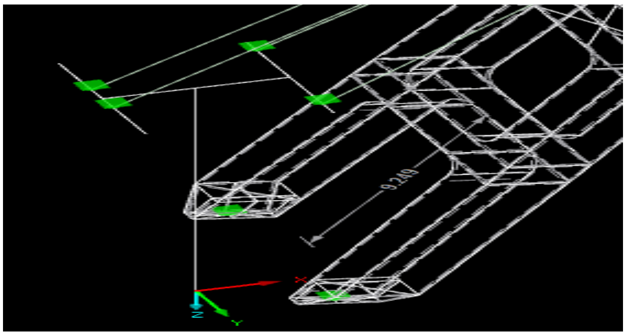


Figure 16: Boundary conditions for the boom supports

3. **Stiffness of the wires**: The wires of the crane are designed with the young's modulus as provided with the manufacturer. The wire thickness is kept as 80 percent of the nominal diameter and the length of the crane wires are also kept same as the actual crane at that boom angle and the outreach. For the finite element model the wires are used as the tension members as they are only capable to take the tension loads .The list of the crane wire diameters and the lengths are provided in the table given below.

Table 5: The Youngs modulus for crane wire at various outreach

Boom Angle	Outreach	Wire diameter Luff (m) Wire diameter Hoist (m)	Young's mod Luff Young's Modulus Hoist	Length of the Luff rope(m) Length of the hoist rope(m)
69.74	16m	115mm	102 000N/mm²	25.065
		115mm	94000N/mm²	21.065
54.04	25m	115mm	102 000N/mm²	28.613
		115mm	94000N/mm²	25.5
18.17	38m	115mm	102 000N/mm²	38.5594
		115mm	94000N/mm²	35.91

9. MESH GEOMETRY:

The finite element method is considered to be of the best analysis for the calculation of the stresses present on the complex geometry models and the case where the calculations are nonlinear and require complex analytical formulae for the calculation. In such situations the finite element modelling becomes really useful and need to be used for accurate and realistic results for the assignment. But the software has certain limitations and it is dependent on the user how efficiently he has handled the problem to calculate the results. The basic parameters on which the results of the analysis depend are basically based on the following parameters:

1. The boundary conditions chosen.
2. The mesh geometry chosen for the analysis.
3. Mesh size chosen for the analysis.
4. Type of the equation used for the analysis.
5. The software accuracy and stability.
6. Correctness of the model design.
7. Fineness of the meshing and the expertise of the designer.

The analysis of the appropriate mesh required convergence study. This study will be basically concentrated on the selection of the element type. A comparative study will be made and the mesh stress results obtained from the analysis will be compared to the actual results to give us the actual scenario to which mesh shape and the type need to be chosen for the further analysis. Also the size of the mesh depends on the plate thickness, which need to be meshed and the processing capability of the computer processor used.

The mesh type used forms a major concern for the analysis. Hence a study was done on the different types of the meshes available and the best suited for the given problem set.

It is to be noted that also the error in the displacements cannot be eliminated just by increasing the number of elements in depth. This phenomena is called the shear locking.

Figure 17: The shear lag effects on the elements

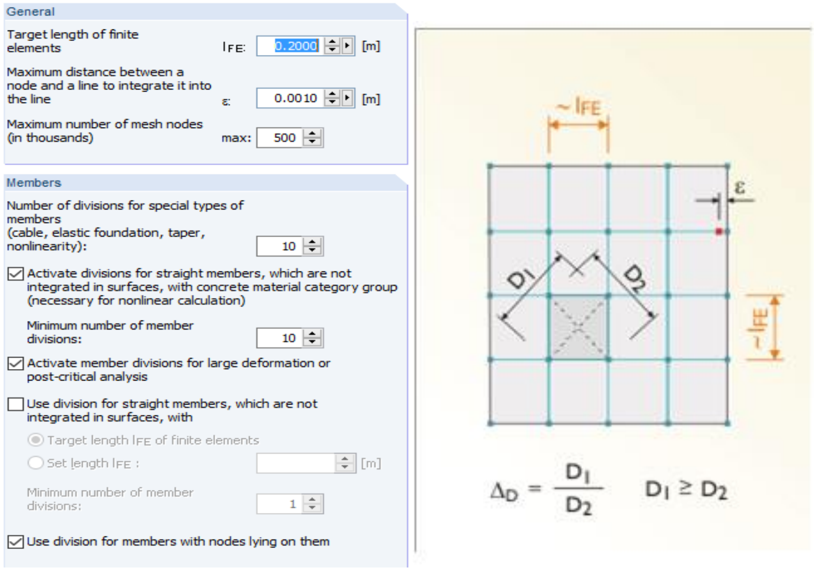


Figure 18: Defining the element size in finite element model

As in the modelling I have used shell type elements, it is necessary to define the shape function for the analysis and to prevent the shear locking as was shown in the previous text. So in order to obtain the correct stress results the shape function chosen is shown below.

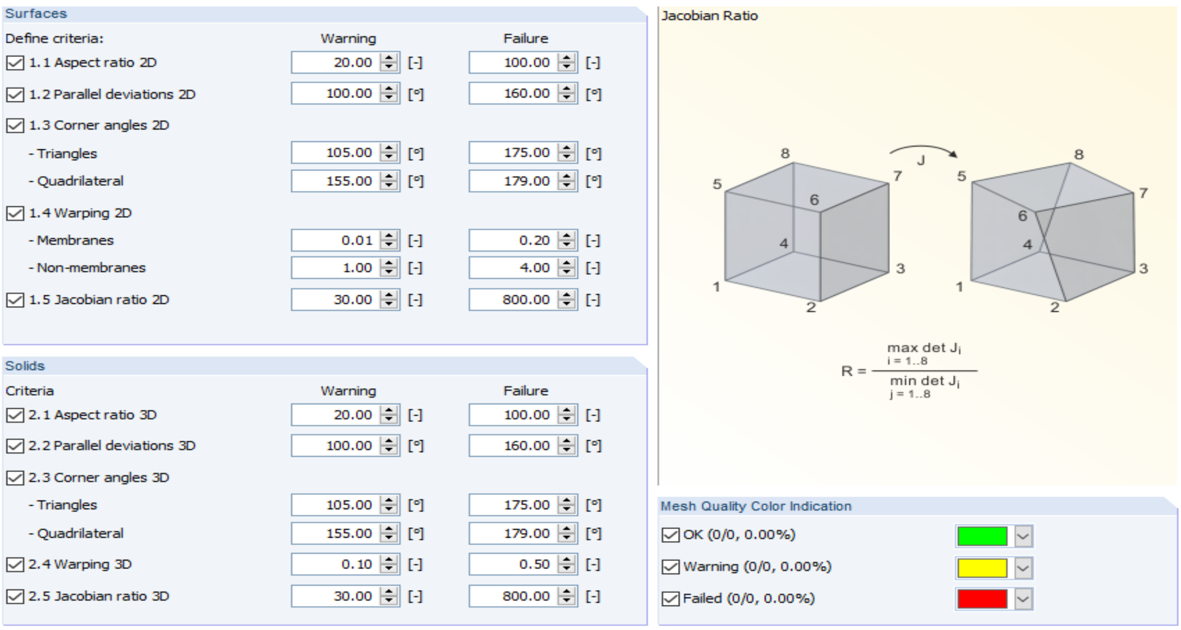


Figure 19: The shape function considered in RFEM

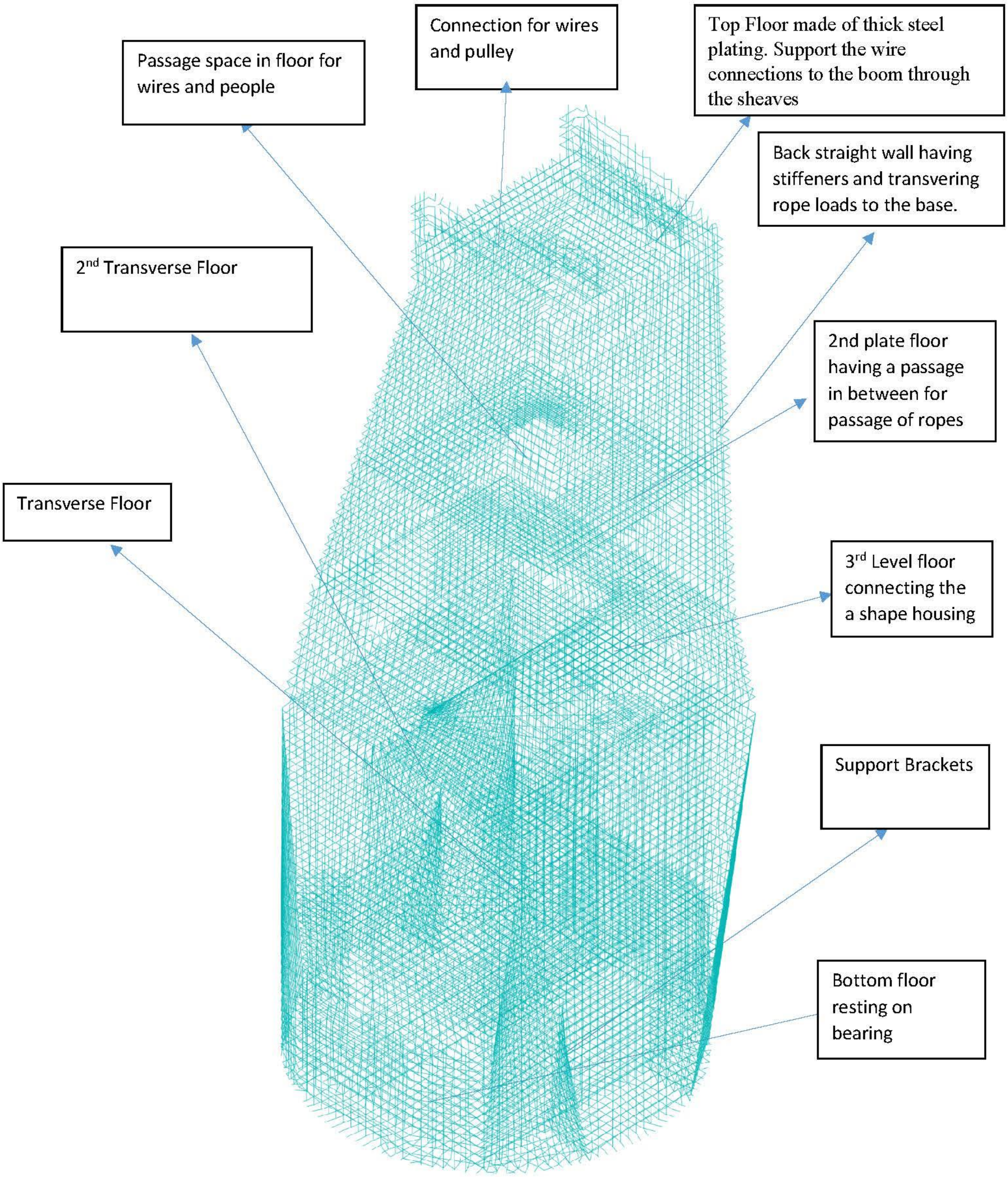


Figure 20: The mesh geometry and the size of mesh chosen for housing.

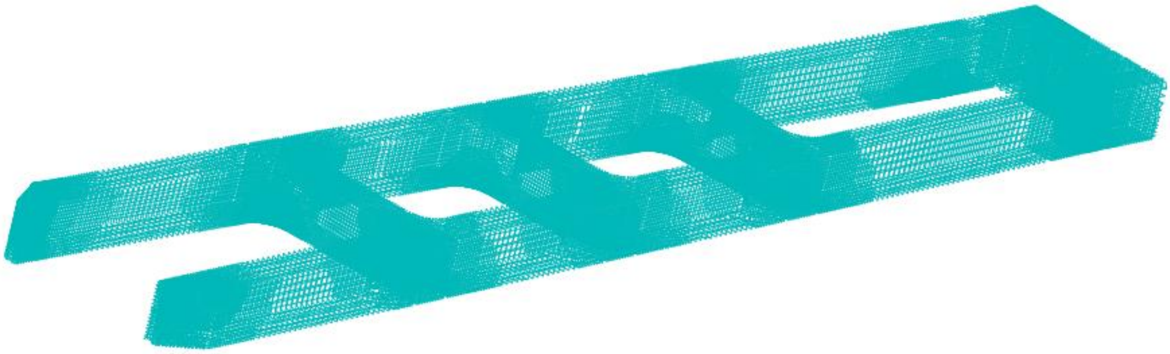


Figure 21: The mesh geometry and the size of mesh for boom

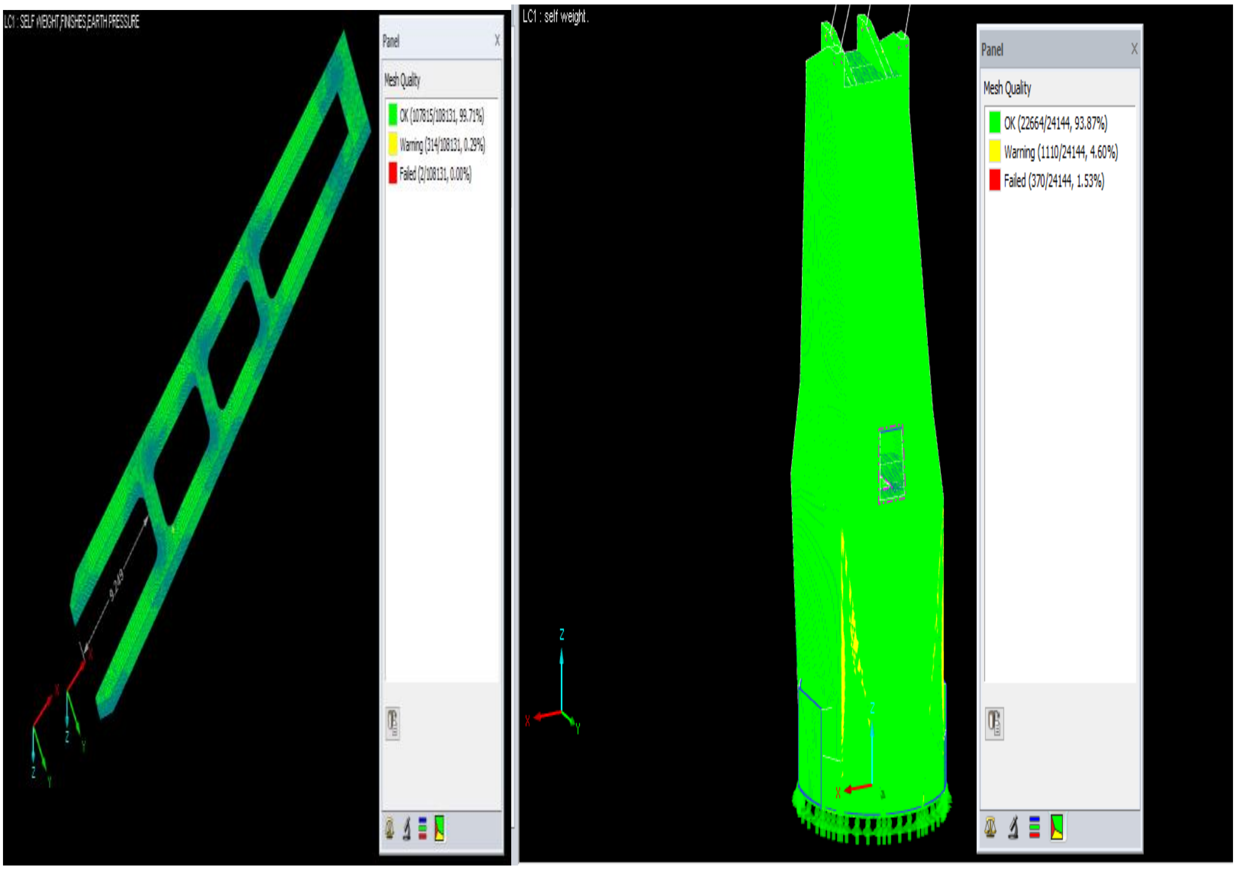


Figure 22: Detecting the flaws in the mesh of the model

10. STUDY OF THE CRANE LOADS

The lifting load regulations are defined in the DNVGL-2.22 lifting appliances under the subsections of section 8 and section 10. These two subsections help to define the regulations for the offshore crane. The parameters dealing with the structural strength of the crane. In order to perform safe cargo handling conditions, it is important to follow certain basic safety parameters. These are provided in the list given below:

- Mean wind velocity should not exceed 20m/sec.
- Safe working loads considered as the final limiting load on the structure.
- Visibility for the load handling by the crane should be like the daylight or equivalent.
- The trim of the ship should not exceed 2 degrees during the load operations.
- The heel should not exceed 5 degrees.
- The combination of the list and the heel gives us a combined effect of 5.4 degrees of inclination.
- For harbour conditions of cargo operations the sea conditions and accelerations are neglected as most probable loading conditions are without wave conditions and we are concentrating on loading and discharging operation by crane in confined waters.

Now keeping in consideration all the rule and loading, I have calculated and found out the loading conditions and the nonlinear effects being applied on the loadings. These parameters are imposed on the finite element model of our crane, which will predict the actual stress, deformations and the loading scenario of the crane and its components.
Study of the heavy lift crane during the loading and offloading conditions of the crane. In order to simplify the calculation and study the load curve of the crane and its components. The crane has been divided into 8 loading conditions with different outreaches of the boom and considering 3 different boom angles. These boom angles have their specific SWL at different boom angles which it can operate on. These outreaches and the load combinations are set as the benchmarks for the load analysis on the crane and its structural components. The three outreach and the boom angles have been represented in the figure given below. As the crane can slew in 360 degrees these have been represented in the form of circles. This forms our load curve, which needs to be further analysed for further calculations.

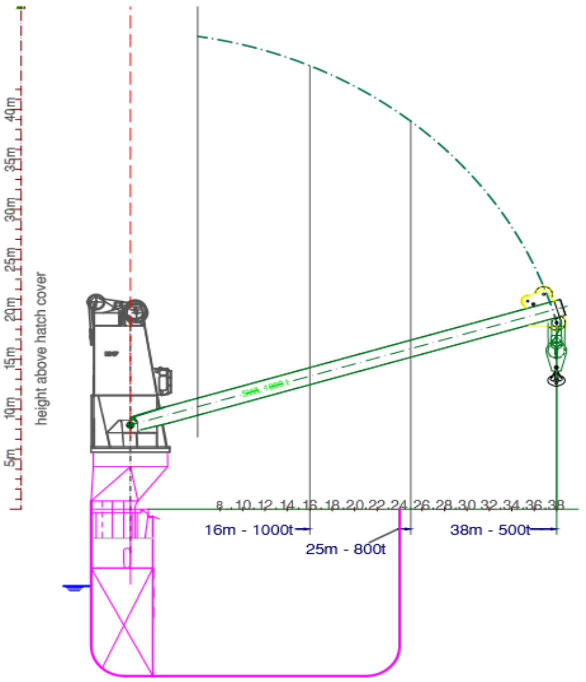


Figure 23: The load case description of the crane

Table 6: Loadings in different load cases

Loads +Wind

Load Case S.No.	Boom angle. (degrees)	load SWL (tons)	Outreach	Weight of Boom(t)	Total Weight	Force P(KN)	Force by Finite element(KN)
1	69.74	1000	16	152	1152	11301.12	11369.8
2	54.04	800	25	152	952	9339.12	9418.21
3	18.17	500	38	152	652	6396.12	6507
4	54.04	500	25	152	652	6396.12	6475.04
5	69.74	500	16	152	652	6396.12	6464.58
6	18.17	350	38	152	502	4924.62	5035.99
7	54.04	350	25	152	502	4924.62	5003.96
8	18.17	250	38	152	402	3943.62	4054.97

Now keeping these load cases as constant we distribute all the loadings on the crane within these load cases and count the number of load operation the crane has conducted for each of the load cases as per the data obtained from the load cell which gives us the loading cycles for 1.8 years of the operating period of the crane. This 1.8 years will be extrapolated for the life of the crane that is 20 years.

It is to be noted the acceleration and the dynamic factor will be added in code.

Table 7: Some important crane specifacions and abbreviations

Component	Weight	
Crane Jib	157	t
Crane House	152	t
Hook SWL 1000t	25	t
Hook SWL 60t	1.7	t
Ropes	29.3	t
Equipment	1.7	t
total	366.7	tons

WLL	Weight of load
Wta	Weight of Boom
WTk	Weight of Housing
WTs	Weight of Wire
Wth60	Cargo Acceleration force
Pswl	Forward acc of Boom
Pwta	Forward acceleration of wire
Pwtk	Forward acceleration of housing

Now if we combine the boom and the housing together and calculate the actual moment at the crane pedestal centre, then we will get following forces and the moments due to various loads being applied due to cargo and the structural weights. This final calculation of the moments is handled by the finite element software. But in order to show the extent of the forces and the influence of the different hoist velocities the analytical calculations have been performed. Considering the actual weights of the wires, structure, dynamic effects and the cargo loads on the crane are considered. It is to be noted that the centre of gravity of the different loads lies at different points and hence have different levers at different boom angles.

Table 8: Loadcase combination 1

SWL	800	T	outstretch	25m
Hoisting Speed	1.5	m/min		
Dead Hoist and wind load.				
	(F /KN)	Factor	Lever	Moment (KNm)
WLL	8093.25	1.05	27.722	235579.1303
Wta	1540.17	1.05	16.029	25921.75418
WTk	1491.12	1.05	-0.018	-28.182168
WTs	287.433	1.05	12.578	3796.098888
Wth60	16.677	1.05	27.769	486.2587937
Total WT	12000.0825			
Pswl	80.6		30.025	2420.015
Pwta	42.9		19.201	823.7229
Pwtk	35.1		6.677	234.3627
total P	158.6			
Axial force	11841.4825			
Radial force	1555.866			
Resultant moment				269233.1606

Table 9: Loadcase combination 2

SWL	500	T	Outstretch	38m
Hoisting Speed	2.2	m/min		

Dead Hoist and wind load.

	(F /KN)	Factor	lever	Moment (KNm)
WLL	5150.25	1.05	38.952	210643.165
Wta	1540.17	1.05	23.134	37411.8074
WTk	1491.12	1.05	-0.018	-28.182168
WTs	287.433	1.05	18.5	5583.38603
Wth60	16.677	1.05	39.273	687.703612
Total WT	8909.9325			
Pswl	57		9.266	528.162
Pwta	42.9		7.305	313.3845
Pwtk	35.1		6.677	234.3627
total P	135			
Axial force	8774.9325			
Radial force	1324.35			
Resultant moment				255373.789

Table 10: Loadcase combination 3

SWl	1000	t	outstretch	16m
Hoisting Speed	1.3	m/min		
Dead Hoist and wind load.				
	(F /KN)	factor	lever	Moment (KNm)
WLL	10055.25	1.05	19.225	202977.7903
Wta	1540.17	1.05	10.896	17620.77694
WTk	1507.797	1.05	-0.018	-28.4973633
WTs	287.433	1.05	8.402	2535.762669
Wth60	16.677	1.05	19.184	335.9281464
Total WT	14077.69335			
Pswl	80.6		36.166	2914.9796
Pwta	42.9		22.581	968.7249
Pwtk	35.1		6.677	234.3627
total P	158.6			
Axial force	13919.09335			
Radial force	1555.866			
Resultant moment				227559.8279

11. SIMULATION OF LOAD CASES

In order to analyse the stress data set of the crane a special approach is followed. In this approach the crane is modelled in the finite element software. This modelling is done in two parts. 1st is the boom section which is a long supported structural component of the crane which has one end loaded with the cargo loads and the other end which is resting on the bearing pin and the other end, which is being supported by the crane wires. While the other part is the housing. Housing is the vertical steel structure which gives support to the boom and as well as encloses the machinery components of the crane which consist of the winches, the wire reel and the six hydraulic winch driving mechanism. It consists of the hollow steel structure about 20 meters high from the ship deck and has the wire connections for the boom at its top end and the bottom end has a special connection of the boom using the bearing pin. The driving winches consist of the hydraulic mechanism which provides the luffing, slewing and the hoisting and lowering motions to the crane boom and the cargo resting on the boom tip. These two components combine to form our crane and will be discussed in the coming sections. The crane together is a complex part with different motions of the boom and the housing combined, hence in order to simplify the problem the crane has been divided into two components. The wires supporting the boom consist of one is luffing wire 2 in number and the other is the hoisting wire 2 in number. These two pairs of the wires support the top part of the boom. The correctness of the finite element results to the real case scenario is ensured by keeping in mind the following parameters which are matched with the real crane in order to give us the same behavior features as that present on the live crane. The features which are kept constant are:

- The material properties of the crane structure.
- The weld locations between the connecting plating.
- The dimensions of the crane to the real crane.
- The plating thicknesses at various locations of the crane.
- The length of the wire ropes at various loading conditions.
- The material properties of the wire.
- Dimeter and the young’s modulus of the wire materials.
- The boundary conditions and the type of the fixtures used for the connection of the crane housing and the boom.
- The position of the connection of the wire on the housing and the boom.
- The number of the nodal connections on the real crane and the model.
- The structural members supporting the structure,its locations and the material properties.

- The total weight of the actual crane housing and the boom with the actual structure weight.
- Special consideration of the curves and the rounding's provided in order to reduce the stresses on the real crane model.
- The connections between the structural components of the crane members with their specific locations and the connection types.

The load cases have been generated using a special analysis as had been discussed in the previous sections.

In the above scenario the load case is kept according to the Case 2 loading as provided in the Lloyds and the DNV GL guidelines as discussed in the sections above. As we recollect these guidelines ensure us to use the loads on the crane which comprise of:

- Cargo loads as specified by the load case table given above (maximum safe working load).
- The self-weight of the structural steel.
- The wind loads as calculated from wind speed of 20m/sec on the longer edge of the boom section.
- The dynamic loads of the lowering and the hoisting.
- The effects of 5 degrees of heel and 2 degrees of list. Which give us a cumulative inclination of 5.4 degrees.

Now these loads will be applied on the boom tip and the stresses and the deflections will be obtained from the boom section. Then the reaction forces obtained from the boom will be used as the loads in the housing in order to find the reaction and stresses at the structure of the housing. This stress value and the deflections will be closely studied to determine the hot spots and the critical locations in terms of the stresses. And finally we will use the stress values and the welds and the plate characteristics to determine the fatigue in the structural components of the crane structure. Hence, in order to go about with the determination of the stresses and the deflections of the boom and the housing section we will go about it in the detailed manner.

Load case 1.

The detailed explanation of the load case has been provided in this section. It is seen that in this case a load of 1000 tons is applied on the boom tip and the boom angle is kept at 69.74 degrees as can be seen from the table given above. This is the largest load which is equivalent to the SWL of the crane. In this load case scenario the boom is first studied for the linear analysis and then the results are compared with the nonlinear analysed. When it is seen to be correct in terms of the stresses and the deflections then this load case is monitored for the actual scenario that is analytic with the inclination of 5.4 degrees and the wind forces and self-loads and other dynamic loads on the crane structural components.

An application of the loads at the boom it is seen that the boom shows a deflection of 350.4 mm. This deflection value was then checked with the results of the finite element model of the boom provided by the manufacturer which showed a value of 350.3 mm. The results are very close with the variation of just 0.1% from this we can conclude that the model is seen to show the real case deflections and the results as has been shown by the actual crane. The picture depicting my results and the manufacturer finite element results have been provided below for viewing:

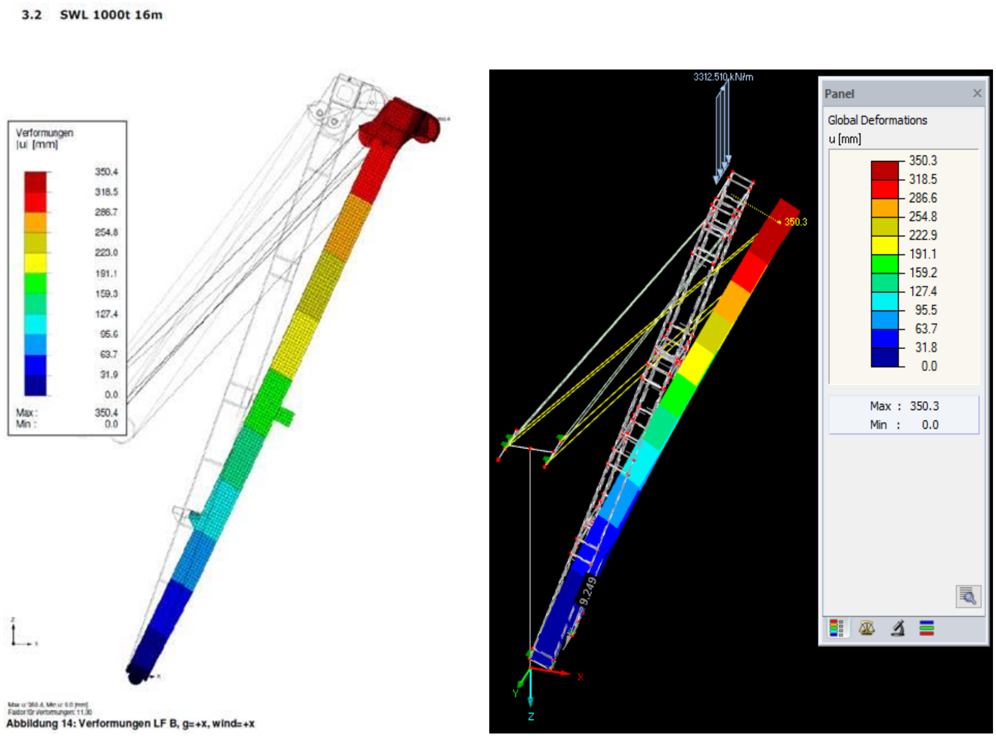


Figure 24: Comparision between the manufacturer and design FEM

Now moving forward with the analysis the load case is further analysed for the linear and the nonlinear analysis. Now using the linear and the nonlinear analysis the boom deflections and the stresses are seen to vary a little bit. The linear analysis shows a deflection of 360.6 while the nonlinear analysis shows of 364.1. There has been slight variation of about 0.0935%. And this variation is obvious as nonlinear analysis needs to account for the effects of the deflection on the member due to bending moment. And the shear deformations causing moments and more deflections. But the values are found under the limit and hence our results show fairly good resemblance to the real case scenario.

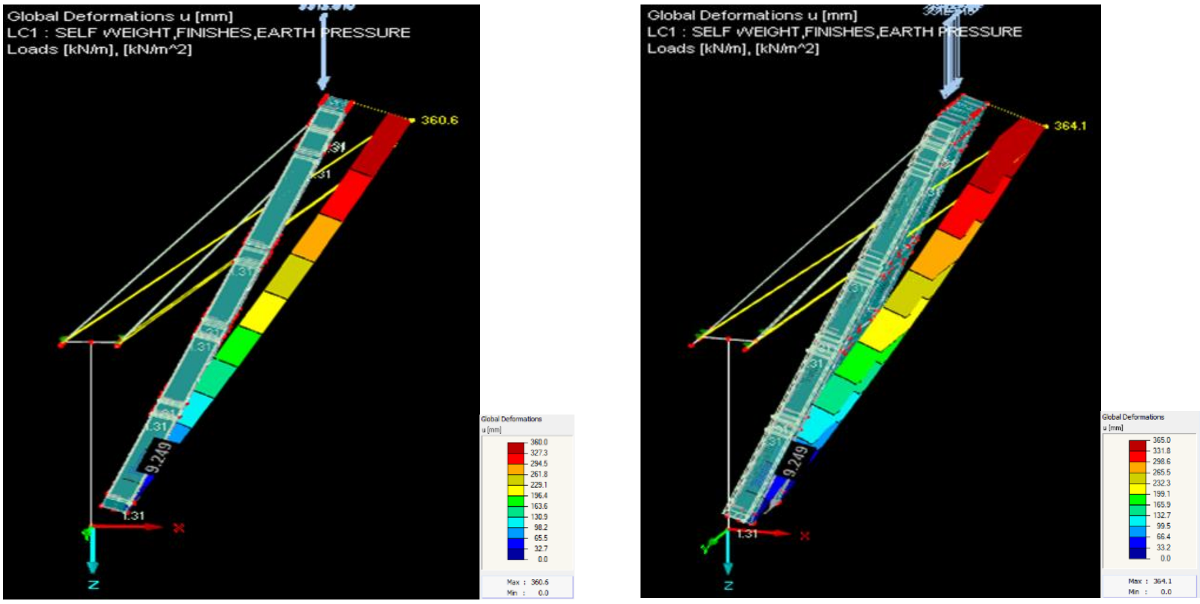


Figure 25: Linear Vs Nonlinear analysis

Similarly the boom section is then analysed for the actual scenario which is required for the extraction of the stress history at the nodal coordinates. This analysis considers all the parameters mentioned in the previous section of inclination and other effects. The results obtained from these analyses showed a deflection of 390.3 mm. This data and the stress tensor history is extracted for the further analysis. And needs to be tested for fatigue .

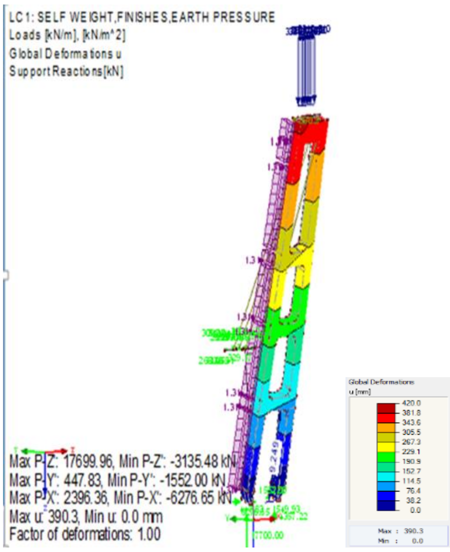


Figure 26: Non linear deflections of the boom with inclination of ship.

The reaction forces obtained at the connection points for the wire ropes and the bottom of the boom are imposed as the load cases for the housing section at the boom connection points there. The reaction force obtained from the boom analysis at the connection nodes of the wires at the housing (nodes 256 to 259) and the connection point of the boom at the housing bearing (Node 2 and 3) is as given below:

Table 11: Reaction forces from loadcase 1

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-1549.93	447.83	4357.22	0.00	0.00	-171.31
3	-6276.65	-1552.00	17700.00	0.00	0.00	-66.31
256	2396.36	322.55	-3135.48	0.00	0.00	0.00
257	2360.40	268.48	-3092.94	0.00	0.00	0.00
258	1440.18	106.41	-2100.54	0.00	0.00	0.00
259	1629.64	329.17	-2358.42	0.00	0.00	0.00
Σ Forces	0.00	-77.56	11369.80			
Σ Loads	0.00	-77.56	11369.80			

Now we move on to the housing part. The following forces on being applied to the top wire connection points and the bottom bearing pin connection the housing is seen to show certain deflection. In this case it is also analysed using the linear, nonlinear and then the load case with the inclination and all of the cases have shown different results as expected. The results of the final analysis with inclination are as shown in the figure given below:

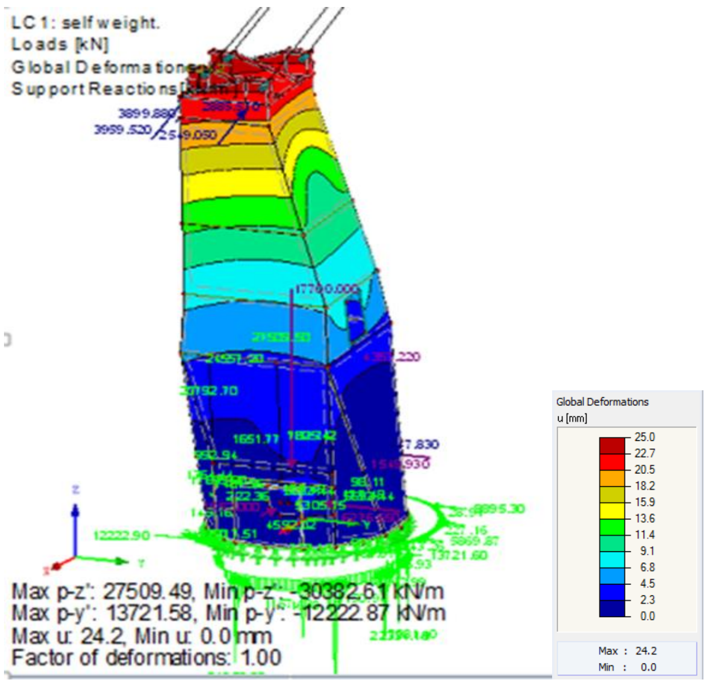


Figure 27: Deformation in the crane housing due to boom loads.

From the above load case figure it is seen that the housing reaction forces are not symmetrical (represented by the green lines on the base). This is basically because the inclination and the moment created due to the wind force. This results in different loads on the two legs of the boom resting on the housing pin. It is seen that there is also a tendency for the housing to move forward due to the horizontal force component acting due to the pull of the wires on the top of the housing of the structure. These forces result in a resultant deflection at the top with a value of 24.2mm. The deflection is seen to be maximum at the top and it gradually reduces to zero at the base. The housing is fixed on a freely rotating bearing at the base and the bearing is free to allow the rotations about x, y and the z coordinates. These rotations are handled by the special swirl bearing which supports the loads of the housing.

Now we need to observe the hot spots created in the structure due to these loads. These hot spots are again seen in the stress history results in the finite element model. The stability of the model is also seen using the RStab feature of the software which shows us the critical locations in terms of the stress. The results are as given below:

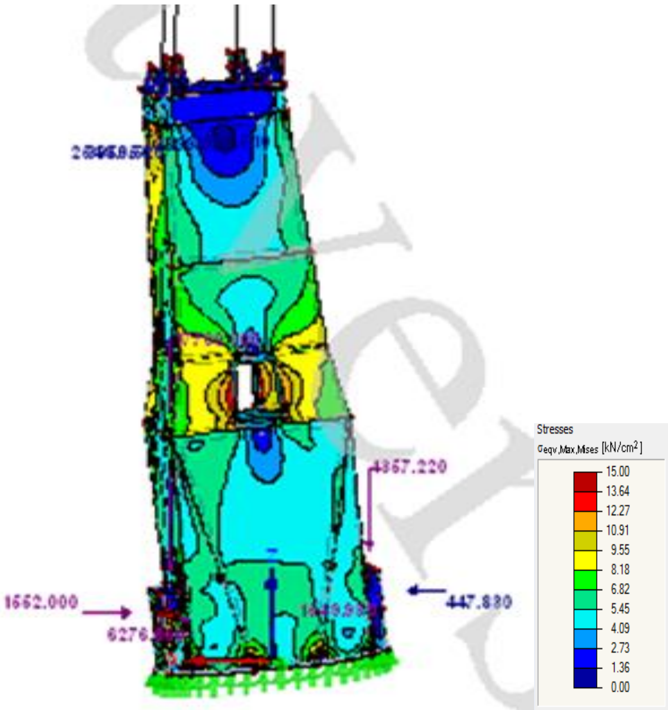


Figure 28: Stress in the crane housing and hot spots.

The structure shows fairly good acceptance to the loads of the cargo and is seen to sustain the adverse of the loading scenario. This is the maximum load case is seen to pass the test and the stress results will be analysed in the coming sections of the fatigue test for the structure. It is seen that this load case is seen to occur only a limited time in the life of the crane. And has a very little contribution to the fatigue life.

The hot spots observed in this load case almost resemble the loaded sections in all the subsequent load cases. The plating of concern having high stresses is seen at the locations of:

Boom section hot spots:

- The top plate section supporting the wire sheaves.

- The curved sections of the boom inner plating. Having connections of the boom arms.
- The lowermost plate having the connection of the boom to the bottom bearing connection.

Housing hot spots:

- The top plate section supporting the wire loads.
- The forward most face plate.
- The forward plating having the operator window cut out.
- The back plate having window cut out is shown to have stress points along its top and side edges.
- The bottom most plate supporting the housing loads.
- The plating supporting the bearing boom pin load.

The areas shown in green colour represent plate sections with hot spots:

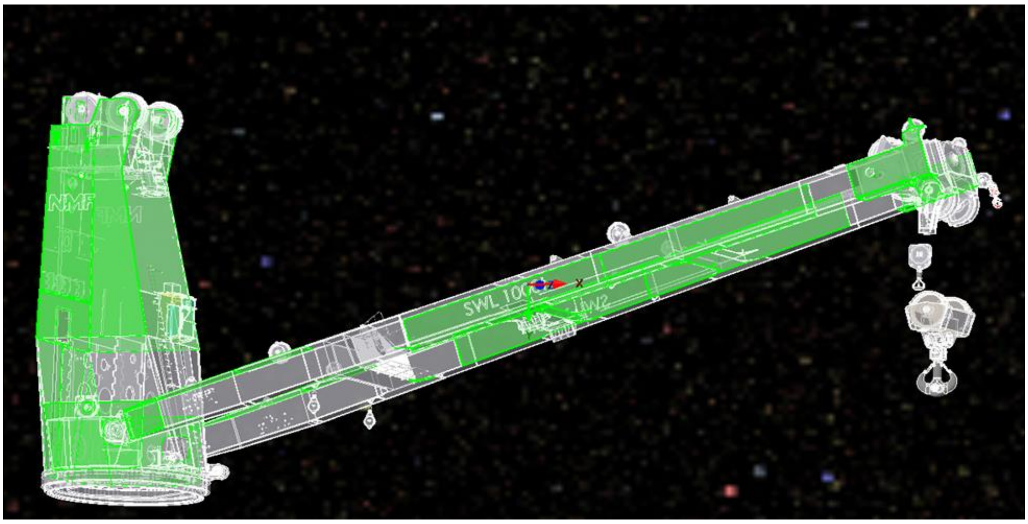


Figure 29: Hot spot location of the crane structure

Load case 2:

Now we move on to the next load case which comprises of the cargo loads of 800 tons and the boom angle of 54.04 degrees. This boom angle forces is observed and it is seen that the lateral deflection of the boom is seen to be 286.8mm for the linear analysis, and 288.4 for the non-linear results. These analyses are compared simultaneously in the figures given below. Similarly as in the case one, the reaction forces at the connections are analyzed and are imposed as the loading conditions on the housing to obtain the results. The final load case with the inclinations and the required static and the dynamic loads have been provided below. This analysis shows slight variation to the results without inclination due to the uneven distribution of the loads at the two legs of the crane boom.

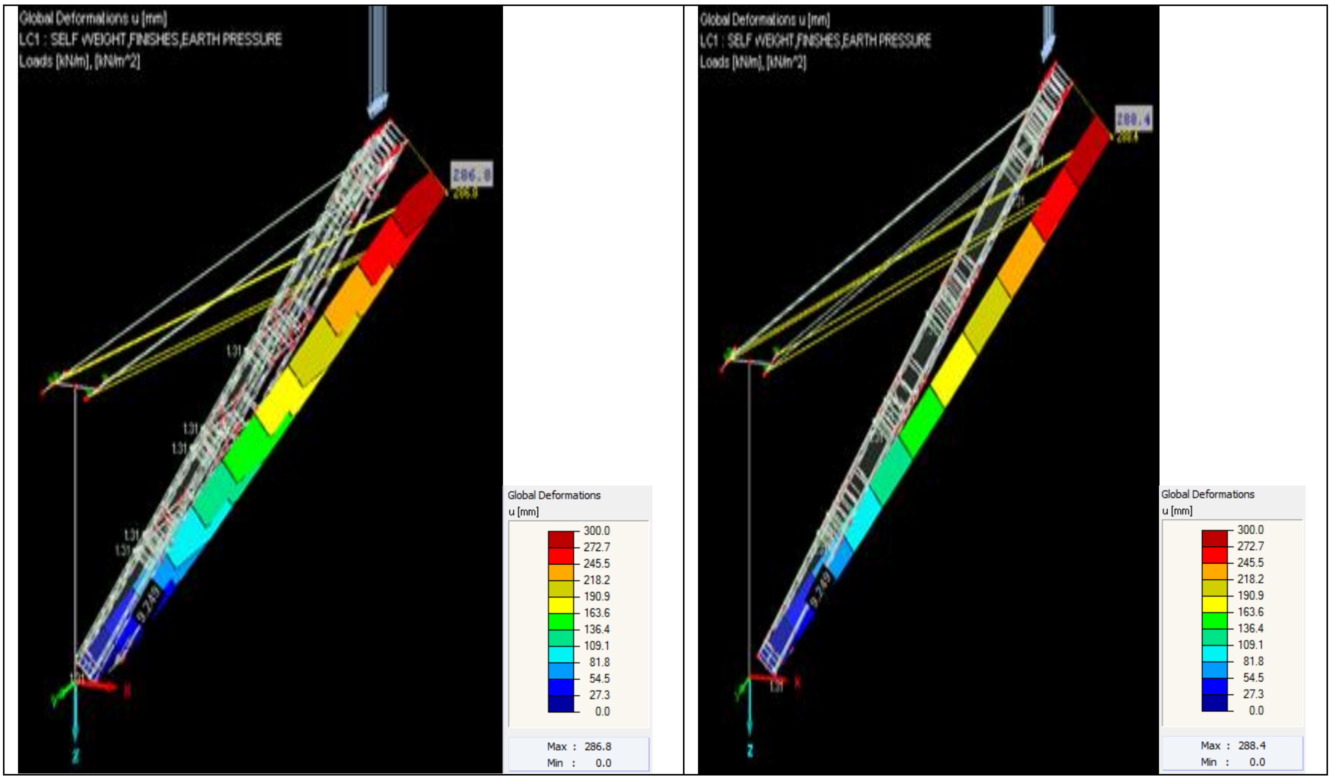


Figure 30: Linear Vs Nonlinear analysis for Boom

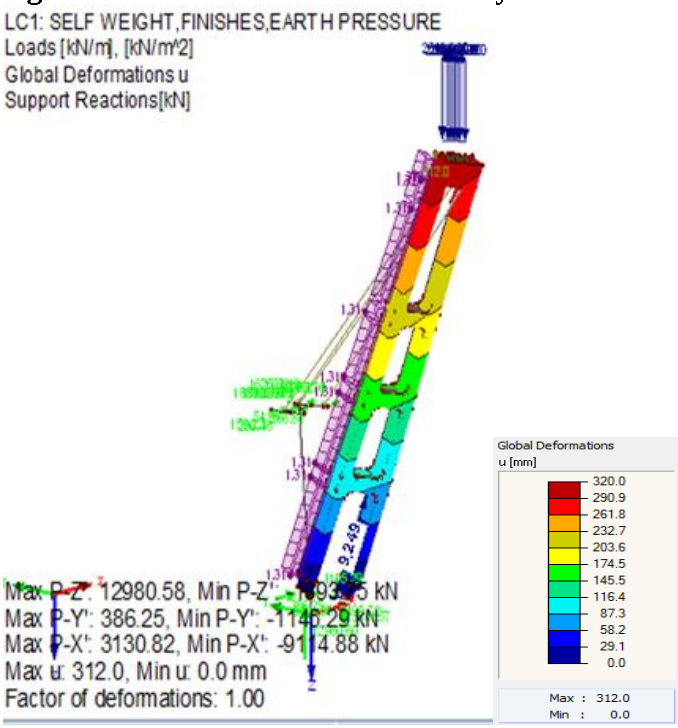


Figure 31: Analyses with inclination

The reaction forces obtained at the connection points are as shown in the table:

Table 12: Reaction forces from loadcase 2

Node No.	Support Forces [kN]			Support Moments [kNm]		
	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-2490.72	386.25	3541.57	0.00	0.00	-38.61
3	-9114.88	-1145.29	12980.60	0.00	0.00	-5.12
256	3130.82	200.46	-1893.75	0.00	0.00	0.00
257	3113.91	159.22	-1887.25	0.00	0.00	0.00
258	2607.12	54.98	-1625.32	0.00	0.00	0.00
259	2753.76	266.81	-1697.62	0.00	0.00	0.00
Forces	0.00	-77.56	9418.21			
Loads	0.00	-77.56	9418.21			

As in the previous case these forces are applied to the nodes connecting the boom to the housing to obtain the deflection and the stressed, hot spots on the housing as shown in the figure given below:

The hot spots and the stress history were further analysed and the data set obtained has been used for the study of the fatigue on the members. The stress data set for the nodal history is attached in the appendix for reference.

Load case 3:

Similarly as the above load case for the linear, nonlinear and the load case with the ship inclination of 5.4 degrees the boom is analysed for this set of the load case. It is seen that the lateral deflection for the boom linear case is: 305.6mm in the nonlinear load case is: 307mm and for the boom with the inclination is found to be: 318.2mm. The three load cases are given below for further review. It is seen after observing the reaction force that as the boom angle is inclined so the forces acting in the x direction are greater than the resultant force in the z direction. This load case is special in terms of the maximum outreach of the boom. From the above case it is seen that this load case will create maximum deflection of the housing in the x direction. Although the boom is loaded with just 500 tons of the load still this creates the limiting case for the housing structure. The housing structure is seen to lift loads no more than 500 tons when the boom is at 18.17 degrees. Considering this condition the capacity of the crane is restricted but it is seen at this angle the boom arm outreach is 38 meters which gives us a radius span of the boom of 38 meters in 360 degree. This span and the lifting capacity makes this crane special in its type and among the cranes which have the combination of outreach and span to this extent.

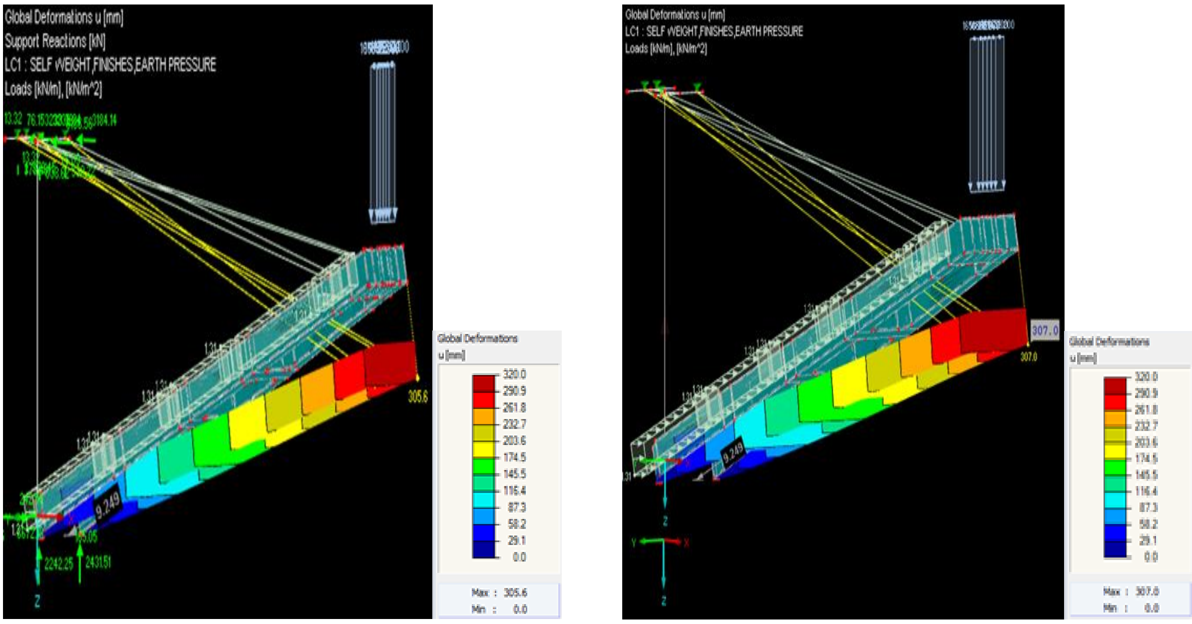


Figure 32: Linear vs Nonlinear analysis for loadcase 2

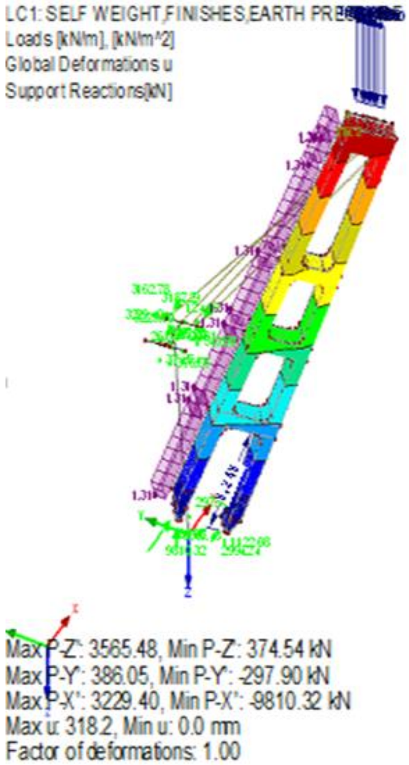


Figure 33: Analysis with inclination

The reaction force data set is given below for the further analysis. In this data set, it can be easily seen that the x direction forces in this load case in enormous and due to the outreach of the boom a great difference in the force distribution. This forms the basis of the load on the housing where it will try to pull the housing in the X direction, creating maximum deflection of 34.6 mm in the forward direction.

Table 13: Reaction forces from boom loadcase 3

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-2994.24	386.05	1122.08	0.00	0.00	445.33
3	-9810.32	-297.90	3565.48	0.00	0.00	313.15
256	3225.10	-20.81	376.55	0.00	0.00	0.00
257	3229.40	-47.28	374.54	0.00	0.00	0.00
258	3162.77	-124.15	525.29	0.00	0.00	0.00
259	3187.29	26.54	543.58	0.00	0.00	0.00
Forces	0.00	-77.56	6507.52			
Loads	0.00	-77.56	6507.52			

Since this load case forms a critical loading for the housing hence the results are compared with the finite element data set provided by the manufacturer. It is seen that the linear case when compared shows a deflection of 46.9 while the manufacturer finite element shows deflection of 61.3. This shows a variation of 19%. Although our deflection is found to be lower than the manufacturer model it is seen that the behavior of the stress distribution and the nature of the deflections are similar for both cases.

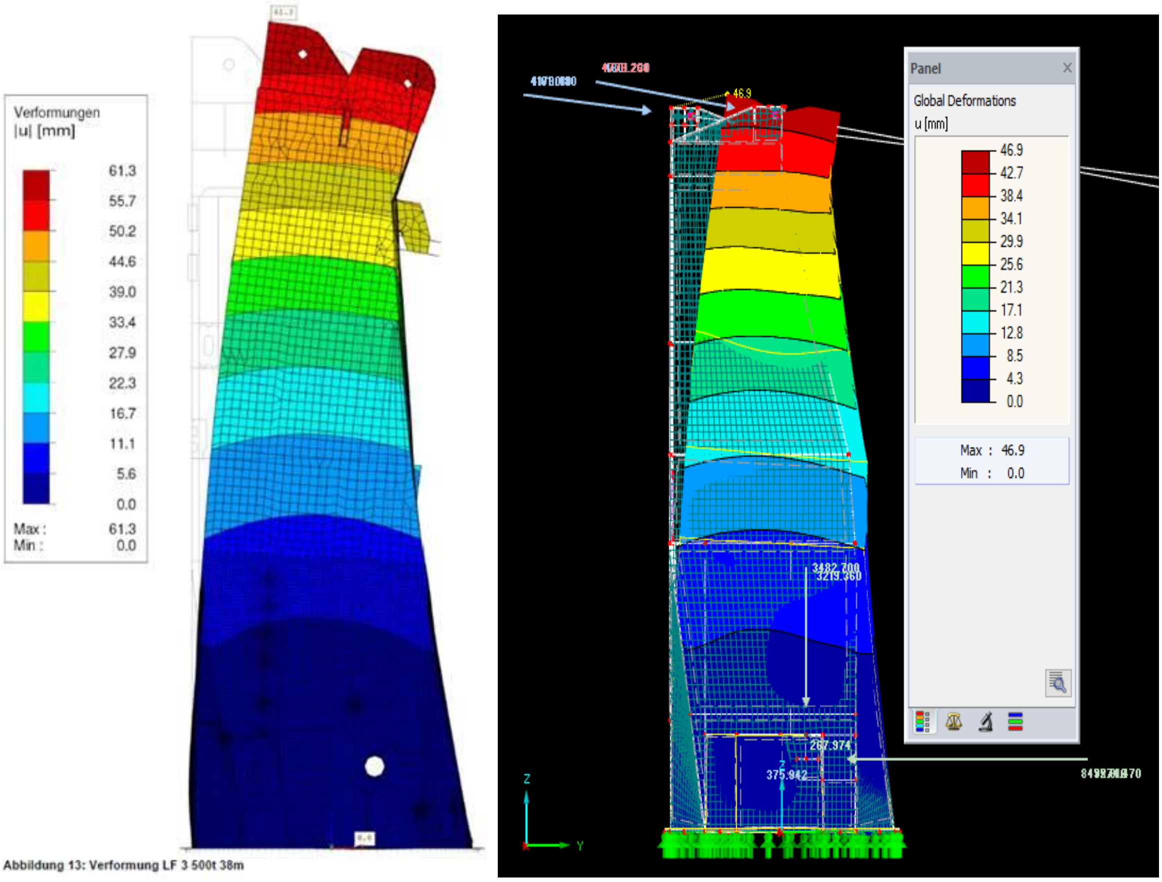


Figure 34: Comparision of housing deflection between manufacturer FEM and design FEM

The above load cases were judged for the loads applied on the housing and the housing deflection and the stress data set are as given in the figure below:

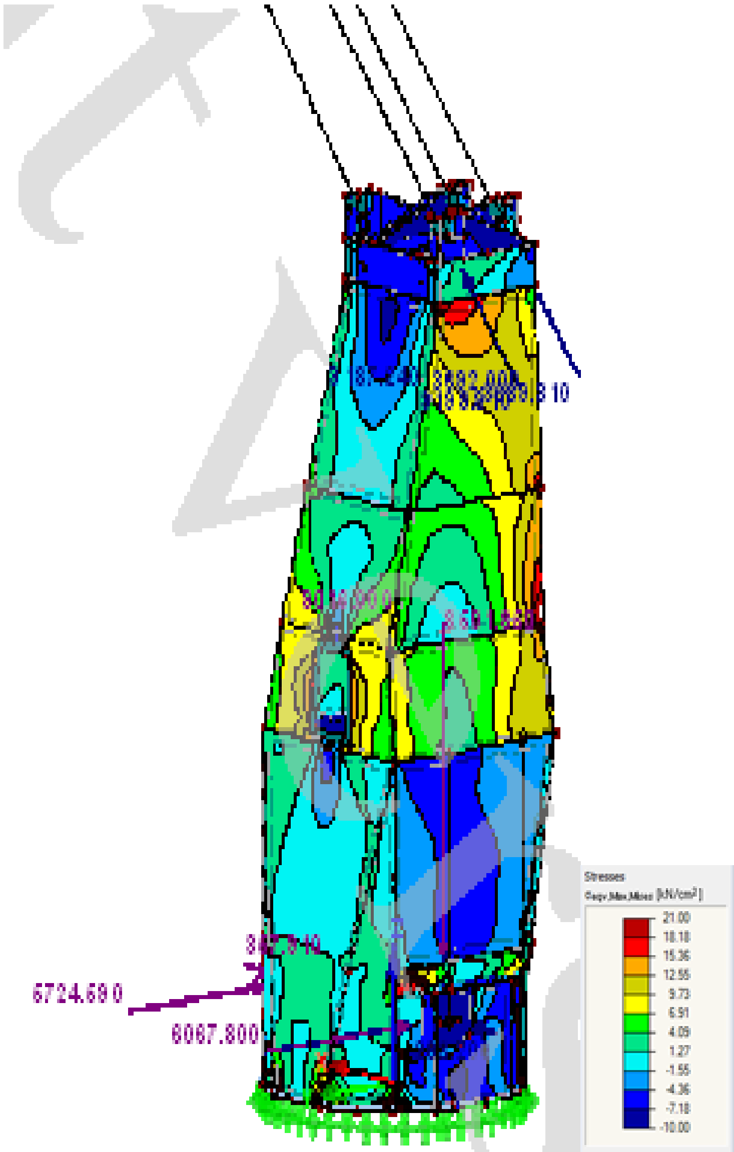


Figure 35: Deflections and stress analysis

From these results it is seen that the deflection for the housing is seen to be 34.6mm. The stress in the housing is determined. It is seen that this results comply with the structural requirements. This load case experiences a large number of operating cycles during its life and hence contribute significant amount to the fatigue life of the component. The stress data sets are loaded in the appendix and used in the MATLAB code for the analysis of the fatigue.

Load cases 4 to 8:

It is seen that the load cases from 4 to 8 are the repetition of the boom angles and the outreaches from the first three load cases only the boom loading is varied in order to obtain intermediate values of the loads. These load cases are not that severe as the first three load cases it is usually lower loading of the crane boom. In order to explain all of those I have created a cumulative study of all the load cases. It represents the linear and the nonlinear boom and the housing deflections at all the load cases and then gives us the load case comparisons when the boom is at an inclination of 5.4 degrees. The stress history of all these load cases is retrieved and are tested for the fatigue analysis in the later sections. The cycles of the operations for the crane are counted in the earlier sections are utilized for the fatigue calculations.

Load Case 4

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-4052.42	177.40	5826.44	0.00	0.00	-26.74
3	-3709.22	-252.94	5340.76	0.00	0.00	-1.76
256	2072.78	12.92	-1240.08	0.00	0.00	0.00
257	2075.02	-13.78	-1241.44	0.00	0.00	0.00
258	1812.01	-68.94	-1108.47	0.00	0.00	0.00
259	1801.84	67.78	-1102.17	0.00	0.00	0.00
Σ Forces	0.00	-77.56	6475.04			
Σ Loads	0.00	-77.56	6475.04			

Load case 5

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-886.98	215.25	2655.35	0.00	0.00	-87.97
3	-3340.62	-847.32	9565.95	0.00	0.00	-35.67
256	1287.70	173.32	-1680.26	0.00	0.00	0.00
257	1268.61	144.30	-1657.71	0.00	0.00	0.00
258	786.05	58.08	-1142.06	0.00	0.00	0.00
259	885.25	178.81	-1276.69	0.00	0.00	0.00
Σ Forces	0.00	-77.56	6464.58			
Σ Loads	0.00	-77.56	6464.58			

Load Case 6

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-2290.16	275.63	952.32	0.00	0.00	303.62
3	-7223.96	-229.95	2714.30	0.00	0.00	212.83
256	2382.30	-15.37	282.18	0.00	0.00	0.00
257	2384.83	-34.92	280.62	0.00	0.00	0.00
258	2363.88	-92.79	396.37	0.00	0.00	0.00
259	2383.11	19.84	410.20	0.00	0.00	0.00
Σ Forces	0.00	-77.56	5035.99			
Σ Loads	0.00	-77.56	5035.99			

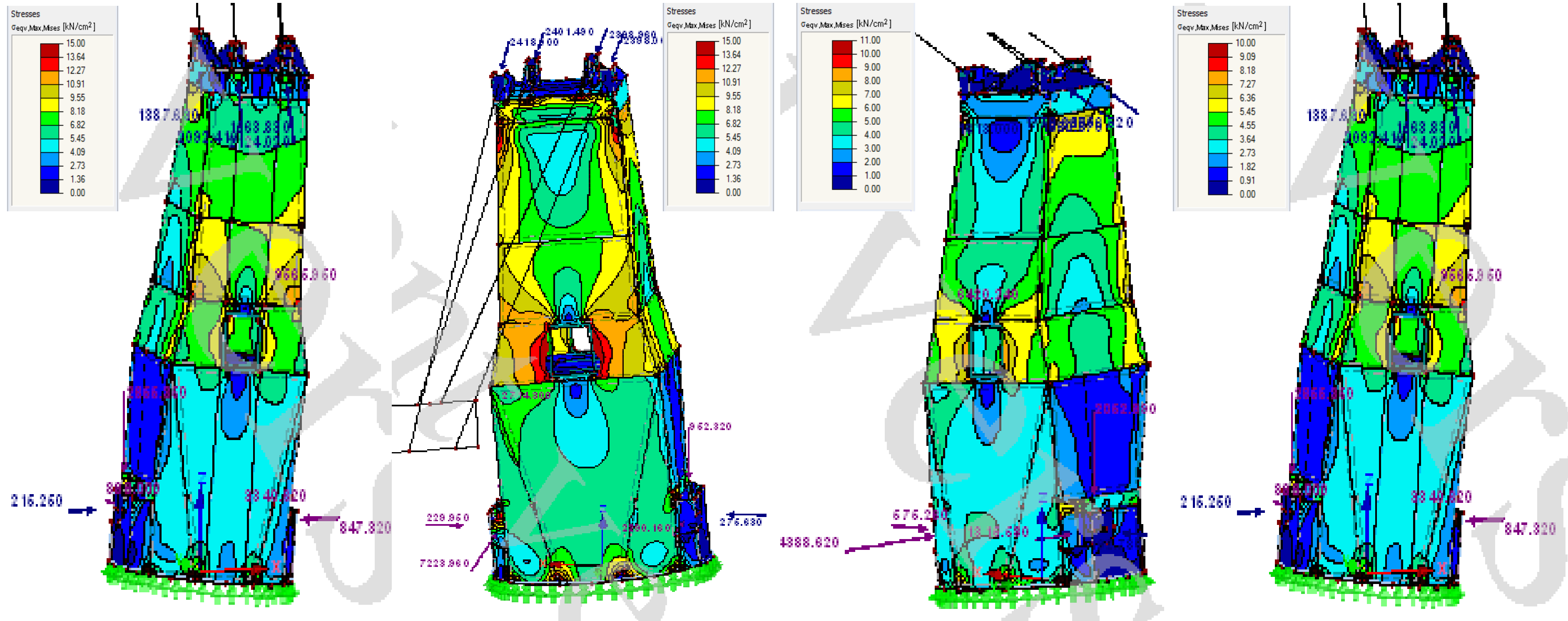
Load Case 7

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-1315.58	162.78	2052.99	0.00	0.00	-18.82
3	-4388.62	-575.26	6420.24	0.00	0.00	-6.24
256	1520.52	97.36	-913.85	0.00	0.00	0.00
257	1511.88	77.31	-910.44	0.00	0.00	0.00
258	1300.94	27.43	-805.56	0.00	0.00	0.00
259	1370.86	132.82	-839.62	0.00	0.00	0.00
Σ Forces	0.00	-77.56	5003.76			
Σ Loads	0.00	-77.56	5003.76			

Load Case 8

Node	Support Forces [kN]			Support Moments [kNm]		
No.	$P_{X'}$	$P_{Y'}$	$P_{Z'}$	$M_{X'}$	$M_{Y'}$	$M_{Z'}$
2	-1820.78	202.01	839.15	0.00	0.00	209.15
3	-5499.73	-184.65	2146.85	0.00	0.00	145.95
256	1820.43	-11.75	219.26	0.00	0.00	0.00
257	1821.79	-26.67	218.01	0.00	0.00	0.00
258	1831.29	-71.89	310.43	0.00	0.00	0.00
259	1847.00	15.38	321.27	0.00	0.00	0.00
Σ Forces	0.00	-77.56	4054.97			
Σ Loads	0.00	-77.56	4054.97			

Stress history of Housing with inclination 5.4 degree



Study	Load Case 5	Load Case 6	Load Case 7	Load Case 8
Max Stress	15.10KN/cm2	15KN/cm2	11KN/cm2	10KN/cm2
Load	500 tons	350 tons	350 tons	250 tons
Boom Angle	69.74 degree	18.17 degree	54.04 degree	18.17 degree

12. HOT SPOT ANALYSIS:

The detection of the hot spots in the structure forms quite an important part in the structure analysis. It is important to locate the areas of high stress concentration points so that it becomes easier to locate the crack initialization in the structure. The plating's with more stress are the ones we will be more concerned with during the survey. So using the stress data set we had obtained using our finite element model we go about locating the areas of high stress points in the structure. As the structure is divided into the boom part and the housing part so we go about the analysis in two steps.

Boom hot spots

The boom tip area as we know is supporting the cargo loads and as well as the pulling force of the rope sheaves. It causes areas of stress concentration in the wire connection areas on the boom tip. In order to deal with high stresses at boom tip a thicker plate is provides the loads from the cargo are then spread on this plate which are then transmitted to rest of the boom structure through the boom arms and the wire ropes.

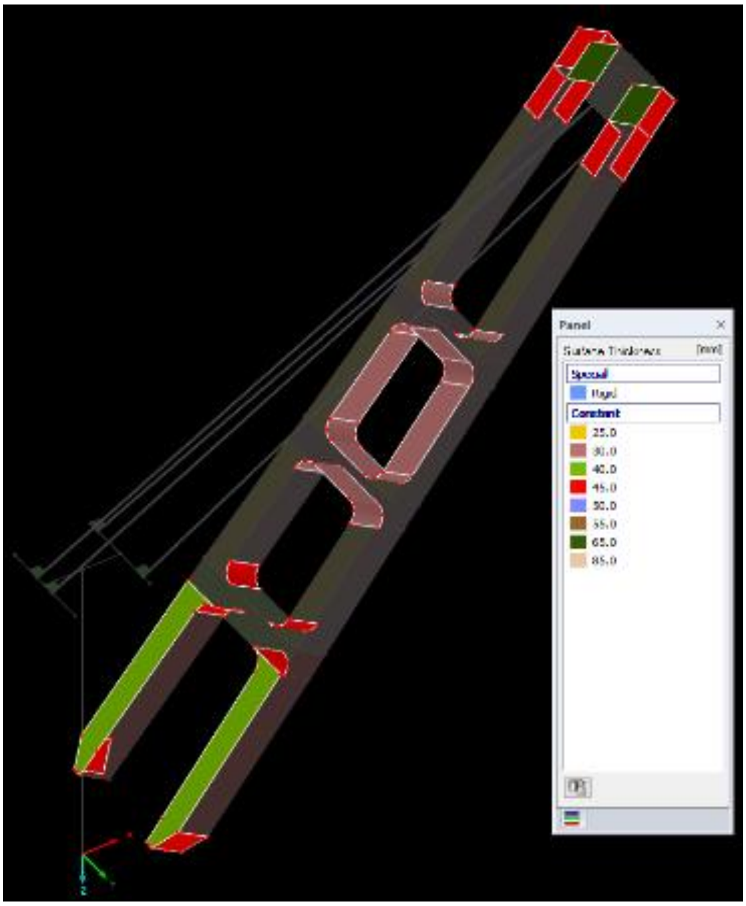


Figure 36: The concerned plates for the hot spots

The force transmits from the boom arm to the boom pin bearing. The deflection caused on the boom tip reduces the final forces as it is transmitted to the rest of the boom areas. The characteristics of the model are seen to be the same.

The loads obtained at the boom pin and the wires, then apply to the housing section of the model. It is seen that few places represent areas of hot spots in our finite element model. These places of concern are.

- The forward section of the boom tip plate.
- The inside curved edges of the boom section.
- The front plate of the boom which reaches up to the boom pin.
- The side plates connecting the boom to the boom pin.

The detailed hot spots of the structure are represented in the figure 38 of this section.

Crane housing hot spots

The housing of the crane section is also seen to possess areas of large stresses on the plate sections. These plates showing the maximum stresses are highlighted below in the figure for easy location. This includes the forward plate. The areas around the window cut outs and the back wall of the crane housing, which carries the wire loads and transfers it to the housing foundation structure. The following areas which were discussed above can be seen in the figures given below as missing sections of the plating's.

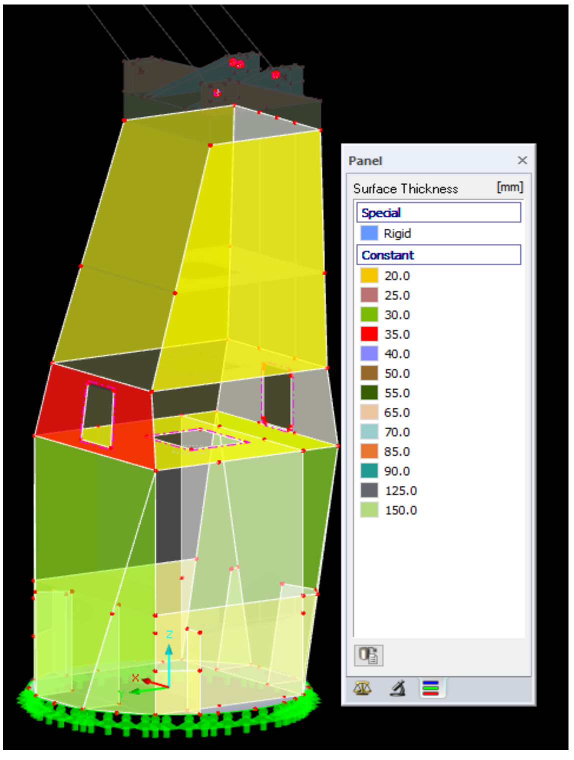


Figure 37: Plate thicknesses and hotspot shown by missing plates

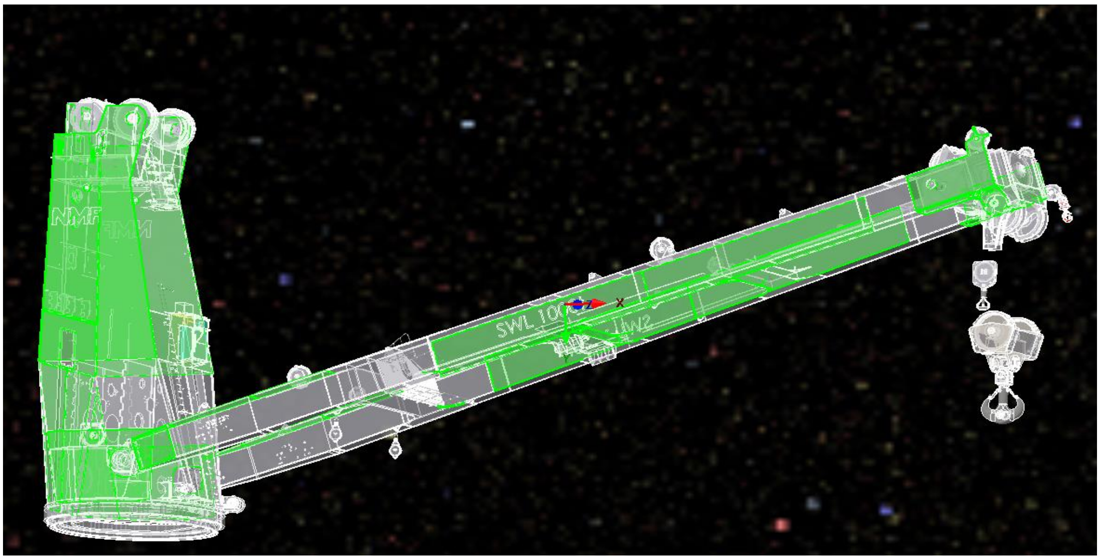


Figure 38: The hot spot analysis on the whole crane structure plating

The stress data set obtained after the analysis of the structural parts of the crane are provided in the appendix. It is seen that the stress values of the structural components need to be analysed and certain corrections need to be made on its parameters in order to enable us to do the calculations for the fatigue on these parts. The corrections and the modifications on the values of the stress which need to be made on the values are explained in the sections given below.

The static analysis of the stresses needs to be converted to the cyclic stress history in order to calculate for the fatigue. It is seen that the crane loading cycle consists of lifting of the cargo, swinging the cargo on board and then releasing it. In such a scenario, it is seen that the stress cycle experienced by the crane has one time when the crane is loaded with the cargo load and then it needs to be released of the load. Hence it is seen that the stress curve is a half cycle, which only consist of the downward loads and no force acts against the gravitational force. It is also seen that the boom element has its self-weight and it needs to be always acting downwards. This self-weight and the wind force acting on the crane results in the increase of the mean stress on the structural components of the crane. This increase in the mean stress further reduces the fatigue life of the structural components. In order to calculate the mean stress acting on the structural components we need to use special rules and the formulae which are given in the coming section.

13. RAINFLOW COUNTING METHOD FOR THE FATIGUE

The rain flow counting method earlier called Pagoda roof method. It was introduced by Mitsubishi and Endo (1968). This forms the method for cycle counting for the fatigue analysis. But in my case as the stress cycle history for the crane is obtained from the load cell so I had not used the method. But this method will prove useful for further analysis for the next researcher for the sea going condition fatigue analysis of the structure hence I will explain it in brief. Ref: [29]
This gives the time series analysis of the peak and the valleys as shown in the figure below. This is a recognized method for counting the half cycles for the fatigue calculation for the member subjected to cyclic loading. The method is explained in brief as to how to go about the approach:
The stress curve is assumed to be a combination of hills and valleys as peaks and troughs.
We start from the valleys and allow the raindrop to fall from the roof. And the pattern of the calculation represents the flow of the falling water.

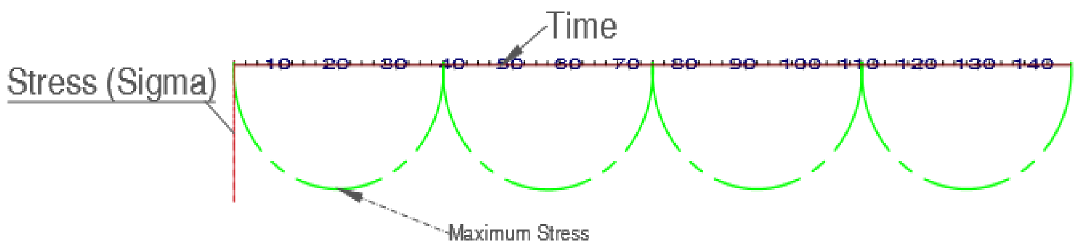


Figure 39: Stress curve for crane loading on crane structure

There are certain rules we need to follow in order to get the correct results:

1. The lines connecting the peaks and the valleys form a series of pagodas rules.
2. Each rain flow begins at the start of the time series and end at the end of every peak or valley.
3. Rainflow initiated at the peak drops down until it reaches a peak more positive value (or a valley more negative value) compared to the peak where it started.
4. Rainflow stops when it reaches the rain flow from above.
5. Rainflow terminates at the end of the time series.
6. Horizontal length of the each rain flow is counted as a half cycle of that stress range.
7. Where half cycle represents one valley and one peak. An example of the above case is shown using a simplified figure which will very well explain the method.

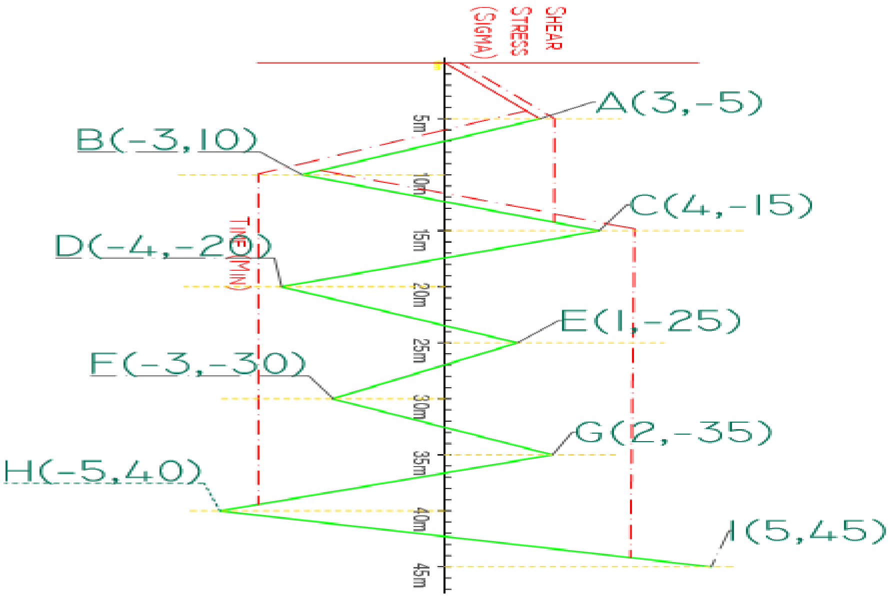


Figure 40: Example of cyclic stress curve to be analysed using Rainflow counting mechanism

This figure represents the time vs shear stress (sigma). The shear stress is in the form of peaks and the valleys. The main motive of the rainflow counting mechanism is to collect all the stress cycles having the same values. This is done using the rules discussed before and the method is so special it has been accepted by the ASTM society. And it forms an acceptable method to count the number of cycles of the stress for the fatigue calculations.

In order to explain the method more precisely I would go into is details. The stress vs time cycle seen above is inverted by 90 degrees so that the curve looks as shown in the figure given above. Then it is assumed that the rainflows from the starting edge of the stress curve and it drops down when it reaches the sharp edge. This one length of the stress represents a half cycle. These half cycles are then collected and summed in a tabular format which contains three variables. 1st column forms the path which the rain has followed. 2nd column is the cycle count which shows how much distance has been covered and the third column is the stress value.

Table 14: Counting cycle and stress range

Path	Cycles	σ
A-B	0.5	3
B-C	0.5	6
C-D	0.5	8
D-E	0.5	5
E-F	0.5	4
F-G	0.5	5
G-H	0.5	5
H-I	0.5	8

Now moving further, we see that the stress has to be converted to stress range, which is the required parameter in the y axis of the S-N curve. The stresses which we had obtained in the previous case has to be summed together to see how many cycles of the particular stress value do we have for a given time history. This is done as shown in the figure below. It is seen that all half cycles combine to form complete cycles. This is done using the special Matlab code which sums up the cycles and tells us the number of cycles. This is one of the critical steps of the fatigue counting method and is very important to count the cycles correctly. In our case the stress history is limited to the loading and offloading conditions of the crane and hence the number is finite and can be easily dealt with. But when we are considering the wave loads over 1 or 1.8 years, then in such cases the stress history contains many stress cycles and it is really complex to count the cycles manually. In this case we need a MATLAB program which filters the signal and converts them into the number of cycles and the stress range which will be then processed for the fatigue calculations.

Table 15: Number of of cycles with same mean stress

Σ RANGE	CYCLES	PATH
8	1	$C-D, H-I$
7	0	0
6	0.5	$B-C$
5	1.5	$D-E, F-G, G-H$
4	0.5	$E-F$
3	0.5	$A-B$
2	0	0
1	0	0

Based fatigue damage estimates, we have made a comparison of the different methods of the counting methods and the damage calculated by them.

$$D_{RC} \leq D_{RFC} \leq D_{LCC} \leq D_{PC} \tag{11}$$

Where:

- D_{RC}=Range Counting.
- D_{RFC}=Rainflow Counting (Proposed by ASTM)
- D_{LCC}=Level Crossing Counting
- D_{PC}=Peak counting
- NB=Narrow band approximation.

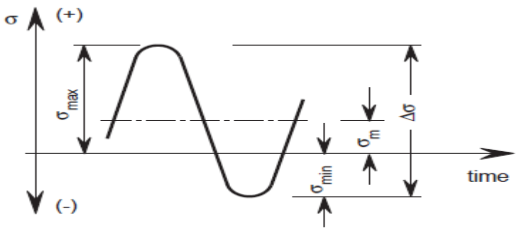
As the Rainflow counting method is more reliable and is accepted by most of the society as the method for the cycle counting for the stress cycles so we go with this method leaving the others. But in my case as the cycle of the crane are less and can be counted by the simple excel sheets for each load case so although the code was formed has not been put to use for the cycle counting of the crane load cases. But for the calculation of the cycles of the stresses created due to the wave effects on the crane when it is stored in the rested position this cycle counting method can be used and forms a vital tool in the analysis for next stage of the thesis.

Table 16: Number of cycle counted using load cell

Load case	max. SWL [t]	max. outreach [m]	Load cycles 20 years estimated	Load cycles 20 years real	Load cycles 1,8 years estimated	Load cycles 1,8 years real
1	1000	16	2080	11	182	1
2	800	25	3120	286	273	25
3	500	38	2080	34	182	3
4	500	25	3120	23	273	2
5	500	16	1040	0	91	0
6	350	38	3120	229	273	20
7	350	25	3120	11	273	1
8	250	38	3120	2560	273	224
		Total:	20800	3154	1820	276

14. EFFECT OF MEAN STRESS CORRECTION ON THE FATIGUE LIFE

The fatigue life of a component is influenced by two parameters that are the mean stress and the number of the cycle the member is subjected to (alternating stress). The SN curves mainly deal with the alternating stress (Stress Range) while it neglects the effect of the mean stress effect on the structure. So it is seen that the alternating stresses are seen to have these effects. The tensile loading of the structure tends to widen the crack and the compressive stresses tend to close it. Hence among the two cases, alternating stress value needs to be determined as it forms a vital part, in order to calculate the correct fatigue life of the component the mean stress also plays a vital role and need to be incorporated in the calculations. The two types of the stresses can be explained by the diagram given below: Ref: [08]



- $\Delta\sigma$ = Applied stress range $(\sigma_{max} - \sigma_{min})$ (N/mm^2)
- σ_{max} = maximum upper stress of a stress cycle (N/mm^2)
- σ_{min} = maximum lower stress of stress cycle (N/mm^2)
- $\Delta\sigma_{max}$ = Applied peak stress range within a stress range spectrum (N/mm^2)

$$\sigma_m = \text{Mean stress} \left(\left(\frac{\sigma_{max}}{2} + \frac{\sigma_{min}}{2} \right) \right) (\text{N/mm}^2) \tag{12}$$

- $\Delta\sigma_p$ = Permissible stress range (N/mm^2)

It is seen that the factors such as the rise in temperature, wind loads, self-weight and the loads such as the hook load always creates a rise in the mean stress on the crane structure. This rise in mean stress results in lowering of the fatigue life of the components.

The mean stress corrections are not easy to find and hence were not considered for the SN curve method for the fatigue analysis. Hence, I have used some of the approaches proposed by the researchers and made a comparative study regarding it in order to select the best method suitable for the crane loaded structure.

In cycles which are completely non-reversing (as in my case) then the mean stress calculated value is not zero. In such a case we need a correction mechanism to calculate these mean stresses as they are having a significant effect on the characteristics of the fatigue life of the component. The machine components which usually experience cyclic loading due to high revolutions of the fluctuating loads usually experience a fatigue is meant the stress ratio characterised by R=0 to R=-0.5, while in the components experiencing the complete reversal of the stresses experience the mean stress ratio of R=1 to R=-1.Where R defined as the stress ratio between maximum and minimum stress occurring during the cycle. In these calculations, the yield strength of the material at room temperature is normally used, as the temperature has effects on the mean stress value .

Now we will look into the various stress correction methods which are selected and can be useful for calculation of the accurate mean stress for the crane model. The selection of these models are widely on the type of the loading and the boom angle condition of the crane and also partly on the cycle of the loading chosen for the analysis. This comparative study is performed in order to calculate the best-suited method which does not overestimate our results and nether underestimates the number of cycles for the lifetime of the member. The rise in mean stress results in lowering of the fatigue life of the components.[Ref: Metal Fatigue In Engineering, John Wiley, 2001 and Goodman J (1930) Mechanics applied to Engineering, Volume 1]. Ref: [30]

The description of the various methods for the mean stress correction are presented below:

1. Gerber mean stress correction method.
2. Goodman mean stress correction method.
3. Soderberg mean stress correction method.
4. Marrow mean stress correction method.
5. Walker mean stress correction method.
6. SWT mean stress correction method.

a. Gerber mean stress correction method:

This is the most widely used stress correction methods. This method is represented by the formulae given below.[Ref: Fatigue and Fracture of Engineering Structures, A.Ince and G.Glinka, Backwell Publishing Ltd] [31]

$$\frac{\sigma_a}{\sigma_{ar}}+\left(\frac{\sigma_m}{\sigma_u}\right)^2=1 \quad (13)$$

Where:

- σ_a=Stress amplitude.
- σ_{ar}=Value of stress equivalent to fully reversed stress.
- σ_m=Mean stress.
- σ_u=Material ultimate strength.

Gerber formula represents an equation of a parabola with respect to the model boundary line. In order to calibrate the model a factor called as R_m is used. Where R_m is the regression coefficient found by experiment and is explained in the lower tables below it transforms the formulae to the given equation:

$$\frac{\sigma_a}{\sigma_{ar}}=1-\left(\frac{\sigma_m}{R_m}\right)^2 \quad (14)$$

Using this number of cycle of fatigue can be calculated this can be done by the given formulae:

$$N_{cal}=N_k*\left(\frac{\sigma_{ar}}{\sigma_{af}}\right)^k \quad (15)$$

Where:

- N_{cal}=Number of cycles to failure as calculated after correction.
- N_k=Number of cycles to failure.
- σ_{af}=Stress amplitude.
- σ_{ar}=Value of stress equivalent to fully reversed stress.
- k=Slope of the fatigue curve

But following this formula has certain drawbacks:

The Gerber formula does not distinguish between the compression and the tension stresses.

It is usually seen that the compression stresses increase the fatigue life while the tensile forces reduces it, but in this formula this characteristic is not accounted.

In order to solve this problem a new model was designed which is called the

b. Goodman's model

The Goodman model has rectified the flaws that were found in the Gerber model and is more advanced and shows better results than the Gerber model. [Ref:15]

$$\frac{\sigma_a}{\sigma_{ar}}+\left(\frac{\sigma_m}{\sigma_u}\right)=1 \tag{16}$$

Where the parameters used are the same as the Gerber model except the correction of the square. The parameters given in the above formulae are:

- σ_a=Stress amplitude.
- σ_{ar}=Value of stress equivalent to fully reversed stress.
- σ_m=Mean stress.
- σ_u=Material ultimate strength.

The advantages of this model are that model makes use of the material constants determined on the basis of the monotonic tensile test. So this model is easy to transform into the different material model as the constant value is easy to determine.

The model bears some of the drawbacks too, which makes the model to go for certain corrections to rectify the results. The few drawbacks of the model are:

1. As the model uses the tensile test results for the determination of the high cycle fatigue so it determines the static properties of the material. Hence, using this model is not able to determine the fatigue behaviour of the material in the right way as it is more accurately determined by the dynamic characteristics of the material properties rather than the static ones.
2. The factors such as the cycle hardening or softening of the material are not considered in the calculations.
3. This signifies that the material means stress is calculated without incorporating any of the factors which determine the mean stress sensitivity of the material.

In order to meet with the given drawbacks of the two models we move on to more refined and precise methods. This is

c. Smith, Watson and the Topper method

This is a very popular method which was designed in USA in 1970.[REF:14]This method is basically used in the strain based fatigue analysis by combining the strain equation proposed by Manson multiplied by Basque equation. The equation is defined by the following formulae:

$$\sigma_{ar} = \sqrt{\sigma_{max}\sigma_a} \tag{17}$$

$$\sigma_{ar} = \sigma_{max}\sqrt{\frac{1-R}{2}} \tag{18}$$

Where:

- σ_{max}=Maximum stress of the cycle ($\sigma_m + \sigma_a$)
- R =Stress Ratio
- σ_a = Stress amplitude

$$\sigma_{ar} = \sqrt{\sigma_{max}\epsilon_a} \tag{19}$$

This method proves to be conservative in the low cycle fatigue region. It is seen that the SWT mean stress correction gives a good result for both steel and the aluminium materials. It is seen that Marrow and SWT methods are most popular but the SWT is more accurate when compared to Marrow.

The SWT even after providing many positive results possesses some of the drawbacks which I would like to point upon.

1. In the SWT method parameter provides a rather non conservative result when it is used for the calculation of the compressive mean stresses. Therefore for the boom section the major stresses being compressive due to the suspended loads this method for stress correction is not used.
2. The SWT method does not fall good when dealing with the von misses stress data.[REF: International Journal of Engineering and Tech IJENS Vol 14] Therefore as I have used the results for the stresses using the Von Misses stresses from the finite element software hence I cannot use SWT method in order to avoid uncertainty of the results.

d. Comparison of the mean stress correction

In this part of the study a comparative study of the mean stress is made using the different models discussed and the S-N curve data.[REF:13]. The problem is much more clearly explained in the coming section which shows a clear distinction of the fatigue cycles calculated:

1. Gerber stress correction.
2. Goodman mean stress correction.
3. Smith, Watson and Tupper stress correction.
4. S-N curve data provided using the experimentation of the material.

It is seen that the different models following different approaches differ in the calculated fatigue life. But it is seen that the different models are suitable for different loading conditions and the different class of the material. In order to find the best suitable for our application we need to carry out a MATLAB experimental analysis. For the steel material used by us in the crane boom and the housing it is seen that the Marrow model is too conservative while the SWT method cannot be used because we need a summary of the Von Misses stress for the final analysis. Therefore considering these situation we select the Gerber rule.

From the literature it can be seen that the certain methods are suitable for certain analysis an example of the few of the examples are:

- SWT method of mean stress correction: Basically used for the aluminium and steel and seen to give a fairly good results. Drawback of this method is that it fails to work with the von misses stress data. But is fairly good with the other type of stress analysis.

- Millers rule for the mean stress correction: This gives a coarse result of the stress life. It is an old method and certain methods to correct the drawback have been found out regarding this method.
- Walker method of mean stress correction: This method simplifies the proposed method of SWT. It has introduced a coefficient (ϒ) which changes according to the material properties of the material and hence makes the formulae more material specific.

The significance of the rise in temperature on the mean stress effect. It is seen that few of the components such as the sheaves and the wire ropes on the crane will experience a rise of temperature during operation and the hoisting and lowering operations. This rise in temperature results in the reduction of the fatigue life of the component. Such a phenomena can be calculated using the research of the regression coefficients which has been provided by the researchers [REF: 9] 'J. Warren and D. Y. Wei, "A microscopic stored energy approach to generalize fatigue life stress ratios," *International Journal of Fatigue*, vol. 32, no. 11'. The result of the analysis for the temperature rise of 260 degrees has been calculated for the stress ratio of 0.05 at 260 degree for the various mean stress correction methods specified above:

The Regression coefficient values for various stress ratios have been obtained from the research paper as per [REF: 9]. These values are for the material type S-355 steel with properties as of the crane.

Stress ratio, R	T (°C)	A	q	R-sq^a
−1.00	260	1.2052	0.08906	0.83392
−0.30	260	1.6185	0.12903	0.88150
0.05	260	1.9562	0.15831	0.83834
−1.00	427	2.0829	0.12652	0.92928
−0.50	427	1.7352	0.12031	0.96745
−0.30	427	2.5149	0.16188	0.93461
0.05	427	1.8988	0.14937	0.61695
−1.00	538	2.4364	0.13748	0.94248
−0.50	538	1.5933	0.11022	0.68894
−0.25	538	2.3464	0.15238	0.95618
0.10	538	2.1980	0.15840	0.68902
−1.00	649	0.9486	0.05910	0.98988
−0.50	649	0.9853	0.06634	0.85683
−0.25	649	1.7054	0.11933	0.85279
−0.05	649	1.2785	0.10368	0.83803
−1.00	760	0.8554	0.04583	0.82717
−0.50	760	0.7373	0.03641	0.63333
−0.30	760	0.7252	0.03982	0.45920

Model parameters.

T (°C)	T/T_m	μ^* (GPa)	σ_o (MPa)	σ_{uts} (MPa)	Q_o (kJ/mole)
260	0.31	1460	81.7	1400	285
427	0.41	1389	41.2	1240	270
538	0.47	1341	56.9	1240	261
649	0.56	1269	64.4	1200	460
760	0.60	1197	–	940	–

Figure 41: Reference from experimental paper Ref: [33]

The calculated results obtained for various methods of mean stress correction method for the crane fatigue cycles for the structural component having temperature of 260 degrees and the similar input material properties and the stress reversals experienced are as given below:

```
Number of cycle to failure according to experimental S-N curve:
Exp. :1935078
Number of cycle to failure according:
Goodman: 631046
Gerber: 1679367
SWT: 364523
```

The rise in the mean stress can also be calculated due to the wind loads, self-weights or the temperature variations on the structure as shown in the given example. This example also shows that the rise in temperature showed no change in the life cycles for the material for the experimental data as no mean stress effect was incorporated while significant reduction in the total stress cycles is seen for the other mean stress corrected results. These results depict the real scenario and the actual conditions and depending on the area of use they can be made to use by incorporating then in the formulae.

15. FATIGUE ANALYSIS OF THE CRANE STRUCTURE

The fatigue strength of the structural components of the crane is basically determined by [REF: 13]:

- The material of the structural components of the crane.
- The shape, surface conditions, state of corrosion, size (scale factor) and other factors constituting stress concentration.
- The ratio R between the minimum and the maximum stress which occur between the various cyclic loads add is used in order to determine the mean stress.
- The stress spectrum considered.
- The number of the stress cycles the component is subjected to.

In order to begin with the fatigue calculation, the static stresses obtained from the finite element model need to be converted into cyclic stresses. In such a case the R factor comes into consideration. For the alternating stresses, R is considered to be {-1} for a fully reversing cycle. The stress obtained from the finite element model analysis of the structure. The curve representing the stress range for the crane loading discharging condition is shown in *figure 42* given below. It is seen that the stress is maximum when the crane load is on the boom tip then as the crane starts to discharge the loads the curve comes towards the maximum stress value and finally ends at 0 value. The X axis shows the time and boom span taken for one loading discharging condition on the crane. And the count of such stress time curve forms the number of stress cycles the crane is subjected for 1 or 1.8 years of the time span. This cycle data is obtained from the load cell values which has recorded data for 1.8 years and is then extrapolated to 20 years of the life of the crane.

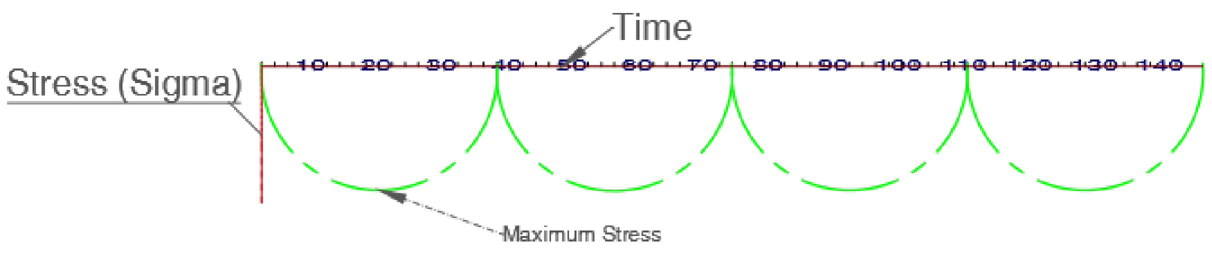


Figure 42: Cycles as experienced during crane loadings

In the rule requirements provided by' *DNVGL 2015 page 4-19 Ref: [17]* 'it has been stated if the number of the stress cycles experienced by the crane structural components are below 20000 the fatigue calculations may be omitted. *(DN 015 EN Ref: [08] rule* state if loading is below 8000 fatigue checking is not required.)

It is seen that the approach followed for the calculation of the fatigue depends on the method of obtaining the stress of the component and the data which needs to be processed. Depending on the stress tensors or the hot spot stress or the notch stress values taken for the calculation determines the method chosen. Now I will be discussing the two most popular and commonly used approaches for the calculation and would find out the best suited among them for our analysis and the calculations.

a. Damage criteria approach.

The nominal stress approach is the simplest approach which is followed. It is a far-field approach in which the stress acting due to the force, moments or a combination of the load is analysed to find out the possible site for the crack initialization. As these approaches require all the structural areas of the crane to be analysed so it is desirable to create a finite element of the structure. The locations of the hot spots are found and the curve to represent the similar details for the fatigue is plotted.

The hot spot stress is the stress acting at the toe of the weld. It is a point where the stress concentration is the highest. These possess the points of the hot spots where the crack initialization may take place. It basically comprises the membrane and the bending stress component. Such an analysis predicts the strength of all the welds of that type, but it requires a more detailed finite element stress approach to deal with it.

Although this method has a drawback that it requires a lot of data and the calculations to plot an accurate viable distribution. The possibility of the shortage of the data which is a common feature and the drawback. In my case, I have chosen this method to predict the life of the specimen and the crane components because the operating life of the crane which I am analysing on the ship has only undergone 1.8 years of its lifetime with the crane operation. And for the operator, it is essential to keep a track with the crane that it will survive the lifetime of the ship. And also to predict the lifting capacity after years of operation the operator needs to know which is the areas of concern which need to be stiffened or additional metal work needs to be carried out on it.

Few of the data points followed for this type of approach is:

- Calculation of fatigue damage is made till initialization of crack
- This method is valid for machinery having fewer operations during its lifetime.
- The rules state that in order to implement this approach there needs to be a continuous stress monitoring device placed on the crane and in our case we have load cell on boom tip for counting stress cycles and stress loadings on the structure.
- This approach is used where the consequences of the accident may be quite severe if it takes place.

So seeing all these points we judge out that this approach is best suited for our case.

b. Safe life Approach

The second method of the fatigue analysis is by to determine constant stress level and different constant stress range for the model. It rather calculates the remaining life of the component rather than the damage during the life approach and is not suitable for our analysis.

16. DAMAGE CRITERIA APPROACH

The Miner's Rule cumulative damage it states that the failure of a component occurs when the number of the operation cycles approaches the total fatigue cycles of the component. It can be represented in a simple equation stating: Ref: [10]

$$D = \sum_{i=1}^{n_b} \frac{n_i}{N_i} \tag{20}$$

Where:

- D=Damage of the concerned node.
- n_b=Total number of load cases.
- n_i=Counted cycle using the rain flow analysis.
- N_i=Total cycles for failure after the correction factors.

This method is found to be useful under the following conditions:

- Calculation of fatigue till initialization of crack
- Valid for machines having fewer operations during its lifetime
- Need to have continuous stress monitoring device.(Load cell on boom tip)
- For machines whose, consequences during failure may be catastrophic.

This approach can be further subdivided to find out the damage:

1. Plate sections.
2. Weld sections

a. Plate analysis:

In order to determine the fatigue failure on the plate sections the following conditions need to be kept in mind:

- Carried out to check failure of plate sections of the structure
- Notch case of 120 to 160 used for the analysis.

I would like to explain the detailed approach used for the plate analysis for fatigue:

In order to calculate the fatigue for the plate of the structure. The structural component comprising of the boom and the housing are divided into different plate sections depending on the thicknesses of the plates used for its constructions. Each of these plates is further divided into coarse grids in the finite element software. These grids are 0.5 meters to 1 meter square in size. Then using the finite element software the stresses are found for each of these grids. Now we have generated a database for the stresses at various grids and the coordinate in x,y and z defining the location of the grid centres in the finite element model. This is basically done so that the grid points having maximum stress can be located in the real model.

Now we move on to the next stage of the analysis. Here we are basically concerned with the calculation of the fatigue. In order to calculate the fatigue we need three parameters:

- The stress at the point of calculation.
- The notch case of that section.
- The number of cycles the element section has already gone through during its present life.

The first two cases are basically required to calculate the maximum number of the cycle the material can withstand before facing fatigue crack. While the third case has been already calculated by us using the stress counting method.

The stress at the grid point was obtained from the finite element software as discussed in the first paragraph.

Regarding the notch case, a special approach has been followed. In this approach, I have located the junctions where the plate sections change and it is obvious that these changed plate sections will be having weld connections between them. So all these changing plate sections are located and a notch case lower than the plate section that is 80 to 112 (compared to 120 for the plate) is used. Now in the first step, a notch case of 80 is used. This notch case is then checked for the accordance with three basic equations which need to be fulfilled for the calculation. These equations are:

$$\frac{\Upsilon_{Ff} \cdot \Delta\sigma_{E,2}}{\Delta\sigma_C / \Upsilon_{Mf}} \leq 1.0 \tag{21}$$

$$\frac{\Upsilon_{Ff} \cdot \Delta\tau_{E,2}}{\Delta\tau_C / \Upsilon_{Mf}} \leq 1.0 \tag{22}$$

$$\left(\frac{\Upsilon_{Ff} \cdot \Delta\sigma_{E,2}}{\Delta\sigma_C / \Upsilon_{Mf}}\right)^3 + \left(\frac{\Upsilon_{Ff} \cdot \Delta\tau_{E,2}}{\Delta\tau_C / \Upsilon_{Mf}}\right)^3 \leq 1.0 \tag{23}$$

Where:

- Υ_{Ff} x $\Delta\sigma_{E,2}$=λxΔσ
- λ=damage ratio
- $\Delta\sigma_{E,2}$=Endurance stress range
- $\Delta\sigma_C$ =Stress Range

- Υ_{Mf} =Consequences of failure
- $\Delta\tau_C$=Torque range
- $\Delta\sigma_C$ =The fatigue strength for 2x10^6 cycles are represented on the curve given below
- $\Delta\tau_R$=The fatigue strength of the shear stress is given by the curve below.

If the validity of any of this equation is not fulfilled then a notch case of 112 instead of 90 is tried. And even if then the equation is not meeting the requirements of the right side coefficients to be less than one then the final and the best weld case of 120 is used. This is done using the condition statements of the MATLAB code.

Once the notch case has been decided then using the equation given below a maximum number of cycles that particular section can withstand is calculated. The formulae used for the above calculation is as given below:

$$N_R = 2x10^6 \frac{(\Delta\sigma_C/\Upsilon_{Mf})\hat{}m}{\Delta\sigma_{ch}^m} \tag{24}$$

Example: $$2x10^6 \frac{(90/1.15)\hat{}3}{192\hat{}3} = 103515 \text{ cycles} \tag{25}$$

Where:

- N_R=Maximum cycle for damage of component (Using SN curve for S355 material)
- $\Delta\sigma_{ch}$=Endurance stress range from finite element grid stress for that location.
- $\Delta\sigma_C$ =Stress Range using the specific notch case selected for that location.
- Υ_{Mf} =Consequences of failure (1.15 in our case)

Now using these calculated values I can calculate the damage at that particular location of the section. The formulae used for the calculation will be the Millers rule:

$$D = \sum_{i=1}^{n_b} \frac{n_i}{N_i} \tag{26}$$

Where:

- D=Damage of the concerned node.
- n_b=Total number of load cases.
- n_i=Counted cycles.
- N_i=Total cycles for failure after the correction factors.

This calculation is done for all the load cases and the cumulative damage is calculated which will give the final damage at that particular location of the structure.

Using this approach an output file containing the history of all plate grid point locations with their index and the coordinates are created and the respective cumulative damage which they will experience in 25 years of the lifetime of the component. This data is seen and we now know the notch case at all these grid points, their stress history and the cumulative damage. This forms the final result and can be used for the prediction of the fatigue life of the various plate sections forming the crane housing and the boom structure.

The output file for the nodal damage results looks like this:

Grid No.	X Coordinate	Y Coordinate	Z Coordinate	Damage
19	14.8547	-2.3735	-37.1244	0.1061
20	15.3547	-2.3735	-37.1244	0.0837
23	14.3547	-2.3735	-37.6244	0.2021
24	14.8547	-2.3735	-37.6244	0.1995
9	7.7474	1.8561	-19.5786	0.1069
10	8.2474	1.8523	-19.5786	0.0612
11	8.7474	1.8485	-19.5786	0.1909
16	7.7473	1.8458	-20.0785	0.1214
17	8.2473	1.842	-20.0785	0.0472
18	8.7473	1.8382	-20.0785	0.1859
23	7.7473	1.8356	-20.5784	0.1297
24	8.2472	1.8318	-20.5784	0.0467
25	8.7472	1.828	-20.5784	0.0351
26	9.2472	1.8242	-20.5784	0.1017
31	8.2472	1.8215	-21.0783	0.0505
32	8.7472	1.8177	-21.0783	0.0644
33	9.2471	1.8139	-21.0783	0.0373

b. Weld analysis:

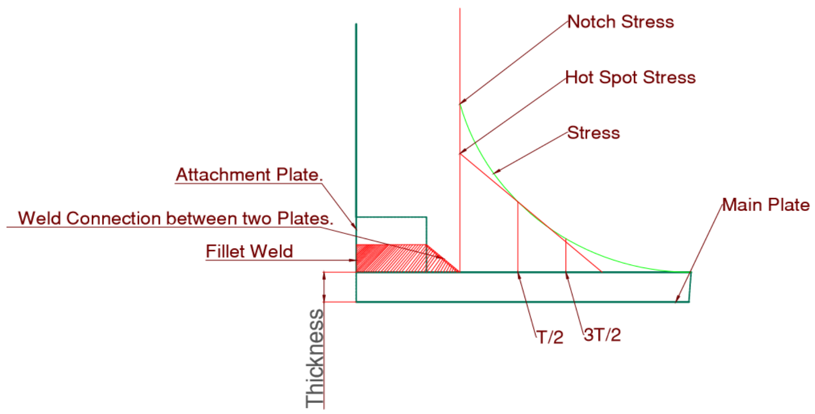


Figure 43: Weld stress

This approach is used for the analysis of the weld for the failure by fatigue. In this case the following steps are to be followed.

- Carried out to check failure of welds.
- SN curve type of 80 to 120 used for the analysis of the fatigue in this case.

- Analysed by special RFAT module defining the boom section as a linear isotropic element for the analysis.
- This is a more elaborate approach where the boom section is assumed to be composed of a combination of 40 member elements of different cross sections. These cross sections are then defined for the notch cases at 16 locations. The areas of the welds are given the notch case lower than that of the normal plating. Using this approach the entire section of the boom is analysed and finally, a graphical and history of the damage at each defined section is obtained. This is then plotted in form of a graphical solution which can be read easily.

Using the damage characteristic as given in the European *EN 1993-1-9* rules calculate the fatigue damage of the structure of crane.
The fatigue damage is basically based on the structural characteristics such as the cross section, weld properties, material properties, and the structure supporting and the weight. While the second parameter governing the fatigue behaviour is the stress applied to the structure.

The internal forces are analysed by the structural analyses of the member and its cross sections. The welds are to be considered as the points which are most susceptible to the fatigue failure. These points may be welds, corners, sharp edges and positions where there is already a crack. The fatigue design basically compares the nominal stress range $\Delta\sigma$ and $\Delta\tau$ with the design values of $\Delta\sigma_R$ and $\Delta\tau_R$ stress ranges. The method for the calculation is explained in more details in the coming section.

c. Design equations and the calculation of the parameters:

The damage equivalent stress ranges $\Delta\sigma_R$ and $\Delta\tau_R$ relates to 2x10^6 stress cycles and these are to be checked with the applied stress and the torque ranges According to the stress ranges for the nominal and the shear stresses have limitations which are given by the given equations:

$$\Delta\sigma \leq 1.5 f_y \tag{27}$$

$$\Delta\tau \leq \frac{1.5\, f_y}{\sqrt{3}} \tag{28}$$

Where:

- $\Delta\sigma$=Stress range (As was shown to calculate using rain flow method)
- f_y=Yield stress of the material (355 N/mm^2).
- $\Delta\tau$=Torque range.

These values are checked and analysed for each node.

d. The design equation for the fatigue:

The stresses on the action side are determined by the serviceability level. For the partial safety factor Υ_{Mf} =1. 0 applies. These assessment methods were analysed in the previous sections to account for the various parts of the crane and its influence on the damage to that part on the working of the crane and its components.

Table 17: Safety factors used for different approaches for fatigue calculation

Assessment method	**Consequence of failure**	
	Low	**High**
Damage tolerant	1.00	1.15
Safe life	1.15	1.35

The design for the fatigue follows the following equations. This covers the basic equation which is again tested for the validity of the stress range calculated for each nodal point.

$$\frac{\Upsilon_{Ff}.\Delta\sigma_{E,2}}{\Delta\sigma_C/\Upsilon_{Mf}} \leq 1.0 \tag{29}$$

$$\frac{\Upsilon_{Ff}.\Delta\tau_{E,2}}{\Delta\tau_C/\Upsilon_{Mf}} \leq 1.0 \tag{30}$$

Where:

- Υ_{Ff} x $\Delta\sigma_{E,2}$=λxΔσ
- λ=damage ratio
- $\Delta\sigma_{E,2}$=Endurance stress range
- $\Delta\sigma_C$ =Stress Range
- Υ_{Mf} =Consequences of failure
- $\Delta\tau_C$=Torque range

Simultaneous effect of the normal and the shear force effects are to be compared

$$\left(\frac{\Upsilon_{Ff}.\Delta\sigma_{E,2}}{\Delta\sigma_C/\Upsilon_{Mf}}\right)^3 + \left(\frac{\Upsilon_{Ff}.\Delta\tau_{E,2}}{\Delta\tau_C/\Upsilon_{Mf}}\right)^3 \leq 1.0 \tag{31}$$

- $\Delta\sigma_C$ =The fatigue strength for 2x10^6 cycles are represented on the curve given below
- $\Delta\tau_R$=The fatigue strength of the shear stress is given by the curve below.

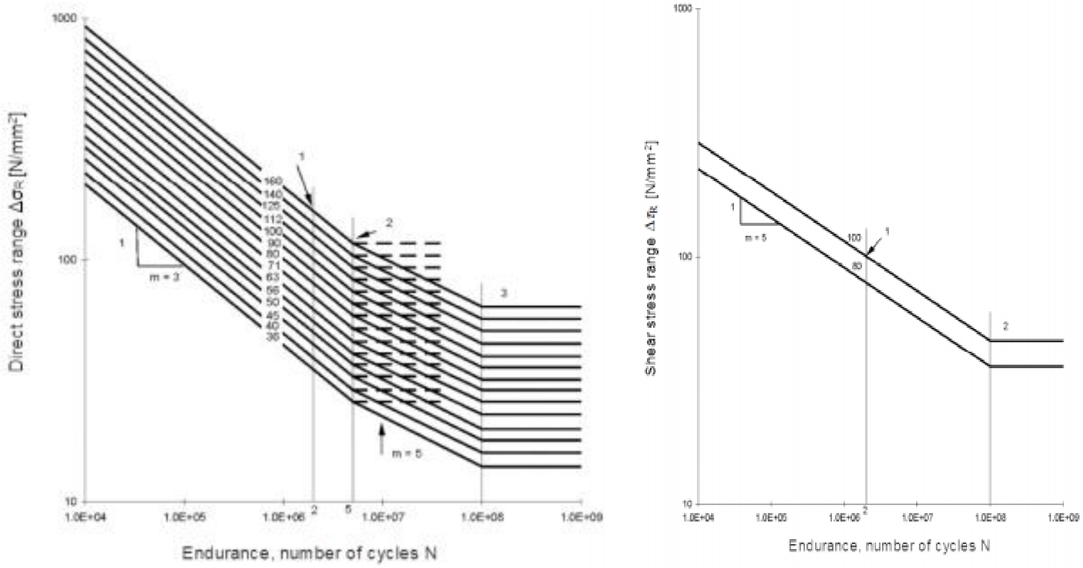


Figure 44: Stress Strain curve .[REF:Eurocode EC 3-1-9]

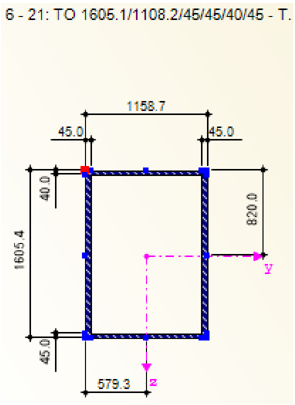
$$N_R = 2x10^6 \frac{(\Delta\sigma_C/\Upsilon_{Mf})^\wedge m}{\Delta\sigma_{ch}^m} \qquad (32)$$

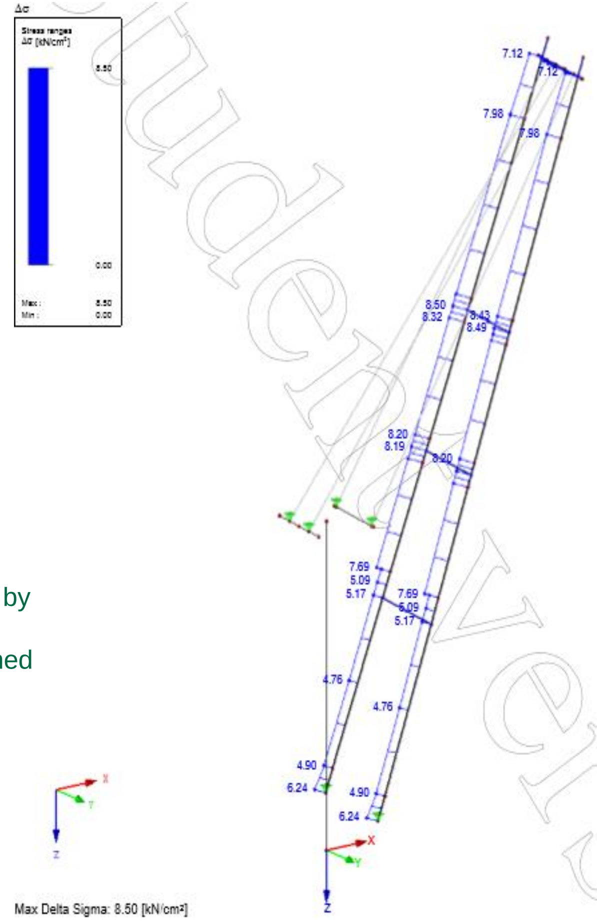
This is carried out for each nodal stress obtained from the finite element analysis. It gives us the number of cycles the component can sustain before failure. Ref: [10]

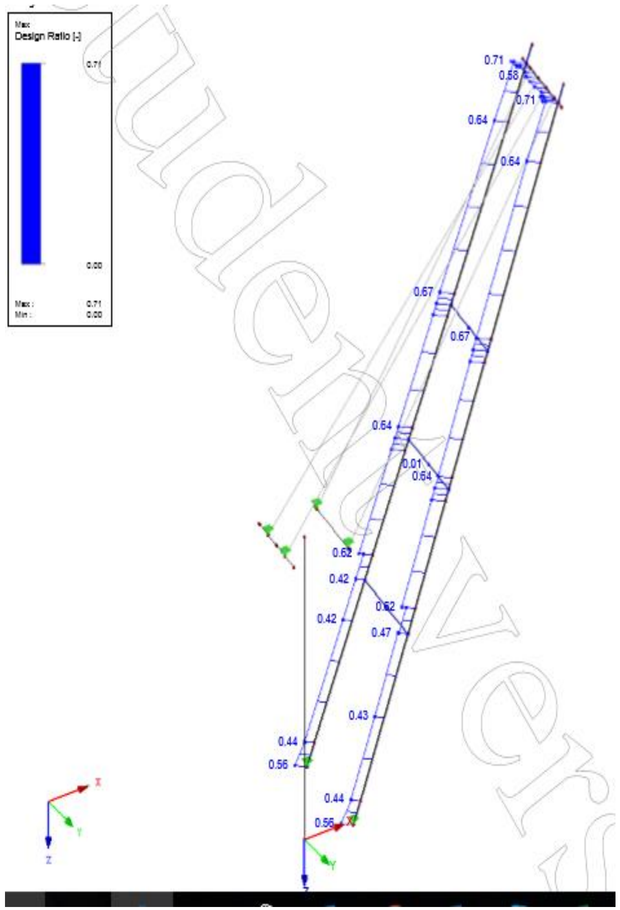
1. Then we obtain the damage equivalent factor λ from the general formulae:

$$\lambda = \frac{1}{2x10^6} x \sum_{1}^{8} (\frac{\Delta\sigma_{ch}}{\max \Delta\sigma}^{m} x \, n_E)^{\frac{1}{m}} \qquad (33)$$

Weld Fatigue determination approach







- Boom divided into 40 member elements.
- Each member notches defined at 16 locations.
- Mean stress calculated by finite element method.
- All 8 load cases combined to calculate damage

33

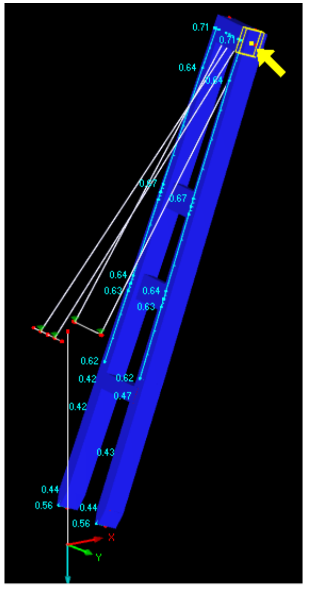
Locating Maximum Fatigue Damage

Maximum damage given by manufacturer

Outreach [m]	SWL [t]	Damage ratio for 1 load cycle D_1	Load cycles n	Damage ratio D
≤ 16	≤ 500	0,000005968	1040	0,006206235
	≤ 1000	0,000031545	2080	0,065612683
≤ 25	≤ 350	0,000009908	3120	0,030913944
	≤ 500	0,000020310	3120	0,063366019
	≤ 800	0,000061810	3120	0,192845744
≤ 38	≤ 250	0,000020435	3120	0,063757248
	≤ 350	0,000031758	3120	0,099084534
	≤ 500	0,000067995	2080	0,141428567
		Total	20800	0,663214975

Maximum damage calculated on welds :

=~ 0.71
Out of 1.00



Damage difference=0.71-0.663=0.047

35

17. CONCLUSION

From the above thesis, we can derive important findings . Among them some of the major findings are:

It is seen from the experimental calculations that the load case with maximum boom outreach (38mts) is the limiting load case for crane operation. This is said because it is seen from the experimental data that even though the boom is loaded with 500 tonnes of load even then:

- Housing deflections are maximum (46.9mm)
- Horizontal forces on bearing are maximum (as observed from the finite element resultant force)

The maximum fatigue damage is found to occur on the boom tip after 25 years of lifetime. This can be observed by viewing the weld history results of the boom section generated by the RFAT finite element software.

Structure welds are more prone to fatigue failure compared to the plating

The housing bottom plating and the foundation shape is critical for analysis and hot spot point of view. This can be seen that in all the load cases the bottom plating of the crane housing has stress concentration points. This can be reduced by changing the form of the housing bottom section. By creating a rounded section instead of the sharp edges as is seen in the crane housing.

The window areas on housing need to be minimised to give more structural strength. It is seen that the front and the back housing walls are the major load bearing parts and the cut outs in it for the windows tend to create hot spots at the corners and the edges of the structure.

The last and the final observation is that when we compare the classification rules for the lifting appliances among the Lloyd's Register and the DNVGL it is seen that even though they have different design factor values. But in the end, it sums out to be almost the same for both the rules.

Some of the places marked as the hot spots and the platings more prone to the fatigue failure are the ones which need to be checked thoroughly for the crack initialization and the weaknesses. These places need to be marked and more frequent inspection and intense surveys to be carried out. It is seen that the cranes present on the heavy lift vessels are very less prone to the fatigue failure at the plate sections while the fatigue life at the welds are more. This is basically as the welds have a lower notch case. So the welds are supposed to fail before then the plate section.

18. REFERENCES

/1/ NMF Crane Manuel, Operations and Maintainance

/2/ Standards For Offshore and Platform Lifting Appliances DNVGL 0378, Section 10, Page 100, Edition May 2016

/3/ DNV-GL RU 0050 Cranes on Crane Vessels

/4/ Code for Lifting Appliances in a Marine Environment, Lloyds Register (Jan 2016), Chapter 4 page 80, "Cranes and submersible Lifting Appliances".

/5/ Code for Lifting Appliances in a Marine Environment, Lloyds Register (Jan 2016), Chapter 4, Section 2.3.4

/6/ "Federation European De La Manutention", Booklet 4, 'Checking For Fatigue and Choice of Mechanism Components'.

/7/ Rules for the Classification of Inland Navigation Vessels PART B – Hull Design and Construction

/8/ Eurocode 3.DIN 15 018-1 Krane, "Rules for the design of hoisting Appliances".

/9/ Eurocode 3 DIN 15 001-1 Krane

/10/ Eurocode 3.1.9 DIN 4132, Fatigue Assessment.

/11/ Technical Standards Committee Guidelines for Maritime Lifting and Lowering 0027/ND

/12/ Journal of Construction Steel Research

/13/ Reliability Based Design Guidelines of Fatigue Of Ship Structure

/14/ A microscopic stored energy approach to generalize fatigue life stress ratios," International Journal of Fatigue, vol. 32, no. 11.

/15/ Standards For Shipboard Lifting Appliances DNVGL ST 0377

/16/ Guidelines for Marine Lifting & Lowering Operations DNVGL ST-N001

/17/ Guidelines for Fatigue Design of Offshore Steel Structures 2010 DNV-RP 203

/18/ Standards For Offshore and Platform Lifting Appliances DNVGL-ST- 0378, Section 10, Page 84, Edition May 2016

/19/ Germanischer Lloyd, Loading Gear on Seagoing Ships and Offshore Installations, Edition July 2015]

/20/ Journal of Construction Steel Research, ESEVIER 'A comparison of fatigue between S-355 and S-690 grade steels

/21/ Fatigue and Fracture of Engineering Materials and Structure 'Modification of Marrow and Smith Watson Topper Mean Stress Correction Models'

/22/ Mean Stress Correction Effects On Fatigue Life Behaviour of Steel Alloys, By Quasim Bader, Emad Kadum.International Journal of Engineering and Technology Vol-14,No 04.

/23/ Kranbahanen Besmessung und konstruktive Gestaltung, Seesselberg

/24/ International Journal of Fatigue, vol. 32, no. 11

/25/ Manual RFEM Finite element Manual on RFAT

/26/ Standard for Certification of Lifting Appliances, Oct 2011. Ch-2, Sec.6

/27/ Guidelines for the Design of Crane Supporting Steel Structure

/28/ Manual RFEM Software.

/29/ Website: http://nptel.ac.in

/30/ Metal Fatigue In Engineering, John Wiley, 2001 and Goodman J (1930) Mechanics applied to Engineering, Volume 1

/31/ Fatigue and Fracture of Engineering Structures, A.Ince and G.Glinka, Backwell Publishing Ltd

/32/ International Journal of Engineering and Tech IJENS Vol 14

/33/ International Journal of Fatigue, vol. 32, no. 11

APPENDIX A

Boom Hot Spot Locations

	Input Data:					
	Material Properties:		S-355			
	Material Model:		Linear Elastic			
S.No	**Load Case No**	**Loading Repeats**	**Stress Type**	**Hot spot Location**	**Maximum Stress [MPa]**	**Fatigue Failure(Y/N)**
1	Loadcase 1	20800	Von Mises	2099/725/2099/1142	-186	N
2	Loadcase 2	20800	Von Mises	2141/735/2108/1154	-145	N
3	Loadcase 3	20800	Von Mises	2180/1348/721/1180	-143	N
4	Loadcase 4	31200	Von Mises	2141/699/2108	-114.9	N
5	Loadcase 5	10400	Von Mises	2099/725/659	108.2	N
6	Loadcase 6	31200	Von Mises	2180/721/254/1180	-113	N
7	Loadcase 7	31200	Von Mises	2141/699/2108	-86	N
8	Loadcase 8	31200	Von Mises	2180/721/254/1180	-92.5	N

Housing Hot Spot Locations

S.No	Load Case No	Loading Repeats	Stress Type	Hot spot Location	Maximum Stress [MPa]	Fatigue Failure(Y/N)
1	Loadcase 1	20800	Von Mises	10656 /12438 /18473	-210	N
2	Loadcase 2	20800	Von Mises	10658 /158 / 16760	-190	N
3	Loadcase 3	20800	Von Mises	190 / 10658 /12438	-163	N
4	Loadcase 4	31200	Von Mises	190 /222/ 10656	-155	N
5	Loadcase 5	10400	Von Mises	190 / 21543 /10657	108.2	N
6	Loadcase 6	31200	Von Mises	158 / 16760 / 222	-113	N
7	Loadcase 7	31200	Von Mises	10657 / 222	-86	N
8	Loadcase 8	31200	Von Mises	12438/ 190	-92.5	N

- It can be seen that the hot spot location depends on the boom angle and the magnitude depends on the loading weight.

APPENDIX B

Summary Of the Design Coefficients used for the Crane Structure			
S.No	**Regulation Followed**	**Lloyd's values** Code for lifting appliances in marine environment (edition Jan 2016)	**DNV-GL Values** Standard for shipboard lifting appliances DNVGL-ST-0377
(I)	**Shipboard Cranes**		
1	Vertical dynamic factor (Minimum)	Ref: Code for Lifting Appliances Fh.min = 1,1 (for jib crane) Fh.min = 1,15 (for jib type grab cranes)	(Ref DNVGL-ST-0377 Sec 5.4.4) Hoist load=1.15
	Basic Allowable stress/yield stress		
2	Without wind force	Load case 1=0.67 (Ref Ch 4, Sec 2.17)	0.67
3	With wind force	Load case 2=0.75 (Ref Ch 4, Sec 2.17)	0.75
4	Duty Factor	Deck jib crane=1.05 (Ref Ch 4, Sec 2.3)	None
5	Heel/Trim in degree.	5 degree heel 2 degree trim (Ref Ch 1, Sec 2.3)	5 degree heel 2 degree trim (Ref Table 4-1)
	Overall Design Factor		
6	Without wind force	0.86 (Ref Summation of all factors)	0.86
7	With Wind force	0.77 (Ref Summation of all factors)	0.77
8	Shock load	To consider	To consider
9	Fatigue analysis	Operating life 6x10^5 cycles (Ref Ch 4, Sec 2.3.2 & 2.3.4)	Load cycles 6x10^5 (Ref Table 5-7 cranetype C1)

(II)	Offshore Crane		
1	Minimum vertical dynamic factor	offshore crane=1.10 (Ref Ch 4, Sec3.3.2)	factor=1.35 (Ref Table 5.1)
	Basic Allowable Stress		
2	Without wind force	0.67 (Ref Ch 4, Sec 2.17)	0.67
3	With wind force.	0.75 (Ref Ch 4, Sec 2.17)	0.75
4	Duty factor	offshore crane =1.20 (Ref:Ch 4, Sec 3.2)	None
5	Heel/Trim degree.	5 degree heel 2 degree trim. (Ref Ch 1, Sec 2.3 & Ch 4, Sec 3.2 & 3.4)	5 degree list and 2 degree trim (Ref Table 4-1)
	Overall Design Factor		
6	With wind force	0.87	0.87
7	Without wind force	0.97	0.97
8	Shock loads	To consider	To consider
9	Fatigue analysis.	As per rules (Ref Ch 4, Sec 2.3.2 & 2.3.4)	As per the DNVGL ST-0377 rules

Note: The figures marked in red are the factors which have a different value for the two classification societies. Rest all values are the same .

APPENDIX C

Boom output forces and Reaction forces imposed on housing

	Load case1		Load Case 2		Load Case 3		Load Case 4		Load Case 5		Load Case 6		Load Case 7		Load Case 8	
Sum of loads in X	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN
Sum of support forces in X	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN	0.00	kN
Sum of loads in Y	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN
Sum of support forces in Y	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN	-77.56	kN
Sum of loads in Z	11369.80	kN	9418.21	kN	6507.52	kN	6475.04	kN	6464.58	kN	5035.99	kN	5003.76	kN	4054.97	kN
Sum of support forces in Z	11369.80	kN	9418.21	kN	6507.52	kN	6475.04	kN	6464.58	kN	5035.99	kN	5003.76	kN	4054.97	kN
Resultant of reactions about X	16194.100	kNm	11289.600	kNm	2836.090	kNm	-10.229	kNm	8109.210	kNm	2003.160	kNm	4957.320	kNm	1447.880	kNm
Resultant of reactions about Y	-5.66E+04	kNm	-8.18E+04	kNm	-8.68E+04	kNm	-5.19E+04	kNm	-2.83E+04	kNm	-6.07E+04	kNm	-3.58E+04	kNm	-4.34E+04	kNm
Resultant of reactions about Z	-11.258	kNm	-19.413	kNm	-33.998	kNm	-25.948	kNm	-11.365	kNm	-34.059	kNm	-19.531	kNm	-34.087	kNm
Maximum displacement in X-direction	344.5	mm	234.7	mm	95.0	mm	150.3	mm	185.0	mm	69.5	mm	113.4	mm	52.5	mm
Maximum displacement in Y-direction	106.9	mm	81.4	mm	33.0	mm	-5.2	mm	55.2	mm	23.6	mm	37.4	mm	17.3	mm
Maximum displacement in Z-direction	159.9	mm	198.2	mm	306.1	mm	119.9	mm	85.5	mm	225.6	mm	95.5	mm	172.0	mm
Maximum vectorial displacement	390.3	mm	312.0	mm	318.2	mm	189.3	mm	209.0	mm	234.6	mm	150.7	mm	178.9	mm
Maximum rotation about X-axis	8.5	mrad	9.0	mrad	7.9	mrad	-2.8	mrad	4.5	mrad	6.0	mrad	4.4	mrad	4.9	mrad
Maximum rotation about Y-axis	-13.9	mrad	-10.7	mrad	-10.1	mrad	-6.8	mrad	-7.8	mrad	-8.1	mrad	-5.8	mrad	-6.8	mrad
Maximum rotation about Z-axis	-13.9	mrad	-9.2	mrad	-3.3	mrad	-3.7	mrad	-7.4	mrad	-2.5	mrad	-4.6	mrad	-2.1	mrad

Reaction forces to be Imposed on Housing

Sum of loads in X	77.81	kN	-74.33	kN	-62.71	kN	75.09	kN	-76.18	kN	-76.51	kN	-75.51	kN	76.61	kN
Sum of support forces in X	77.81	kN	-74.33	kN	-62.71	kN	75.09	kN	-76.18	kN	-76.51	kN	-75.51	kN	76.61	kN
Sum of loads in Y	396.61	kN	-3337.24	kN	1901.68	kN	26.30	kN	137.73	kN	31.05	kN	20.34	kN	24.66	kN
Sum of support forces in Y	396.61	kN	-3337.24	kN	1901.68	kN	26.30	kN	137.73	kN	31.05	kN	20.34	kN	24.66	kN
Sum of loads in Z	-13402.20	kN	-7363.81	kN	-9864.00	kN	-8223.95	kN	-8271.41	kN	-6509.19	kN	-6741.56	kN	-5576.13	kN
Sum of support forces in Z	-13402.20	kN	-7363.81	kN	-9864.00	kN	-8223.95	kN	-8271.41	kN	-6509.18	kN	-6741.56	kN	-5576.13	kN
Resultant of reactions about X	-1.66E+05	kNm	-1.74E+05	kNm	-2.00E+05	kNm	-1.39E+05	kNm	-9.05E+04	kNm	-1.51E+05	kNm	-1.03E+05	kNm	-1.17E+05	kNm
Resultant of reactions about Y	2549.480	kNm	-655.136	kNm	261.328	kNm	1798.770	kNm	-838.522	kNm	158.717	kNm	-706.669	kNm	666.032	kNm
Resultant of reactions about Z	-709.275	kNm	802.790	kNm	-8145.860	kNm	-1105.900	kNm	747.530	kNm	1807.400	kNm	1112.490	kNm	-1421.810	kNm
Maximum displacement in X-direction	-3.6	mm	-3.4	mm	-5.0	mm	4.1	mm	-1.9	mm	-3.6	mm	-2.3	mm	28.8	mm
Maximum displacement in Y-direction	23.9	mm	24.1	mm	34.0	mm	21.3	mm	12.9	mm	25.3	mm	15.8	mm	19.5	mm
Maximum displacement in Z-direction	6.7	mm	6.8	mm	7.0	mm	5.2	mm	-4.5	mm	5.3	mm	-4.6	mm	-11.9	mm
Maximum vectorial displacement	24.3	mm	24.6	mm	34.6	mm	21.8	mm	13.2	mm	25.7	mm	16.1	mm	30.6	mm
Maximum rotation about X-axis	-4.8	mrad	-5.6	mrad	-15.5	mrad	-6.5	mrad	-3.7	mrad	-11.7	mrad	-4.8	mrad	16.5	mrad
Maximum rotation about Y-axis	9.1	mrad	9.0	mrad	10.3	mrad	6.7	mrad	4.9	mrad	-6.2	mrad	4.9	mrad	19.7	mrad
Maximum rotation about Z-axis	-6.3	mrad	-6.1	mrad	11.8	mrad	-5.6	mrad	-3.6	mrad	8.6	mrad	-4.2	mrad	25.3	mrad

APPENDIX D (FATIGUE CALCULATION)

Block 1. (Similar calculation done for 8 load cases to find cumulative

Calculating the factors for the plate correction factors.

S.No	Cordinates (mm)		Thickness(mm)	Optimized	f (m)	f®	f(w)	f (i)	f (t)	Δσ R	Δσ©	f (s)	Δσ©	n
	y	z	t(mm)	No(+)/Yes(-	Material	Mean stress	Weld Shape	Structure fail	Thickness	(N/mm^2)	(N/mm^2)	Misalignmer	fatigue calulation	
1	516.2	-802.7	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
2	516.2	-827.7	27.5	+	1	1	1	0.9	0.98392783	100	88.5535045	1.0003126	88.5811861	1040
3	486.2	-827.7	25	+	1	1	1	0.9	1	100	90	1.0003126	90.0281338	1040
4	0	-827.7	25	+	1	1	1	0.9	1	100	90	1.0003126	90.0281338	1040
5	-486.2	-827.7	25	+	1	1	1	0.9	1	100	90	1.0003126	90.0281338	1040
6	-516.2	-827.7	27.5	+	1	1	1	0.9	0.98392783	100	88.5535045	1.0003126	88.5811861	1040
7	-516.2	-802.7	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
8	-516.2	0	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
9	-516.2	747.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
10	-516.2	777.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
11	-486.2	777.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
12	0	777.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
13	486.2	777.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
14	516.2	777.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
15	516.2	747.4	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040
16	516.2	0	30	+	1	1	1	0.9	0.96948075	100	87.2532671	1.0003126	87.2805423	1040

c=	0.15	Compressive stress				Δσ s,max	160	applied peak stress		
f (w)	1	Non grinded				Δσ s	159	bending portion		
	1.15	Disc grinded				K'	0.95			
	1.4	Rounded Best				K (m)	1	Stress increase factor due to misalignm		
f (i)	0.9	Without notches								
	0.95	With notches and rounded								
	r=	100	mm	With notch (f i<1)						
n	0.17	welded	t>25mm							
	0.1	toe ground								
Δσ R	100	Butt weld A1 to A6								

S.No	Cordinates (mm)		Thickness(mm)	Optimized	Max Cycles	Δσ ch	max Δσ	m	$\sum_i^n \left(\frac{\Delta\sigma_{ch,ch}}{\max \Delta\sigma}\right)^m$	ϒ (mf)
	y	z	t(mm)	No(+)/Yes(-)						
1	547.6	-803.7	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
2	547.6	-828.7	27.5	+	2000000	88.5811861	184.384789	3	457015.524	1.15
3	517.6	-828.7	25	+	2000000	90.0281338	187.396661	3	479778.986	1.15
4	0	-828.7	25	+	2000000	90.0281338	187.396661	3	479778.986	1.15
5	-517.6	-828.7	25	+	2000000	90.0281338	187.396661	3	479778.986	1.15
6	-547.6	-828.7	27.5	+	2000000	88.5811861	184.384789	3	457015.524	1.15
7	-547.6	-803.7	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
8	-547.6	0	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
9	-547.6	746.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
10	-547.6	776.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
11	-517.6	776.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
12	0	776.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
13	517.6	776.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
14	547.6	776.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
15	547.6	746.4	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15
16	547.6	0	30	+	2000000	87.2805423	181.677455	3	437178.491	1.15

Symbols	**Meaning**
Δσ ch	Corrected Mean Stress
max Δσ	Maximum Mean Stress
ϒ (mf)	Partial Safety Factor (Eurocode 3)
f®	Mean stress correction factor
f(w)	Weld Shape correction factor
f (i)	Structure failure correction
f (t)	Thickness correction factor
Δσ R	Stress range Weld property from SN curve
Δσ©	Corrected stress range
f (s)	Misalignment correction factor

APPENDIX D