

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL

**MATHEMATICAL MODELING AND ANALYSIS OF SHIP-
MOUNTED CRANES WITH SHOCK ABSORBER**



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A THESIS OF MASTER OF SCIENCE
DEPARTMENT OF MECHANICAL ENGINEERING

ASSIST. PROF. DR. SAEED LOTFAN

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LİSANS ÜSTÜ EĞİTİM ENSTİTÜSÜ

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ABSTRACT

This thesis aims to provide a detailed examination of the formulas specified in the Class rules for lifting equipment, offer technical support for the development of these rules, and serve as a reference for future formula improvements and experimental studies. Standards in the maritime industry are developed under the leadership of international technical committees after years of detailed and rigorous work.

Lifting applications on offshore structures differ from cranes used in land facilities. The primary reason for this is the necessity to design cranes in an environment constantly in motion. Factors such as wave height, hook speed, crane spring constant, lifted load, wind speed, ice load, and mass accelerations combine to create various load combinations for crane design. Shock absorbers positively contribute to the lifting capacity, fatigue resistance, and operational safety of the crane.

Since this thesis is based on referencing Class rules, it adheres strictly to the main rule framework. To provide realistic inputs for dynamic systems and expand the scope of the research, a crane design program was developed to determine the spring and damping coefficients for 16 different cranes and 3 different shock absorbers. The accuracy of this program was verified using Finite Element Analysis (FEA) calculations.

The evaluation results showed no significant differences between the load reduction coefficients of cranes without shock absorbers designed according to Class rules and those obtained from simulations. Treating the shock absorber directly as a spring and incorporating it into the load reduction formula is not a healthy method. Resonance effects and the impact of the damping coefficient on the system are also present. Therefore, if such a load reduction is to be applied, the technical details of the shock absorber must be well understood, and solutions must be produced through dynamic system modeling. As stated in the Class rule, the simple load reduction approach when a shock absorber is used should not be the solution for these complex situations.

Another finding was that in dynamic system modeling, as the boom length and weight increase, an increase in stress was observed. The boom weight is not included as a load in the load reduction coefficient formula. In this context, future studies could include time-dependent FEA analyses and detailed experimental studies with strain gauge-equipped crane structures.

Keywords: Lifting appliances, Shock absorber, Dynamic system modeling, Sea state, Class society-based crane design

ÖZET

Bu tezin ana amacı, kaldırma ekipmanlarına yönelik Klas kurallarında belirtilen formülleri daha ayrıntılı bir incelemesini sağlamak, kuralların geliştirilmesi için teknik destek sunmak ve gelecekteki formül iyileştirmeleri ve deneysel çalışmalar için bir referans olmaktır. Denizcilik endüstrisindeki standartlar, yılların detaylı ve ciddi çalışmaları sonucunda uluslararası teknik komiteler öncülüğünde hazırlanır.

Açık deniz yapılarındaki kaldırma uygulamaları, kara tesislerinde kullanılan vinçlerden farklıdır. Bunun başlıca nedeni, sürekli hareket eden bir ortamda vinçler tasarlama gerekliliğidir. Dalga yüksekliği, kanca hızı, vinç yay sabiti, kaldırılan yük, rüzgar hızı, buz yükü ve kütle ivmeleri gibi faktörlerin birleşik etkileri, vinç tasarımı için çeşitli yük kombinasyonlarına neden olur. Şok emiciler, kaldırma kapasitesine, yorgunluk direncine ve vinç operasyonel güvenliğine olumlu katkıda bulunur.

Tezimizde Klas kurallarını referans alarak yola çıkıldığından, ana kural kurgusundan uzaklaşmamıştır. Dinamik sistemler için gerçekçi girdiler sağlamak ve araştırmanın kapsamını genişletmek amacıyla, 16 farklı vinç ve 3 farklı şok emici için yay ve sönümleme katsayılarını belirleyen bir vinç tasarım programı geliştirilmiştir. Bu programın doğruluğu, sonlu elemanlar analizi (FEA) hesaplamaları ile doğrulanmıştır.

Değerlendirme sonuçları, sınıf kurallarına göre tasarlanan şok sönümleyici olmayan kreyinlerin yük azaltma katsayıları ile simülasyonlardan elde edilen sonuçlar arasında önemli farklılıklar olmadığını göstermiştir. Şok emiciyi doğrudan bir yay olarak ele almak ve yük azaltma formülüne dahil etmek sağlıklı bir yöntem değildir. Rezonans etkileri ve sönümleme katsayısının sistem üzerindeki etkisi de mevcuttur. Bu nedenle, böyle bir yük azaltımı uygulanacaksa, şok emicinin teknik detayları iyi anlaşılmalı ve çözümler dinamik sistem modellemesi yoluyla üretilmelidir. Sınıf kuralında belirtildiği gibi, şok emici kullanıldığında basit yük azaltımı yaklaşımı bu karmaşık durumlar için çözüm olmamalıdır.

Bir diğer bulgu, dinamik sistem modellemesinde, bom uzunluğu ve ağırlığı arttıkça gerilimlerde artış gözlemlenmiştir. Bom ağırlığı, yük azaltma katsayısı formülünde bir yük olarak dahil edilmemektedir. Bu bağlamda, gelecekteki çalışmalar, zaman bağımlı FEA analizleri ve gerinim ölçerlerle donatılmış vinç yapıları ile detaylı deneysel çalışmalar içerebilir.

Anahtar Kelimeler: Kaldırma donanımları, Amortisör, Dinamik sistem modellemesi, Deniz durumu, Klas kurallarına dayalı vinç tasarımı

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LIST OF SYMBOLS AND ABBREVIATIONS

$t_{1.2.3.4}$	: Time of the lifting graphic
V_H	: Drum hosting speed during time zones
$y_{1.2.3}$	: Lifting position vertical position
ω	: Period of the sinusoidal wave
M	: Mass
c_s	: Damping constant of the shock absorber system
k	: Spring constant
F	: Force
M	: Mass
C	: Damping coefficient
K	: Spring coefficient
u	: Vertical input
$x_{1.2.3.4.5.6}$	: State Variables
σ_{per}	: Permissible Stress
v	: Safety factor
R_{eH}	: Yield strenght of material
K	: Friction allowance
η	: Hoisting performance
n	: Number of sheave block
SWL	: Safety Working Load in calm water
$\overline{SWL}$	: Reduced Safety Working Load for working in a seaway
$H_{1/3}$	: Significant wave height
T_0	: Average wave period
V_H	: Minimum hook velocity
V_D	: Vertical velocity of cargo deck
f_e	: Influence factor
ψ	: Hoist load coefficient
$\bar{\psi}$	: Hoist load coefficient for working in a seaway
φ	: Hoist load coefficient
g	: Gravitation constant 9.81 m/s ²
c	: Spring constant of lifting appliance
A	: Dimension of the crane see figure 3.15 and 3.16
B	: Dimension of the crane see figure 3.15 and 3.16
C	: Dimension of the crane see figure 3.15 and 3.16
D	: Dimension of the crane see figure 3.15 and 3.16
E	: Dimension of the crane see figure 3.15 and 3.16
F	: Dimension of the crane see figure 3.15 and 3.16
G	: Width of the crane boom see figure 3.15 and 3.16
H	: Height of the crane boom see figure 3.15 and 3.16
J	: Position of the crane see figure 3.15 and 3.16
K	: Crane boom, tower connection point crane see figure 3.15 and 3.16
$K_{1,2}$	: Crane boom, tower connection point crane see figure 3.15 and 3.16
L	: Dimension of the crane see figure 3.15 and 3.16
f°	: Angle of hydraulic cylinder see figure 3.15 and 3.16
t_1	: Thickness of the crane boom top and down plates
t_2	: Thickness of the crane boom side plates

F_w	: Wind load
$\bar{c}$	: Form coefficient,
A	: Area acted on by wind
V	: Wind Speed
Q	: Dynamic Pressure
q	: Wind pressure
$M_{z,j...}$	: Moment
$F_{z,j...}$	: Force
$W_{z,j...}$	: Section Inertia
$\sigma_{z,j...}$	: Bending Stress
$\tau_{z,j...}$	: Torsional Stress
G_{dw}	: Mass of the boom
ρ_{steel}	: Density of the steel
M_{spring}	: Moment of the slewing ring
$\emptyset L_b$	: Slewing ring bolt circle
$\bar{B}$	: Diameter of the slewing ring bolt assembly hole
P	: Outside diameter of the mass
$\emptyset L_a$	: Outside Bolt circle
$\emptyset L_i$	: Inside Bolt circle
α_{ct}	: Notch factor coefficient
a	: Notch factor table dimension
b	: Notch factor table dimension
P	: Design pressure
D_0	: Outside diameter of cylinder tube
e	: Joint efficiency
$\bar{c}$	: Corrosion allowance
t	: Required wall thickness of the cylinder tube
R_{eH}	: Yield strength of material
R_m	: Tensile strength of material
L_{12}	: Buckling length of the cylinder
I_1	: Second moment of area for the cylinder tube, [mm^4]
I_2	: Second moment of area for the piston rod, [mm^4]
D_o	: Outer diameter of the cylinder tube
D_i	: Inner diameter of the cylinder tube
F_E	: Young's modulus of elasticity
F_a	: Actual maximum pushing force
L	: Length of fully extracted hydraulic cylinder between mountings
$k_{Crane\ Structure}$	: Crane spring coefficient
$P_{1,2}$	: Pressure of the shock absorber
$V_{1,2}$	: Volume of the shock absorber
K	: Adiabatic gas constant
Fr	: Friction allowance
nos	: Number of sheaves in the tackle
TL	: Turk Loydu
DNV	: Det Norske Veritas
AHC	: Active Heave Compensation

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1. INTRODUCTION

Before we delve into the topic, it's important to clarify the difference between the terms "Crane" and "Winch," which often cause confusion.

The word "Winch" originates from the English word "Winch." A winch is a general term for machines used to pull or lift loads in a single direction, often involving the winding of a cable onto a drum. On the other hand, "Crane," derived from the English word "Crane" and the German word "Kran," refers to machines that not only lift and lower loads but also allow movement in other axes. Cranes typically have a winch for the movement of the hook block. Due to their versatility, cranes are commonly used in places such as ports, workshops, warehouses, foundries, and more [1].

A crane, also known as a winch, is a machine designed for the lifting and transportation of a load suspended by a hook or other load-bearing device. Today, it is widely used in heavy industry. Although lifting and carrying loads by suspending them is not the preferred method from a safety perspective, it is often chosen out of necessity and practical.

The term "Mooring Winch" refers to a device used for various maneuvers related to a ship's horizontal docking, mooring to a dock, connecting to a tow, or ship-to-ship mooring. This winch is equipped with a grip mechanism and facilitates the vertical movement of an anchor or cable [2], as shown in **Figure 1.1**.



Figure 1.1: Mooring winch on the vessel deck [2].

Example of a winch designed for use on cranes with double drums and wheels on double beams, normally equipped with closed brakes and encoders is also shown in **Figure 1.2 [3]**.



Figure 1.2: Gantry crane winch [3].

Example of a general workshop crane with electric motor assistance, providing movement in 3 directions (x, y, z) is shown below [4].



Figure 1.3: Gantry crane [4].

A machine designed for lifting and transporting a load suspended by a hook or other load-bearing mechanism is called a crane. Nowadays, cranes are widely used in heavy industries. While suspending and transporting loads may not be the preferred method from a safety perspective, it is often chosen out of necessity and practicality. Especially due to the convenience of using pulleys and ropes, it allows tasks that cannot be easily

accomplished by human strength alone to be performed comfortably. For example, it facilitates operations such as loading and unloading ships at ports, handling offshore structures, transporting products and materials in construction projects, and tunnel excavation [5].

1.1. Introduction to Crane Components

An example of a crane, along with details related to its components, is depicted in **Figure 1.4**. Crane designs generally consist of three main parts: structural, mechanical, and drive systems. These parts are discussed in the following sections.

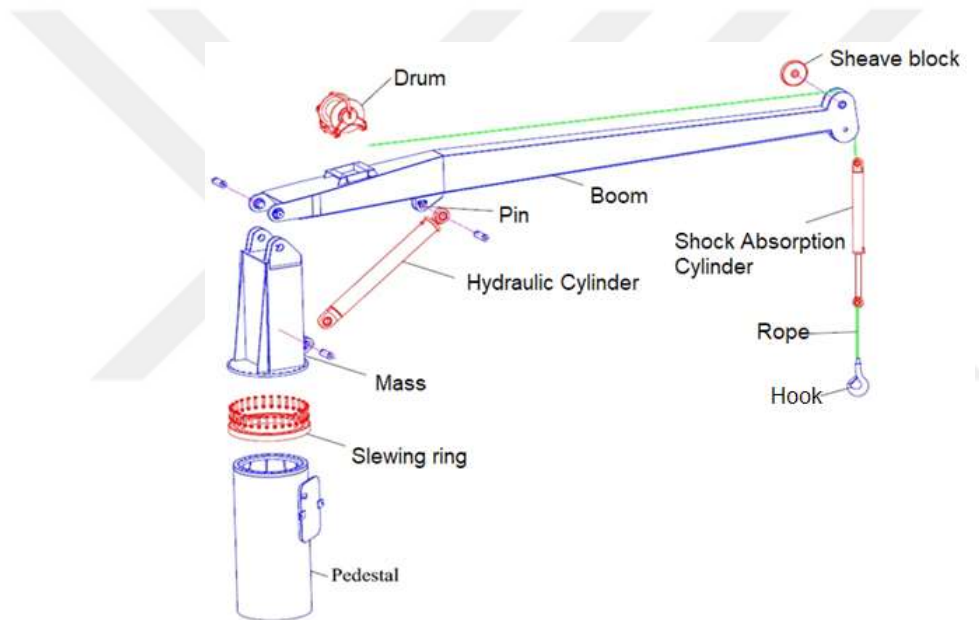


Figure 1.4: Hydraulic power deck crane with shock absorber.

1.1.1. Structural Components

These are elements that directly support the load and are exposed to high bending and shearing forces. Strength evaluations are crucial during the design process. These components are often parts that change position due to the effect of the driving system. In gantry cranes, these elements include beams, booms, towers, chassis, etc. They are typically produced through welded fabrication, so welding knowledge and weld quality evaluation are essential in the design process.

In general applications, high-weldability steels such as A Grade, AH, DH, S355, S235, S690, etc., are preferred. The welding sequence is crucial for the weldability of the material used, especially considering the welding quality and preheating due to the working environment conditions of the crane. Charpy notch values and elongation values for breakage and fracture criteria should be particularly considered when using high-strength steel, and the welding position should be evaluated in terms of the load's strain [7]. The important structural components are explained in the following subsections.

Booms are the parts that are exposed to high bending moments and torsional stresses in the structure. In compact spaces, telescopic, inward-folding models are often preferred, as they occupy less space when folded. In open deck-type structures where there are no elements obstructing the workspace, fixed closed-box booms are generally used as they allow for more economical and lightweight designs. In designs where the maximum bending stress occurs, at the connection point of the hydraulic cylinder, the cross-sectional moment of inertia is increased. For very tall structures, where wind forces have a significant effect, pipes with lower aerodynamic resistance coefficients are used. Since spring constants are crucial in marine cranes, in addition to general strength calculations, especially vertical displacements, local and general buckling strengths, and the coincidence of the ship's natural frequencies with wave periods should also be considered. In the designs of closed boxes, welding sequences and, as far as possible, full penetration welds should be preferred. If full penetration welding cannot be applied, the welds should be made in areas close to the neutral axis where the stresses are low after welding [8].

Towers are structures that connect the pedestal and the turn gear to the boom. After welding the turn gear connection, surface undulations created by chip machining should be removed. The welds of the pedestal should be fully penetrating for ship welding and non-destructive tests (NDT) should be performed [9].

Pins are elements that form the center of rotation of moving parts. Typically, materials such as Duplex Stainless Steel (1.4462 X2CrNiMoN22-5-3), 42CrMo4, Ck45, 316L, etc., are preferred depending on the application. After welding the steel bodies to which the pin is connected (e.g., tower and boom, etc.), chip machining should be carried out to ensure parallelism and socket tolerances on the working surfaces.

Generally, pins rotate on one part and are fixed to the other part. Fixation methods are specified in the DIN standard [6].

Bearings are often used where pins rotate. Bronze-based materials are preferred for bearings that operate in the area where they move. Bronze is softer than steel and does not exhibit hardening under impact, so it does not damage the pin. However, over time, bronze materials will need to be replaced. If there is a risk of deterioration in the bearings and bushings over time, special attention should be paid to fixation. Even if there is clearance in the pins, measures should be taken to prevent them from falling out of the socket. In applications where lubrication is a significant problem and working environments are challenging, graphite bronze bearings can be preferred. In any case, a separate lubrication line should be drawn for each lubrication point, and many lubrication points should not be connected in parallel on the same oil line. Since the oil will always move from the point of low pressure, not all points will be lubricated, leading to damage over time. Another common mistake made during the design phase is miscalculations based on the assumption that pins only carry cutting loads. Especially when there is clearance in the sockets where the pins sit, the pins will not behave like built-in elements over time and will begin to carry bending loads. Therefore, spread loads should be considered on the seating surfaces, and bending and cutting loads should be included in the criteria for straining failures. The simplified pin calculation formula from Brueu Veritas rules is included above [10].

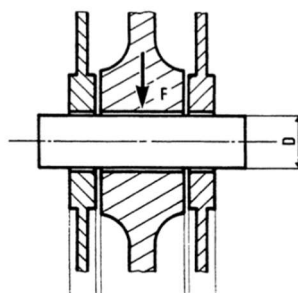


Figure 1.5: Details of the lifting lug [10].

1.1.2. Mechanical Components

Mechanical components are parts responsible for driving the movement of the load and the structural elements of the crane. They typically derive their power from sources such as electric motors, hydraulic rotary elements, and linear mechanical components,

for example, electric motors, hydraulic motors, linear actuators, hydraulic cylinders, etc. These components are designed to convert the power source, which can be continuous oil flow or electrical current, into mechanical motion.

In mechanical component design, considerations go beyond just strength evaluations. They also involve the mass inertias of the structure, service factors of the elements used, and the effects of the mechanism on performance. Examples of mechanical components include pulleys, hydraulic cylinders, crane drums, gear and reducer assemblies, tower rotation gears, pin mechanisms, gear sets, hook blocks, etc. Wear effects are important in the design of these elements. Therefore, design aspects such as geometric tolerances, surface hardening values, bearing specifications, pin design, gear design, rope selection, hook block selection, and tribology are essential, and they often follow machinery-based design principles.

Materials used in the design of these components are often hardenable steels such as 42CrMo4, 8620, 18CrMo4 for gears, shafts, and wheels. In addition, materials like cementation and quenched steels are preferred. For structural elements, materials like Ck45, S355, S235, GS60, etc., are generally chosen. Important mechanical parts are defined below.

Hoisting Winches are equipment used in cranes to facilitate the movement of wire rope. The cylindrical body is manufactured by machining a seamless tube on a lathe. The side flanges of the tube must have a height that provides a minimum safety margin of 2.5 or 3 times the cable's diameter after all the windings are completed. As the number of windings on the drum increases, the diameter of the rope's cross-sectional area also increases, which can reduce the torque value and, consequently, the tensile force of the rope. This should be taken into consideration during the design phase. The length of the rope operated by the drum, additional band brakes on the drum, and grooved guides to ensure the rope winds properly can be added according to safety requirements in practice.

In hoisting winches, both to protect critical sub-components from the environment and to provide a compact design, sintered bronze lamella disk brake systems, and planetary gear reducers are generally placed inside the drum tube (as shown in **Figure 1.6**). The drums are typically driven by a hydraulic motor or an electric motor. In marine

applications, it is also expected that a manual brake release is present to disengage the brake in the event of a system blackout [9].

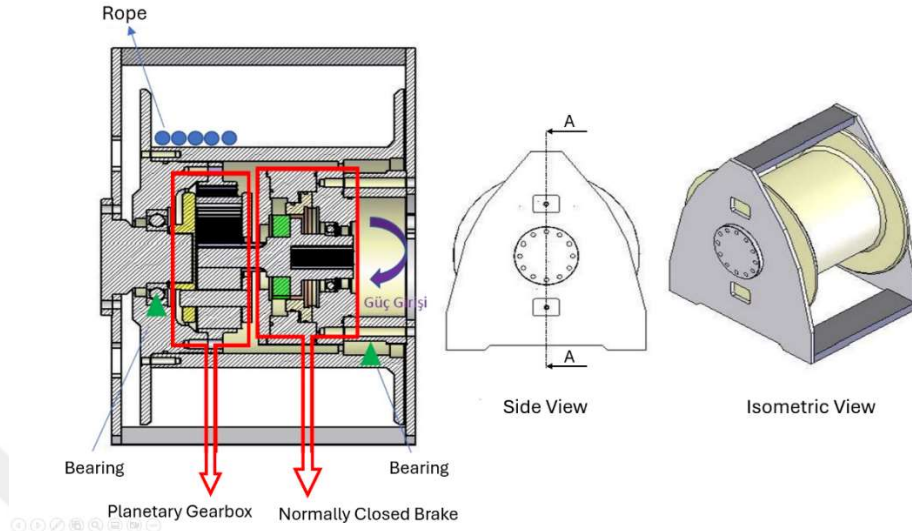


Figure 1.6: Details of the planetary gearbox hoisting winch.

Hydraulic cylinders, as seen in **Figure 1.7**, provide linear motion by high-pressure oil flow (usually between 150-350 bars, depending on the pump's capacity) through oil inlets on both sides. Honed tubes and ground stainless steel or chrome-plated rods are used. These precise surface tolerances increase seal durability. Stainless steel rods are chosen in areas with high corrosion. In the design of these components, buckling calculations, oil flow rates, relationships with valve blocks, and details such as whether the design damages seal elements during assembly should also be considered [11], [12].



Figure 1.7: Section detail of a hydraulic cylinder [11].

Slewing rings can be likened to bearings with enlarged diameters. Their surfaces are hardened, and with the help of balls made of high surface quality and hardness, very low frictions are achieved. Both external and internal gears are cut, allowing for rotational movement. To prevent excessive growth of pinions and reducers dimensionally, pinions with 14-15 teeth are preferred. Thus, pinion gears do not undergo undercutting when external gears are not deeply cut. Inside and outside rings are typically made of 42CrMo4 material using special grinding machines.

The surface hardness of the pinion gear should be greater than that of the rotation gear. Especially in towers and pedestals, the most critical loads are moment loads. The transmission of these moments with the help of balls under very high loads is considered. Depending on the usage, models with ball or cylindrical bearings are available. Ball bearings can be calculated for lifetime according to ISO/TS16281. When design checks are performed, the most critical element often appears in the calculation of the bolt circle. Bolt torque values are crucial, and assembly operations should be carried out according to the reference torque value given by the manufacturer. Generally, 8.8 and 10.9 quality bolts are recommended; 12.9 quality bolts are not preferred [13], [14].

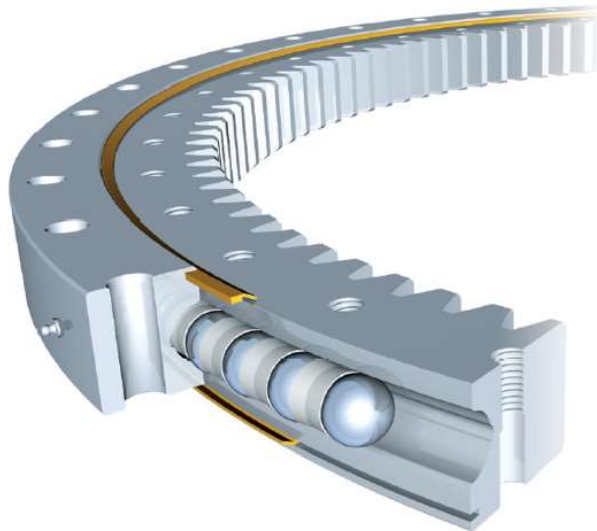


Figure 1.8: Section detail of a slewing ring [14].

Hook blocks are the points where the load is attached to the crane (**Figure 1.9**). The quality and variations of hooks depend on factors such as the lifting capacity and the

crane's intended use. Immediately above the hook blocks, thrust bearings with high compressive strength are used. One common mistake made in production is the thread profile used for hook nuts and teeth on the hook. Although metric threads are opened, circular-shaped special tooth profiles specified in DIN standards must be used to prevent damage to the surfaces subjected to high compressive loads over the years. To increase the number of cycles, the number of rope pulleys on the hook block is increased, but this results in a fourfold decrease in the speed of the crane in vertical direction. Since the speed of each rope pulley will differ, the arrangement of the ropes in hook blocks, especially in multi-rope and double drum hook blocks, must be designed to prevent the hook from flapping [15].

In terms of design, two important details should be considered. The diameter of the pulley groove, where the rope passes, is crucial. If this diameter is chosen lower than the one specified in the standard, it can cause the rope to break and eventually snap over time. In addition, especially in marine cranes, if the pulley at the end of the boom works with wavy seas, much higher lateral loads occur compared to land applications. Therefore, the lateral stresses on the surfaces of the pulleys due to bending and the increase in loads due to the moment effect in bearings can cause rapid damage to the bearings, especially in frequently used pulley blocks. Apart from selecting a standard product for land use, it must also be confirmed with detailed calculations.

Design dimensions for the groove where the rope sits are specified in standards, and it is sufficient to comply with the standard. Additional measures should also be taken to prevent the rope from coming out of the groove [9], [16].

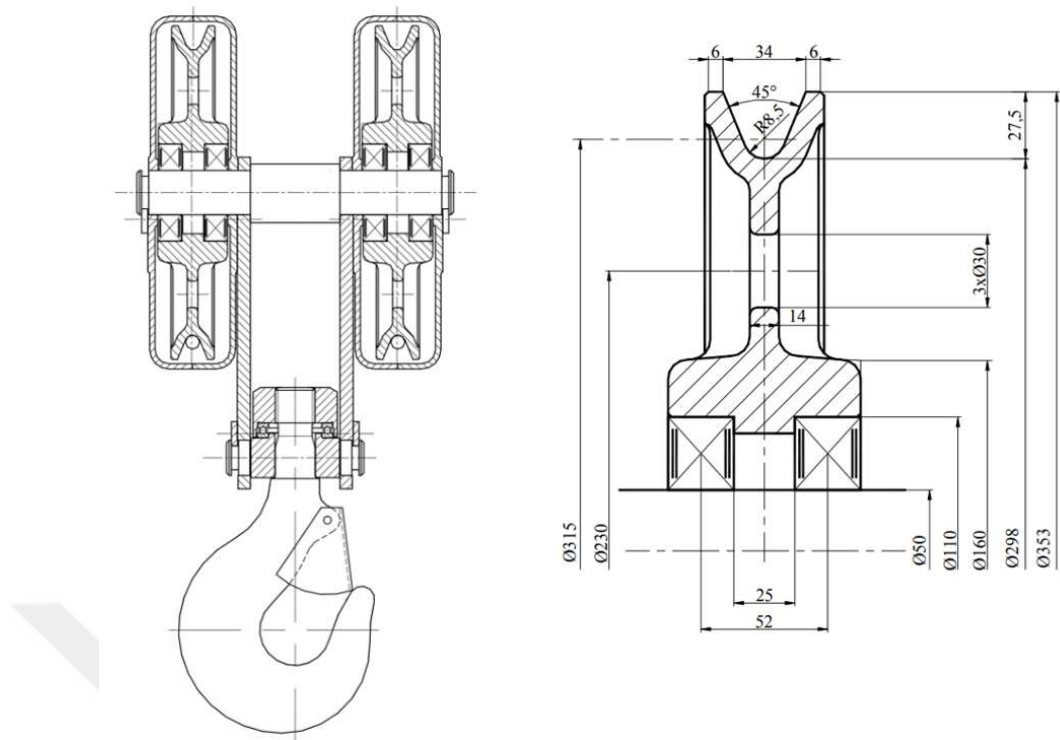


Figure 1.9: Section detail of a sheave block and a hook [16].

In the table below, sheave block and drum diameters for different applications are provided. The pulley and drum diameters in the table are created according to the determined rope diameter. For example, 9 d means the rope diameter will be 9 times.

Table 1.1: Sheave block and drum diameter selection table [9].

Application	Sheave Groove Diameter (d) min.	Cable Drum Diameter (d) min. (groove less)	Cable Drum Diameter (d) min. (grooved)	Nominal Tensile Strength of Wire Rope (N/mm ²)
Ropes not working under load	9 d	10 d	9 d	1570 N/mm ²
Rarely used boom equipment	14 d	16 d	12.5 d	1570 N/mm ²
Frequently used boom equipment	18 d	20 d	16 d	1770 N/mm ²
Deck cranes, type A and B	20 d	22 d	18 d	1770 N/mm ²
Deck cranes, type C1	22 d	25 d	22 d	1770 N/mm ²
Deck cranes, type C2	25 d	28 d	22 d	1770 N/mm ²

1.1.3. Drive Systems

Drive systems are responsible for generating motion within a crane. They can be powered by various sources such as electric power, internal combustion engines, or manual force. Continuous rotating power is connected to a gear reducer, hydraulic pump, or compressor. This allows mechanical components to facilitate the movement of structural elements.

One can think of drive systems as being like the cardiovascular system in our bodies. The power to generate movement comes from here. Mechanical components, in this analogy, are like our muscles. They change their structure in response to the incoming power, resulting in motion. The structural elements can be compared to our bones, providing stability and structure. At the joints, motion is enabled, similar to how our joints work.

In the context of cranes, hydraulic power units are commonly used to drive the system. These units contain various components such as solenoid valves, electric motors, electrical panels, thermostats, cooling fans, oil heaters, etc. Due to the presence of these components, hydraulic power units are often placed in enclosed areas on ships where high IP (Ingress Protection) ratings are required. Pipes are used to transport oil from these enclosed spaces to service areas on the ship where it is needed.

A typical hydraulic circuit for a ship crane requires three main movements: hoisting, slewing, and luffing (raising or lowering the jib). The hydraulic power unit is powered by an electric motor, which drives the hydraulic pump. Hydraulic oil is then directed to the control valve, which is responsible for directing the flow of oil to various parts of the crane. Safety valves on the control valve block determine the maximum operating pressure of the system.

Within the control valve, electrical solenoid valves, hydraulic pilot-operated valves, and mechanical mechanisms condition the flow direction of the oil and control the opening of specific ports. This, in turn, causes the oil to move in the desired direction, allowing hydraulic cylinders to produce linear motion or hydraulic motors to produce rotational motion.

To prevent sudden load imbalances and uncontrolled movements, balance valves are used. These valves ensure smooth and controlled movements, enhancing the safety and efficiency of the crane.

Overall, hydraulic systems play a crucial role in the operation of ship cranes, providing the necessary power and control to perform lifting and maneuvering tasks safely and effectively [17], [18]. An example of hydraulic power pack and flow diagram is shown in **Figure 1.10**.

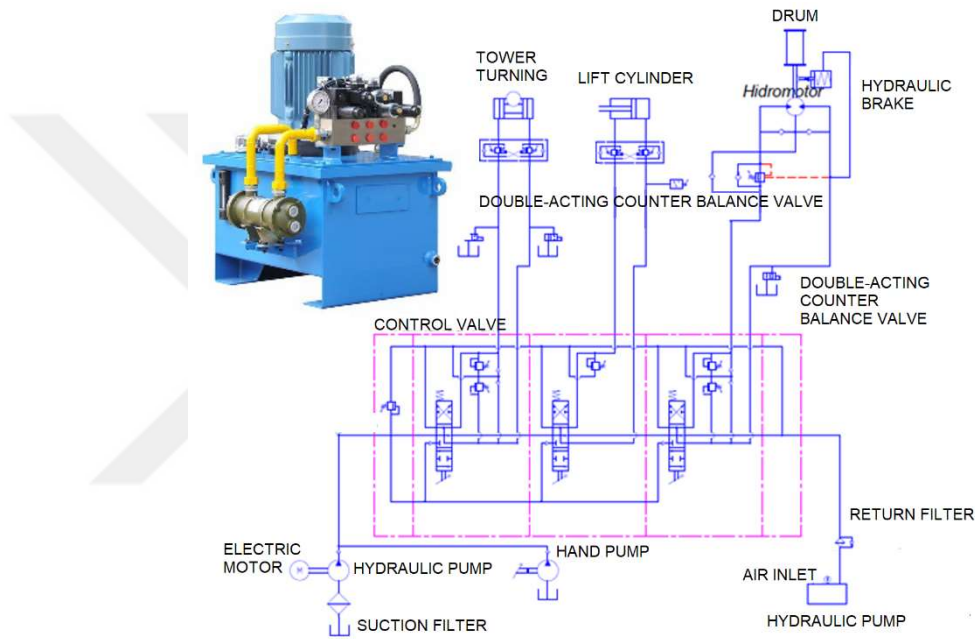


Figure 1.10: Hydraulic power pack and flow diagram [17].

According to the studies and references mentioned so far, two main simulation approaches have been identified: one that includes all the loading and environmental effects on the crane due to the ship's 3D movements, and a simpler method that considers the system as moving only in the z-axis. In our study, we have particularly modeled the system as if it were moving only in the z-axis because the fundamental formulas of Türk Loydu (TL) and Euro Norm (EN) rules are structured this way. This method was chosen to stay aligned with the references.

1.2. Differences Between Crane Applications in Floating and Land-Based Structures

This study explores the disparities between crane applications in floating structures and their counterparts in land-based structures. Whether on land or at sea, cranes share a common foundation in their general design and engineering calculations. Particularly in crane design, European Material Handling Federation (FEM) and DIN standards have played a pivotal role in shaping design principles for many years. Subsequently, these standards evolved into international publications, such as ISO and EN standards [19], [20], [21].

1.2.1. Maritime Applications

In maritime applications, the compliance of lifting equipment designed and manufactured is obligatory to be evaluated and documented by independent entities known as "class" organizations. Each of these class organizations has its own set of technical regulations and formulas, encompassing international standards. Therefore, our focus here lies more on differences in fundamental principles in practice rather than variations in formulas.

Maritime equipment must also conform to criteria defined by the International Labor Organization (ILO) due to the different legal requirements arising from the importance of cargo value and worker safety and health. Decisions made during the 1979 session of the ILO Convention concerning Occupational Safety and Health in Dock Work (No. 152) form the basis for detailed explanations of technical aspects directly concerning worker health and safety, including employee training, certification, periodic testing, load testing, and others. These aspects are comprehensively explained by international authorities to guide law enforcers [22].

One of the most significant distinctions between land-based and maritime crane applications is the presence of inspection and evaluation matrices. In maritime applications, the value of transported goods and the significance of worker health outweigh the value of lifting equipment. Consequently, independent inspection mechanisms have been established to ensure technical compliance for insurance

activities. These inspections categorize vessels under notations, dictating different inspection criteria. This process results in detailed documentation of insurance premiums and risk categories for the vessel [9], [20], [22].

1.2.2. Land-Based Applications

In land-based crane production, there is less emphasis on inspecting crane components in the factory by independent auditors or implementing an internal audit system. In contrast, maritime applications rely on specialized individuals to conduct tests, measurements, inspections, and certification processes. Specific inspection criteria have been developed, outlining how each component should be tested, and which documents should be obtained. The EN10204 standard specifies inspection criteria for products.

For instance, a "3.2 certificate" signifies that an impartial observation firm external to the manufacturing entity conducted the inspection. A "3.1 certificate" indicates that an internal quality control representative within the manufacturing entity conducted the inspection, provided it complies with the criteria of the independent audit committee. Lastly, a "2.2 declaration by the manufacturer" implies that the manufacturer's statement alone is sufficient [23].

1.2.3. Design Criteria and Environmental Considerations

From a design criteria perspective, maritime applications may entail different geometric requirements and distinct stress and torsion criteria due to the movements experienced by floating structures. Some fundamental considerations include:

Extra accelerations and angular deviations along the x, y, and z axes occur due to the movements of floating structures, necessitating their assessment.

Additional calculations must be performed in maritime applications to account for lifting loads under turbulent sea conditions, a factor not typically considered in land-based applications. This involves evaluating parameters such as the crane's spring constant, wave height, drum speed, considerations for rope slackness, additional systems, shock-absorbing cylinders, and active or semi-active damping systems.

Special measures must be taken to prevent galvanic corrosion in aluminum boat and steel crane connections. Detailed project approvals and certification of subcomponents

are required for marine cranes, including comprehensive certification of the crane's assembly on the floating vehicle. Operator cabins for marine cranes are expected to meet specific requirements depending on the crane's category.

Compliance with ILO criteria is mandatory for marine cranes.

Marine cranes often operate on floating platforms, requiring precise operations under varying environmental conditions. Special systems that detect environmental conditions and adjust the hook position actively or passively are therefore implemented [9], [20], [22].

1.3. Motivation and Scope

The primary motivation behind this thesis is the lack of a detailed perspective on the impact of shock absorber systems on cranes transferring loads in rough seas. Upon conducting a literature review, it was found that the effects of shock absorber systems are either specially designed for specific conditions or addressed through simulation-based solutions. In this thesis, the fundamental design and approval criteria were adhered to using the framework formulas of classification societies and EN standards as the primary references. The goal of this study is to provide a new perspective on existing rules. Frequently encountered cranes with and without shock absorbers, with outreach ranges from 7.5 meters to 20 meters and lifting capacities from 5 tons to 20 tons, were initially designed according to classification society criteria. Subsequently, the results obtained from these simulations were compared with the results of classification society rules [9], [10], [19], [20], [21], [22], [24], [28], [29], [31], [32], [43].

Another unique aspect of our study is that the literature typically involves either a single crane system set up or strength calculations for the crane. The most critical point of our study is that it calculates 16 different cranes, analyzing their behaviors and the variations in responses provided by the formulas for cranes of different sizes. To verify the accuracy of the simulations, the study employed two different dynamic system simulation methods and created a program for strength calculations. The results obtained from this program were also verified using FEA-based calculation programs. No literature study was found that encompasses both strength calculations and

dynamic system modeling, as well as design for 16 different cranes. [9], [10], [19], [20], [21], [22], [24], [28], [29], [31], [32], [43].

In conclusion, it should also be noted that the dynamic model developed in this work is a simplified version, yet it produces reliable results.

The general design workflow for ship cranes progresses as follows; Initially, basic design inputs are determined based on the crane's required outreach, lifting capacity, operational load weight and dimensions, environmental and sea conditions, power system, etc. Subsequently, special requirements arise based on the rules to which the crane will be subject. For instance, TL rules mandate the use of cabins for class B cranes or require special precautions against rope breakage for A-frame cranes used in salvage operations. Hydraulic cylinders must also use special valves to prevent hose bursting. The general design is then carried out according to these fundamental requirements, followed by strength and power calculations for cranes in both calm and rough sea conditions. The fundamental formulas used in ISO and EN standards specify basic load reduction factors for operations in rough seas. However, an examination of the literature and standards reveals that these load reduction factors lack a specialized perspective for shock absorber scenarios.

The purpose of this thesis is to provide a detailed examination of the loads acting on cranes with shock absorber systems in the formulations of classification societies, ISO, EN, and FEM standards, and to serve as a reference for future studies [9], [19], [21].

2. COMPARISON OF ACTIVE AND PASSIVE HEAVE COMPENSATION SYSTEMS

Heave compensation systems are designed to eliminate or reduce the effect of sea-induced vessel motion on a payload. The term active heave compensation is a system where the winch paying out cable on the payload is motor-driven and controlled by a computer system that takes input from sensors measuring vessel motion, cable length, and hook load. The computer calculates the cable length required to maintain a constant hook load force and removes vessel motion-induced cable length by paying out or pulling in the cable at a rate faster or slower than the cable paid out by the ocean-induced vessel motion. The greater the force in the cable, the greater the vessel motion, and as such the computer will reduce cable length to maintain a constant force or heave. High-technology active heave compensation winches are able to control the vessel heave seen at the sensor location to $\pm 5\text{cm}$. The benefit of active systems is they can be calculated in saving potential as their uptime could be directly compared to downtime [24], [25], [26]. The evolution history of heave compensation systems is depicted in **Figure 1.2**.

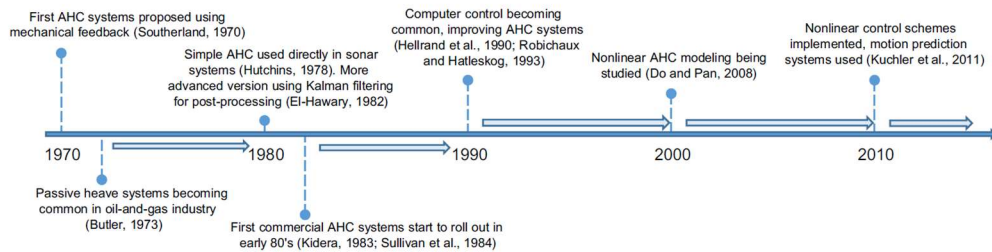


Figure 2.1: History of heave compensation systems [24].

Heave compensation is industry terminology for a system to control the position of an object in a turbulent fluid (**Figure 2.2**). This includes such vessels as ships, offshore oil rigs, and drilling platforms, typically in the vertical or near vertical plane, and in a seaway, it is the ability to maintain position. Traditional deep water offshore construction and drilling operations have in the past generally not used any form of heave compensation and have just accepted the fact that the ocean environment is rough and that there is likely to be significant slippage in the marine riser and drill

string. Such systems can be complicated and expensive, however it is generally accepted that the cost of operating such a system will be outweighed by time saved by having increased operational uptime [27], [28], [29].

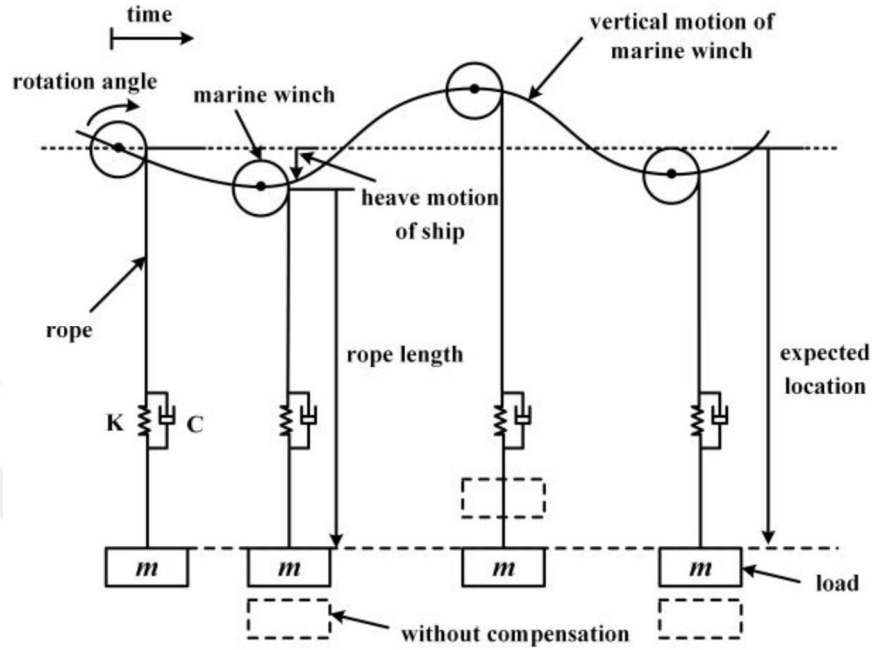


Figure 2.2: Simple diagram of a vertical deep water heave compensation system [29].

2.1. Active Heave Compensation System

The AHC system is mainly used for offshore drilling and lifting appliances. For economic reasons, offshore drilling is generally made on floating vessels, which move depending on wave forces. If the drilling process and lifting operations is done on a fixed platform, wave forces can cause motions of the drilling rig, similar to those experienced on a floating vessel. High wave forces can occur in nearby arctic and stormy areas. By using the AHC system, the drilling rig can still maintain drilling conditions and avoid well control problems caused by motion effects. The measurement devices measure the position and motion of the derrick vessel and the hook load mechanism. The data processing equipment adjusts and controls the compensating force. It automatically activates the system when abnormal force is detected and deactivates it when normal drilling conditions are restored. The heave compensation machinery lifts or slacks the hoist wire, including winch drums and traveling block sheaves. The power supply device can be hydraulic, pneumatic, mechanical, or electrical. It supplies power to the AHC machinery [29].

2.2. Passive Heave Compensation System

The passive heave compensated system is a purely mechanical system designed to maintain a near-constant tension in the lifting wire during the vessel motion. This is achieved by introducing a second wire from the top of the crane boom or drilling platform down to a fixed point on the vessel's structure. The second wire is attached to a compensator unit. Compensation is provided only when the tension in the lifting wire is different from the tension in the compensating wire. Compensation is thus limited to one degree of freedom and no compensation is provided for pendulum motion.

2.3. Differences Between Active and Passive Heave Compensation Systems

The response time of the AHC system, or the speed at which it can react to changes in the motion of the vessel, is dependent on both the control mechanism and energy source. In general, active systems have a faster response time than passive systems. Other factors that affect response time are the masses and stroke lengths of the system components. An AHC with electric power and a small winch drum designed for a small payload will have a very fast response time.

Active systems will always require some form of energy source to power the control mechanism, and both types of systems can make use of electric, hydraulic, pneumatic, or mechanical energy. The cylinder is free to move in the vertical direction and is attached to a shuttle valve, which is in turn connected to the cylinder by hoses. Any vertical movement of the vessel results in the shuttle valve moving and allowing oil to flow between the top and bottom of the cylinder, thus providing a force to move the vessel and payload in opposite directions.

Cost is a major factor in deciding the heave compensation system of a ship. In terms of component cost, the passive heave compensation system is much cheaper. Even at the prototype stage, with physical testing, there are still potential cost savings compared with the build and testing of an active system. The running costs of the system also favor passive technology. The passive system requires no additional power to function because it piggybacks the vessel heave motion from the wire to the payload. This compares with the active system that has a substantial power requirement, translating into higher fuel costs for the vessel.

Another advantage of the passive system is that the hydrodynamic model of the compensation chamber can be tuned to give accurate heave compensation force over a specified range of vessel heave frequencies and amplitudes. Because the hydrodynamic force is predictable, it is possible to design a fail-safe system with known performance. This can be hard to achieve with hydraulic or electric actuators on an active system, especially when the system's health cannot be monitored in real time. Tuning the chamber's hydrodynamic performance can involve physical testing with scale models or with finite element analysis, so it can be costly and time-consuming.

The reliability of the system is very high, leading to less downtime due to component failure. The active system relies on the control system and winch. If either of these components fails, the payload is no longer compensated and large loads on the wire can unroll, damaging the winch and leading to a loss of the load and possibly the ROV itself. The response time of the system, or the speed at which it can react to changes in the motion of the vessel, is dependent on both the control mechanism and energy source. In general, active systems have a faster response time than passive systems. . Other factors that affect response time are the masses and stroke lengths of the system components. An AHC with electric power and a small winch drum designed for a small payload will have a very fast response time.

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The most important limitation of PHC systems is that the system behaves as a simple spring and damping element. The frequency of the ship's movement during waves is important. One of our aims in this study is to observe the natural frequencies of cranes designed according to EN 13852 standards and Class rules (TL and DNV) and their effects on shock receiver systems [20], [30], [31], [32].

2.4. Passive Shock Absorption Systems for Cranes

In this thesis, we are focusing on the passive shock absorption system for the crane. Passive shock absorption systems in cranes are important technological features designed to make lifting operations safer and more efficient. These systems aim to reduce the vibrations and shocks generated during the lifting of loads, creating a safer environment for both crane operators and the loads being lifted. Here's more information about passive shock absorption systems in cranes. Passive shock absorption systems are engineered to absorb sudden vibrations and impacts that occur during crane lifting operations, enhancing the stability and safety of load handling. These systems enable more precise and efficient crane operations.

Operating principle of the passive shock absorption systems are typically designed using hydraulic or pneumatic dampers or shock absorbers. They are used to support crane cables and the hook carrying the load. As the crane moves or during load handling, these systems absorb the vibrations, ensuring smoother and more stable load movement.

The primary advantages of passive shock absorption systems include:

Passive shock absorption systems offer several key advantages that significantly enhance the performance and safety of crane operations. One of the most important benefits is safety. These systems contribute to a safer working environment by preventing load drops and sudden jolts, which can be hazardous for crane operators. Additionally, they help avoid system resonance and reduce the impact of slack rope wave conditions, further minimizing the risk of accidents.

Another major advantage is precision. Passive shock absorbers enhance the accuracy of load handling, allowing for easier positioning and securing of loads with greater control and confidence. This level of precision is especially valuable in operations that require careful handling of heavy or delicate materials.

They also play a vital role in equipment protection. By dampening vibrations and absorbing shocks, these systems reduce mechanical stress on the crane's components, thereby extending the overall lifespan of the equipment and lowering maintenance costs.

From an operational perspective, passive shock absorption systems enhance efficiency. The reduction in unwanted movements allows for quicker and smoother

lifting processes, which translates into faster project completion times and increased productivity.

These systems find wide application in various sectors, including large industrial cranes, port and ship cranes, construction cranes, and other heavy-load lifting operations. Their ability to ensure safe, accurate, and efficient performance makes them a valuable technological advancement in modern crane systems.



3. CREATING THE SIMPLIFIED DYNAMIC MODEL OF A PASSIVE HEAVE COMPENSATOR CRANE BY VESSEL-TO- VESSEL DECK TRANSPORTATION

According to the literature, advanced and simplified methods are used in the design of the loads acting on offshore cranes. In advanced method applications, the modeling approach is created by preparing 3D equations of motion by configuring motion and rotation equations in all directions for the ship to which the crane is attached and the ship on which the load will be lifted, as well as a simplified model that constructs the system with mass-spring damper relations in the vertical, can also be preferred. This is the basic approach of class rule formulas for absorber selection, especially used in offshore operations. In this study, we will prefer the simplified method. The main reason for choosing this method is the ease of calculation and modeling setup. This method offers practical solutions in both surface and seabed operations. The forces resulting from the forward-backward movements and yaw movements of the ship were added during the main design of the crane with the help of empirical formulas in TL rules [29], [31], [32].

In this method, movement velocities and accelerations are applied positively unless stated otherwise. Hydrodynamic forces are applied by simple equations. Results are evaluated with safety margins. In the simplified model system, 2 main factors are designed to affect the change of load position. Sinusoidal waves created by the sea are considered according to the sea condition (As shown in **Figure 3.1**). The expected speeds of the waves and the hook are stated in TL Rules. The graph of the wave was created in MATLAB according to the period, T_0 , and significant wave height values specified in the Table 3.1 [9].

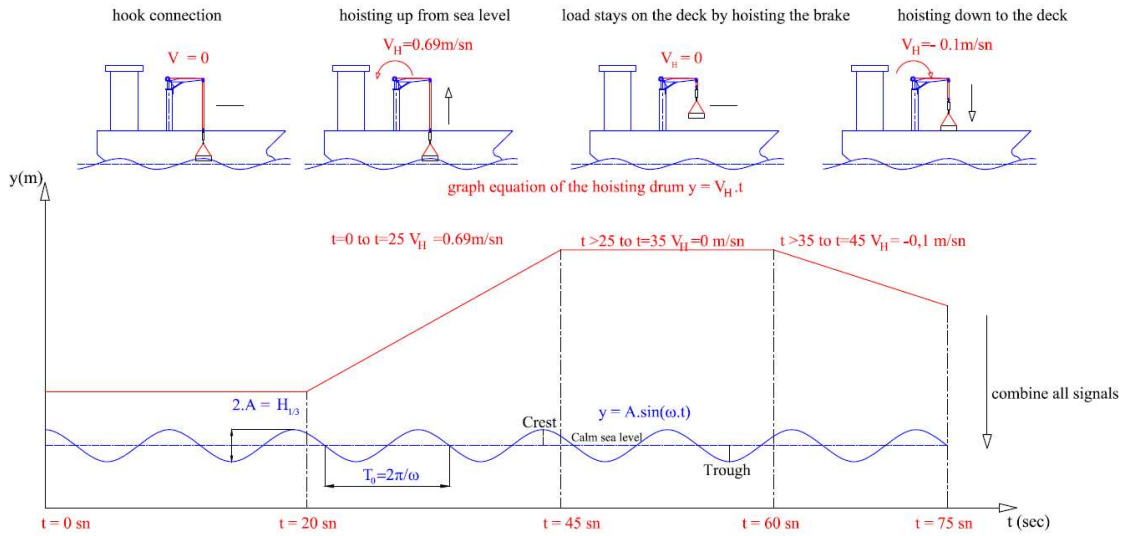


Figure 3.1: Segmented of the crane chart hoisting and significant wave height.

Table 3.1: Relation between significant wave height, wave period, crane hook and deck speed [9].

$H_{1/3}$ [m]	T_0 [s]	Min. V_h [m/min]	V_d [m/s]
0.5	3.1	13.2	0.34
1.0	4.1	19.8	0.61
1,5	4.8	25.2	0.86
2.0	5.4	30	1.09
2.5	5.9	34.2	1.31
3.0	6.3	37.8	1.53
4.0	7.1	45	1.95
6.0	8.3	57.6	2.73

The crane's lifting crane has time-varying movements. Considering that the ship deck is approximately 20 m above sea level, the movements of the ship lifting crane are created in 4 stages. A lift-lift diagram was drawn according to Table 3.1 for 20 to 45 seconds. The cargo, which was brought to the deck level of the ship between 45-60 seconds, was positioned at a fixed height by the shipowners for 15 seconds, and finally, a schedule was created to lower the cargo to the deck at low speed for 15 seconds. These 2 variable graphs were combined with the help of MATLAB to create a single graph. According to this time-dependent vertical motion graph, the input value of dynamic system models is provided.

This detailed road code, which varies over time, will be used in the Working Model 2 program. In our matrix-based MATLAB code, the loads will be applied in the form of a sinusoidal wave and the effect of simple and complex road models on the system will be evaluated.

3.1. Creating Lifting Operation Input Graphics

There are two main factors influencing the load reduction factor formula under rough sea conditions. Equations (3.1) and (3.2) are used to determine this load reduction. There is a relationship between the crane's lifting capacity in calm water SWL and its lifting capacity in rough seas $\overline{SWL}$, which involves the crane's spring constant, the hook speed, and the deck speed. For example, when the wave height is 3 meters, the wave period is 6.3 seconds, the vertical hook speed $V_{H/60}$ is 37.8 m/min, and the deck speed V_D is 1.53 m/s. According to the table in the TL Rules (Table 3.1), two input signals were created based on these values [9].

The relationship between the crane's lifting capacity in calm water (SWL) and its lifting capacity in rough seas $\overline{SWL}$ is described by the equation:

$$\overline{SWL} = SWL \frac{\psi}{\bar{\psi} \cdot f_e} \quad (3.1)$$

In this equation, f_e represents the influence factor, ψ is the hoist load coefficient, and $\bar{\psi}$ is the hoist load coefficient for working in a seaway. The hoist load coefficient for working in a seaway ($\bar{\psi}$) is further determined by the formula:

$$\bar{\psi} = 1 + (V_{H/60} + V_D) \sqrt{\frac{c}{g \cdot \overline{SWL}}} \quad (3.2)$$

Here, g is the gravitational constant (9.81 m/s^2), and c is the spring constant of the lifting appliance. These formulas take into account the dynamic loads imposed by rough sea conditions, adjusting the crane's lifting capacity accordingly

In **Figure 3.1**, the sinusoidal signal resulting from the wave effect is represented by the blue signal. The other input is related to the variables of the drum's operation over time as it lifts the load from the sea. This signal is indicated by the red line. Initially, the hook is attached to the load above the sea, and then it is lifted upwards at the V_h speed. When the load reaches the ship's deck level, there is a pause due to the intervention of the armament, and finally, the load is lowered onto the deck at a low speed. The figure above was prepared according to these time conditions mentioned before.

To generalize the signal obtained from the visuals above and to match the two signals, a MATLAB code was prepared. According to Turkish Lloyd rules time intervals are considered as:

$$t_1 = 0 \text{ to } 20 \text{ s} \quad (3.3)$$

$$t_2 = 20 \text{ to } 45 \text{ s} \quad (3.4)$$

$$t_3 = 45 \text{ to } 60 \text{ s} \quad (3.5)$$

$$t_4 = 60 \text{ to } 75 \text{ s} \quad (3.6)$$

Linear velocities for each interval are also given as:

$$V_{H1} = 0 \text{ m/s (0 to 20 s)} \quad (3.7)$$

$$V_{H2} = 0.69 \text{ m/s (20 to 45 s)} \quad (3.8)$$

$$V_{H3} = 0 \text{ m/s (45 to 60 s)} \quad (3.9)$$

$$V_{H4} = -0.1 \text{ m/s (60 to 75 s)} \quad (3.10)$$

Thus, the displacement functions can be written in the following form:

$$y_1 = A \sin(\omega t_1) + V_{H1} t_1 \quad (3.11)$$

$$y_2 = A \sin(\omega t_2) + (A \sin(\omega t_1(\text{end})) + V_{H1} t_1(\text{end})) + V_{H2}(t_2 - t_2(1)) \quad (3.12)$$

Where y_2 starts from the end value of y_1 and continuing with V_{H2} . Adjust to align with y_1 end value of y_2

$$y_2 = y_2 - y_2(1) + y_1(\text{end}) \quad (3.13)$$

For the third time interval,

$$y_3 = A \sin(\omega t_3) + (A \sin(\omega t_2(\text{end})) + V_{H2} t_2(\text{end}) - t_2(1))) + V_{H3}(t_3 - t_3(1)) \quad (3.14)$$

Similarly,

$$y_3 = y_3 - y_3(1) + y_2(\text{end}) \quad (3.15)$$

$$y_4 = A \sin(\omega t_4) + (A \sin(\omega t_3(\text{end})) + V_{H3} t_3(\text{end}) - t_3(1))) + V_{H4}(t_4 - t_4(1)) \quad (3.16)$$

Thus,

$$y_4 = y_4 - y_4(1) + y_3(\text{end}) \quad (3.17)$$

According to the formulation above, the input displacements obtained for wave heights of 2 meters and 3 meters are depicted in **Figures 3.2** and **Figure 3.3**.

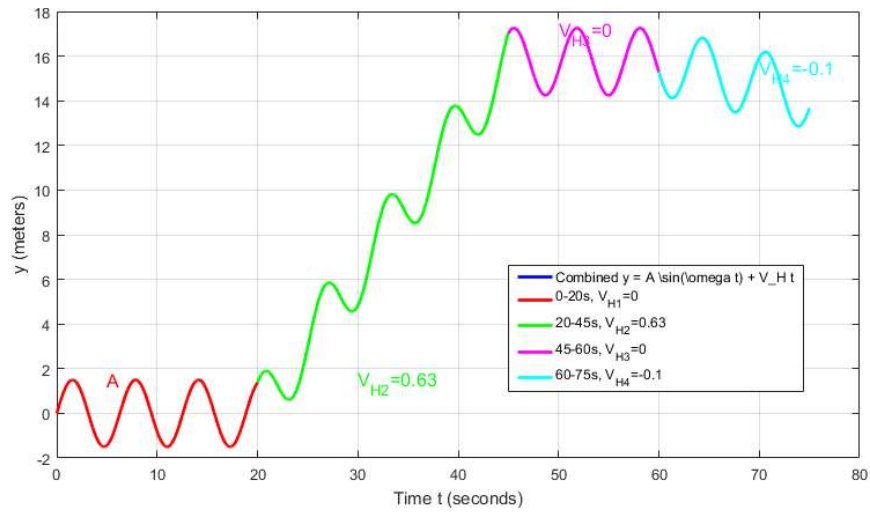


Figure 3.2: 3-meter wave height and lifting input signal.

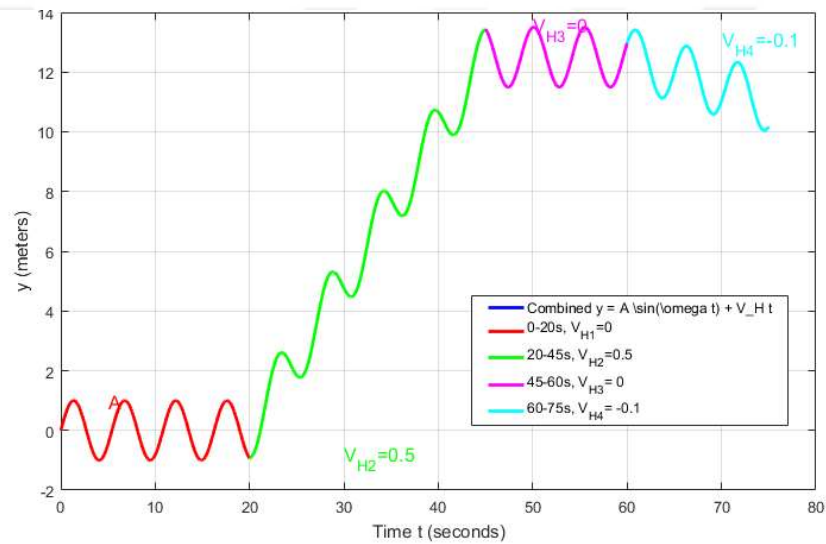


Figure 3.3: 2-meter wave height and lifting input signal.

Different simulations will be performed according to 2 different road inputs. The equations, setup, and visual of the detailed road graph that is used in the Working Model 2D program are below. There are sinusoidal and linear movements in the system. We do not add additional waves that will create a noise effect to the sinusoidal road data, we only add the critical wave effect.

3.1.1. Developing a Simplified Dynamic Model

To create the simulation model of the crane, we first examined the input signals that would affect the crane in the previous section. Now, we will identify the components

affecting the crane simulation and create the simplified dynamic model as shown in **Figure 3.4**.

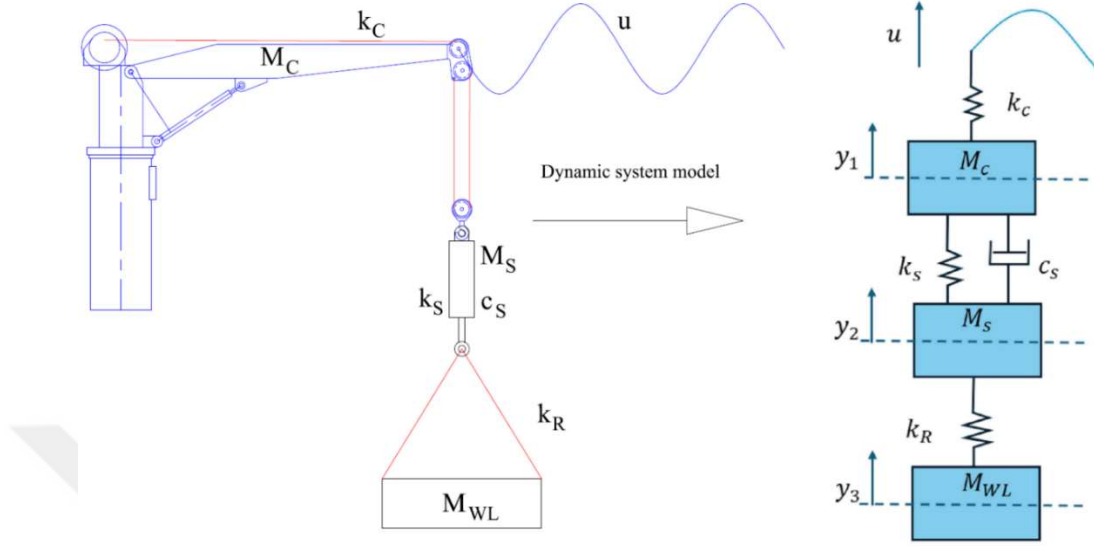


Figure 3.4: Dynamic system model of the crane.

As stated in the TL rules, the spring constant of the crane, hook speed, deck speed, and wave height are of significant importance when lifting loads in rough seas. By transforming the system into a simple mass-spring-damper system and considering the impact of the input signal, matrices can be obtained.

The spring constant of the crane and the ropes from the drum to the hook are denoted as k_C , the boom weight of the crane as M_C , and the mass, spring, and damping constants of the shock absorber as k_S , M_S , c_S , respectively. The spring constants of the suspension ropes are defined as k_R , and the weight of the load to be lifted is indicated as M_{WL} . For all 16 cranes considered in this work, the design of the elements within this system will be carried out in the next section, considering international standards and rules. Moreover, u [m/s] is the relative velocity between the load and hook at the time of pick-up (m/s), [9], [35].

The main spring coefficient of the crane is the combination of the spring coefficient of the crane's construction and the spring coefficients of the ropes that continue from the drum to the hook. This value is stated below as follows.

$$\frac{1}{k_C} = \frac{1}{k_B} + \frac{1}{k_{RC}} \quad (3.18)$$

3.1.2. Free Body Diagram of the System

The free body diagram of the system can be seen in **Figure 3.5**.

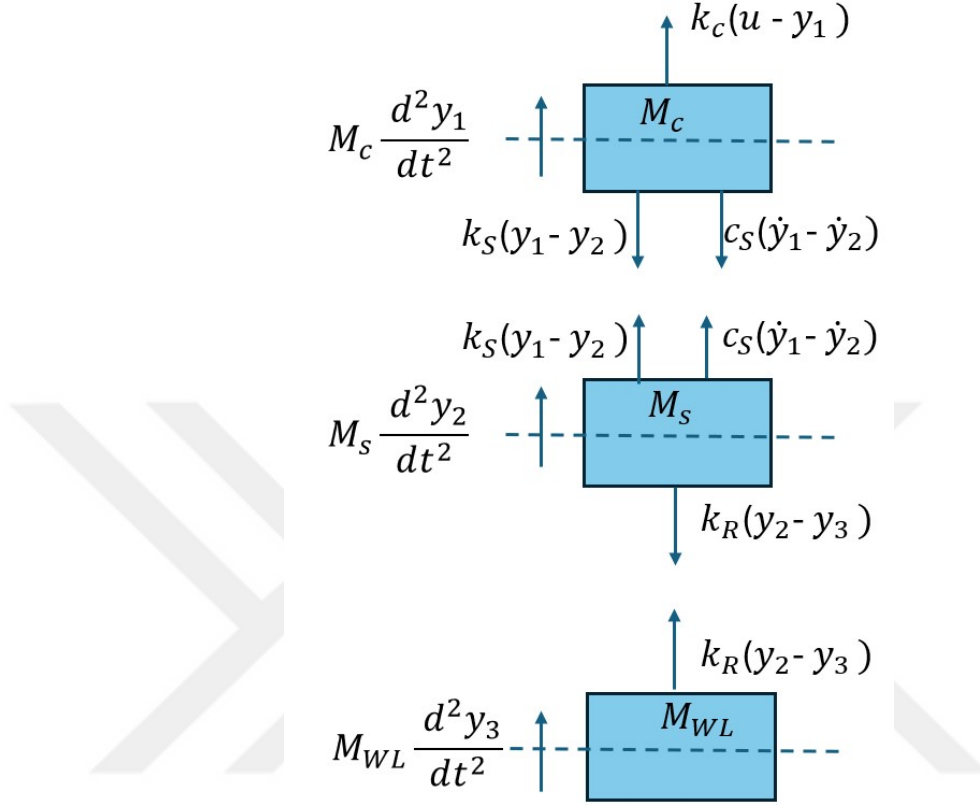


Figure 3.5: Free body diagram.

3.1.3. Equation of Motions

The equations of motion of the system are shown below. State space equations are used to describe the behavior of the dynamic system. They consist of two main equations: the State Equation and the Output Equation. The State Equation is written as $\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}$. This equation shows how the current state of the system, represented by the state vector $\mathbf{x}$, changes over time based on the system's state matrix $\mathbf{A}$ and the influence of the input vector $\mathbf{u}$ through the input matrix $\mathbf{B}$. The Output Equation is written as $\mathbf{y} = \mathbf{C}\mathbf{x} + \mathbf{D}\mathbf{u}$. This equation describes how the output of the system, represented by the output vector $\mathbf{y}$, is determined by the current state of the system through the output matrix $\mathbf{C}$ and the direct effect of the input vector $\mathbf{u}$ through the direct transmission matrix $\mathbf{D}$. In this regard, the dynamic mode can be obtained as:

$$M_{WL}\ddot{y}_3 = k_R(y_2 - y_3) \quad (3.19)$$

$$M_s\ddot{y}_2 = k_R(y_3 - y_2) - k_s(y_2 - y_1) + c_s(\dot{y}_1 - \dot{y}_2) \quad (3.20)$$

$$M_c\ddot{y}_1 = k_c(u - y_1) - k_s(y_1 - y_2) - c_s(\dot{y}_1 - \dot{y}_2) \quad (3.21)$$

and the state-space representation is

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{Ax} + \mathbf{Bu} \\ \mathbf{y} &= \mathbf{Cx} + \mathbf{Du} \end{aligned} \quad (3.22)$$

where state variables are:

$$x_1 = y_1 \quad (3.23)$$

$$x_2 = \dot{y}_1 \quad (3.24)$$

$$x_3 = y_2 \quad (3.25)$$

$$x_4 = \dot{y}_2 \quad (3.26)$$

$$x_5 = y_3 \quad (3.27)$$

$$x_6 = \dot{y}_3 \quad (3.28)$$

By converting to state-space equations, one obtains

$$\dot{x}_1 = x_2 \quad (3.29)$$

$$\dot{x}_2 = \frac{k_c}{M_c}(u - x_1) - \frac{k_s}{M_c}(x_1 - x_3) - \frac{c_s}{M_c}(x_2 - x_4) \quad (3.30)$$

$$\dot{x}_3 = x_4 \quad (3.31)$$

$$\dot{x}_4 = -\frac{k_s}{M_s}(x_3 - x_1) + \frac{c_s}{M_s}(x_2 - x_4) + \frac{k_R}{M_s}(x_5 - x_3) \quad (3.32)$$

$$\dot{x}_5 = x_6 \quad (3.33)$$

$$\dot{x}_6 = \frac{k_R}{M_{WL}}(x_3 - x_5) \quad (3.34)$$

Thus, state matrix of the model is:

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{(k_c+k_s)}{M_c} & -\frac{c_s}{M_c} & \frac{k_s}{M_c} & \frac{c_s}{M_c} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_s}{M_s} & \frac{c_s}{M_s} & -\frac{(k_s+k_R)}{M_s} & -\frac{c_s}{M_s} & \frac{k_R}{M_s} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \frac{k_R}{M_{WL}} & 0 & -\frac{k_R}{M_{WL}} & 0 \end{bmatrix} \quad (3.35)$$

Input matrix of the model is:

$$\mathbf{B} = \begin{bmatrix} \frac{k_c}{M_c} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad (3.36)$$

Output matrix of the model is:

$$\mathbf{C} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \end{bmatrix} \quad (3.37)$$

and finally, direct transmission matrix of the model is:

$$\mathbf{D} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \quad (3.38)$$

According to the matrices obtained, codes that can prepare graphics of the response of motion of the crane. Visuals in the form of animations have also been prepared in the MATLAB program. One can see the details of the codes in Appendix A [35].

3.1.4. Creating a Dynamic Model Integrated Into Working Model 2D

Working Model 2D program is a sub-production of MATLAB. It is a 2D visual-based design and evaluation program that enables kinematic system modeling to be done simply and quickly. This program will be used to prove the accuracy of the mathematical model and simulate movement according to detailed input data. The input data obtained from MATLAB were transferred to the AutoCAD program as time-dependent coordinates, converted to polyline format, and transferred to the Working Model 2 program. We have two separate road data according to wave height of 3 and 2 meters. The boundary conditions and input data simulation are shown in **Figure 3.6**.

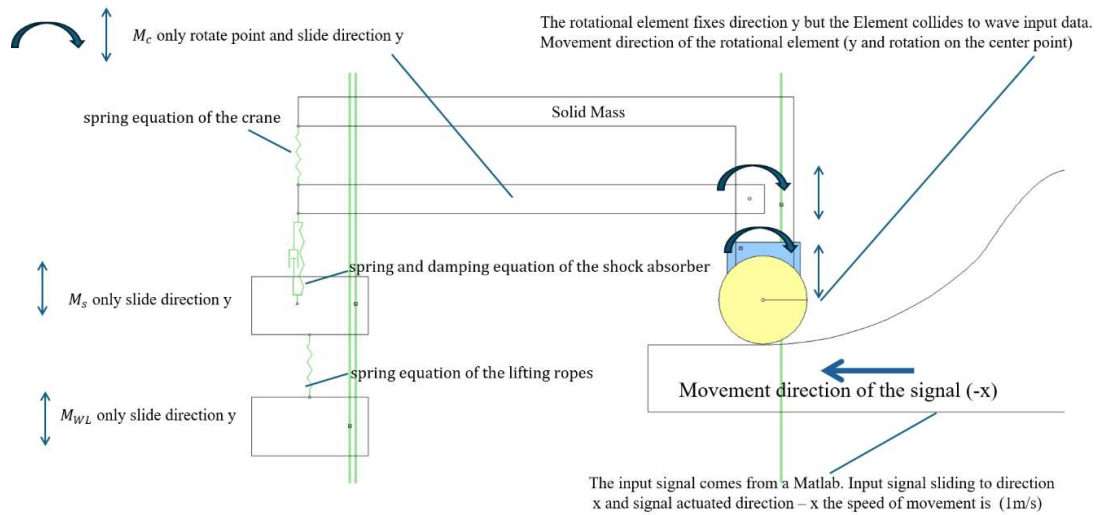


Figure 3.6 Working Model 2 D Inputs.

The combined view of the model and road data is shown in **Figure 3.7**.

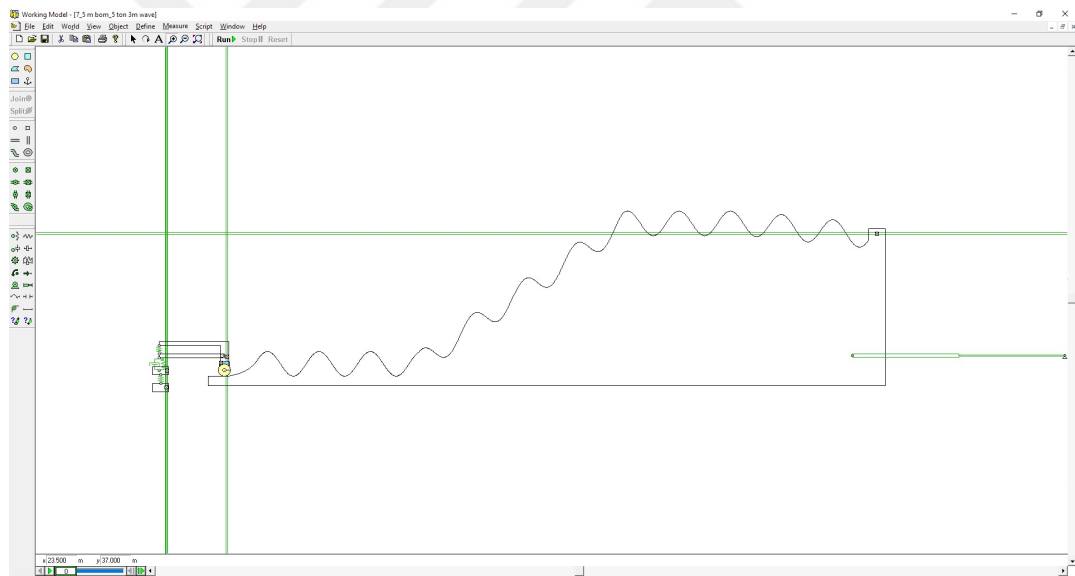


Figure 3.7 Working Model 2 D Inputs.

The properties of the components can be seen in **Figure 3.8**

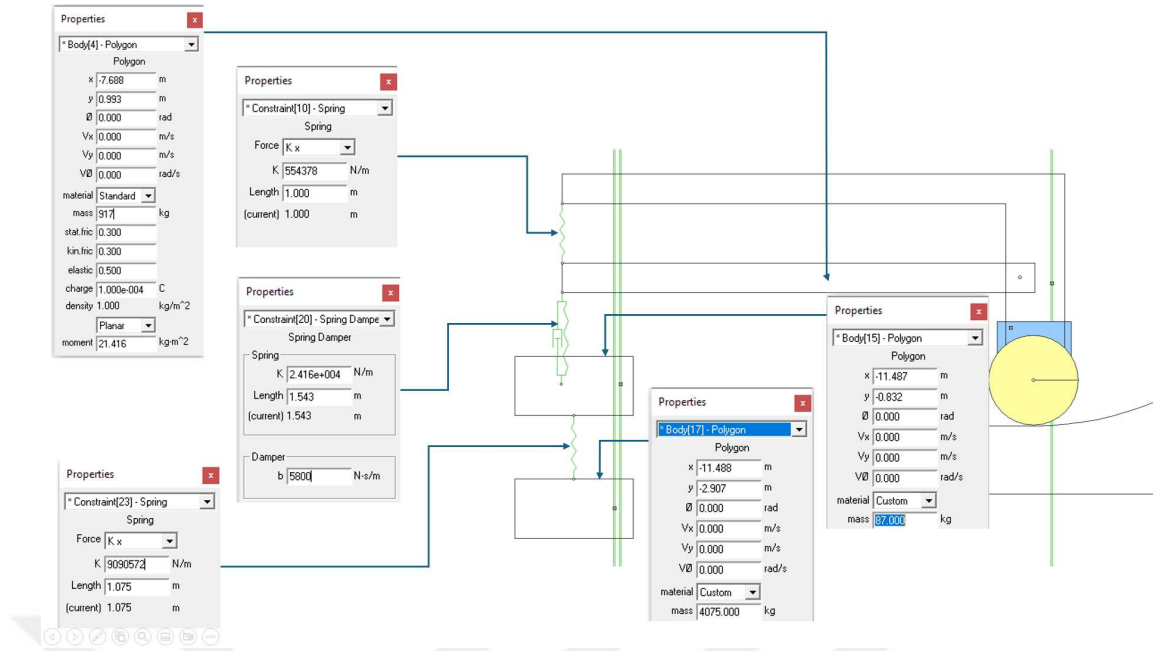


Figure3.8: Working Model 2D System Inputs.

3.2. Designing the Cranes and the Spring Coefficient k_c

The purpose of writing the thesis is to improve the article that TL can reduce the load if it wishes, in case a shock absorber is used, as stated in the TL Rules shown below. For this reason, cranes will be designed according to the requirements of TL Rules Part 50. The technical article of the rule is “If the lifting appliance is equipped with a shock absorber approved by the TL, a reduction of the SWL may be dispensed with totally or partially if the TL agrees.” [9].

In this study, we will design B-type cranes with a lifting capacity of 5 tons, 10 tons, 15 tons and 20 tons with 7.5m, 10m, 15m and 20m reach distances with hydraulic drives, which we frequently encounter, where we will find the spring coefficients. While making these designs, we will speed up the design processes by using parametric modeling. Two main factors cause the springiness of a crane. The first is the springiness of the crane caused by displacement, and the second is the effects caused by the extension of the ropes. Below we can see a typical deck crane and its components. Since the safe stress values on the crane are below the yield point, it will show linear behavior in the elongations against tension according to Hook's law. Thus, we can find the spring coefficient of the crane from the extension corresponding to a reference load.

The item k_c that we mentioned in the system model above is the spring coefficient in the z-axis of the crane if the crane is designed in accordance with Turk Lloyd rules. Since the crane operates below its yield strength under normal operating conditions, the stresses occurring on the steel body are in the linear region of the hook curve, so finding k_c with a linear calculation will be sufficient. Although k_c seems to be a simple coefficient, in fact, in order to find the k_c value, it is necessary to calculate all parts of the entire crane that have a critical effect on its spring.

3.2.1. Design of Crane Components

When designing a crane, there are components that we can and cannot change in the design. Since turning gears, pulleys, hydraulic cylinders, and drums are expensive products of cranes, they are generally selected according to close limit values according to the maximum moment and axial loads. These choices play an important role in the formation of the general dimensions of the pedestal and mass. The diameter of the rope is determined according to the safe load to be lifted, and the drum diameter and pulley diameters are determined according to the diameter of the rope. The stresses occurring in steel bodies are also determined according to the maximum stress values determined in the rule. In addition, we will standardize the basic dimensions of the pedestal mass and hydraulic cylinder for ease of calculation [9], [33].

Finding k_c means designing the crane components in such a way that they do not exceed the safety values allowed by Turk Lloyd rules and are not excessively over-engineered. **Figure 3.9** shows which part of the crane is designed according to which rule. There are many separate details in crane design, but here we focus only on the parts that have a critical impact on finding k_c .

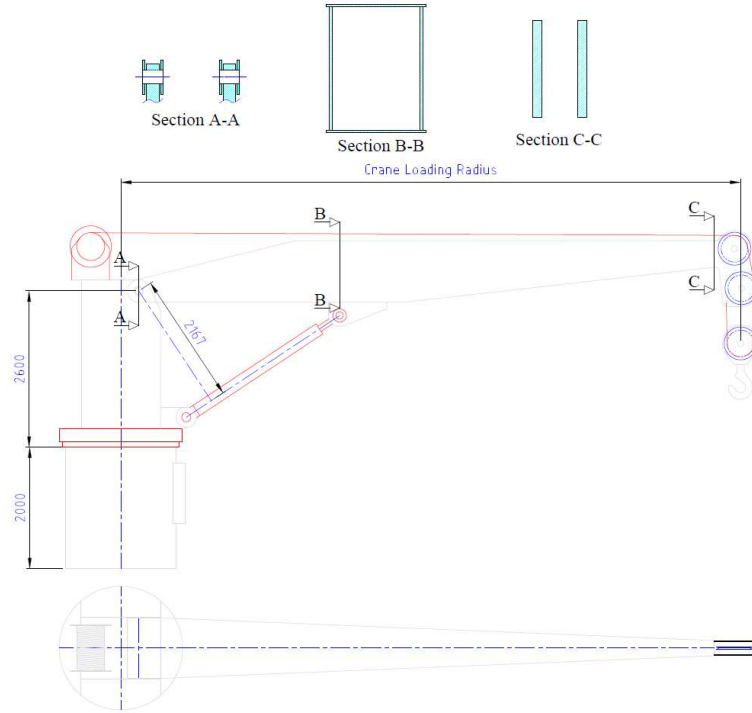


Figure 3.9: General design of the crane to find k_c value.

Slewing ring design rules: TL Rules Chapter 50 Section 4 Table 4.3 occurred maximum bending moment and axial load. Hydraulic cylinder: TL Rules Chapter 50 Section 4 Table 4.3 and DNV-ST-0194. Rope size: TL Rules Chapter 50 Section 8 Table 8.1. Sheave block and drum diameter TL Rules Chapter 50 Section 8 Table 8.2. Safety factor of the structural components TL Rules Chapter 50 Section 2 Table 2.8 and Section 2/E 6.1. Design of the structural elements TL Rules Chapter 50 Sections 2, 3, and 4.

3.2.1.1. Calculation of Acceptable Safe Stresses in Crane Design

We will determine the allowable stresses according to equation (3.39). TL-A 36 quality steel will be preferred as the material of the Boom and Tower. The yield strength of this steel is 355N/mm². It is a certified steel that is used in ships, is ductile, has high weldability, and has good performance in chap notch values.

$$\sigma_{\text{per}} = \frac{0.8 R_{eH} + 4.8}{v} [\text{kN/cm}^2] \quad (3.39)$$

Where R_{eH} is the material's yield strength, and v is the safety factor. This equation shows how much safety coefficient should be applied to which element is used in the

crane. Since the stresses in cranes are generally compound, we will find the von Misses stresses where we find the stress and realize the designs in this way

According to TL rules, with safety factor $n = 1.4$ for combined stress load condition, permissible stress for structural members and welds is:

$$\sigma_{\text{per}} = \frac{0.8 (35.5) + 4.8}{1.4} = 237 \text{N/mm}^2 (\text{Load Condition A})$$

According to TL rules, with safety factor $v = 1.6$ for combined stress load condition if resistance to buckling is required, permissible stress is

$$\sigma_{\text{per}} = \frac{0.8 (35.5) + 4.8}{1.6} = 208 \text{N/mm}^2 (\text{Load Condition A})$$

3.2.1.2. Calculation of the Ship's Static Inclination and Its Effects

In our system setup, the movements are on the z-axis. In TL rule and European Standards, the system is built on the z-axis, and static and dynamic effects are caused by the fore-aft tilt and trim factors of the ship. Movements are included by adding empirical formulas found through experience. The static heel value for ships is 5° and the trim value is 2° . For the sake of convenience when making calculations, we will consider more adverse conditions by thinking as if the ship was tilting 5° in both directions.

We will evaluate our sample models according to the ship, which is the floating vehicle with the highest hell and trim effect. Below is the table of static movements occurring in ships. If the ship operates in high-wave seas. In this case, there is an increase in the angles caused by the sea effect [9].

Table 3.2: Static Inclination of the Ship [9].

Type of floating body	Minimum static inclination Heel	Minimum static inclination Trim
Ships and similar floating bodies	$\pm 5^\circ$	$\pm 2^\circ$
Pontoons	$\pm 3^\circ$	$\pm 2^\circ$
Floating docks	$\pm 2^\circ$	$\pm 2^\circ$
Floating cranes up to 100 t SWL	$\pm 5^\circ$	$\pm 2^\circ$
Floating cranes over 100 t SWL	$\pm 3^\circ$	$\pm 2^\circ$
Semi-submersibles	$\pm 3^\circ$	$\pm 3^\circ$
Fixed platforms	$\pm 1^\circ$	$\pm 1^\circ$

3.2.1.3. Calculation Of The Diameter Of The Crane Ropes

Steel rope with a breaking strength of 1770N/mm² will be preferred in the design of all our cranes. In the selection of ropes, load changes affecting wavy situations $\psi \leq 1.6$ are not considered, but the friction occurring in the pulley blocks will be included in the calculations. Considering that all pulleys are designed with bearings, the pulley friction coefficient is 2% ($K = 1.02$) efficiency loss is included.

Table 3.3: Coefficient of utilization for wire ropes [9].

K ₁ Coefficient of utilization for wire ropes for running rigging	
Safe working load of lifting appliances or loose gear	
Up to 10 t	5
10 t - 160 t	$10000 / (8.85 \times \text{SWL} + 1910)$

Every time the rope passes through the pulleys, there is a decrease in its performance. The bearing elements used (bearing or bronze bearings, etc.) have a serious impact on this performance. This value must be calculated meticulously, especially in the large A-frame and gantry cranes with very high lifting capacity. The relevant formula is shown in equation (3.40). Formula of the hoisting performance reduction from sheave block is

$$\eta = \frac{Fr^{nos(P-1)}}{Fr^{nos}-1} \quad (3.40)$$

where $Fr = 1.02$ or 1.05 respectively. In this regard, different crane sheaves in the tackles are shown in **Figure 3.10**.

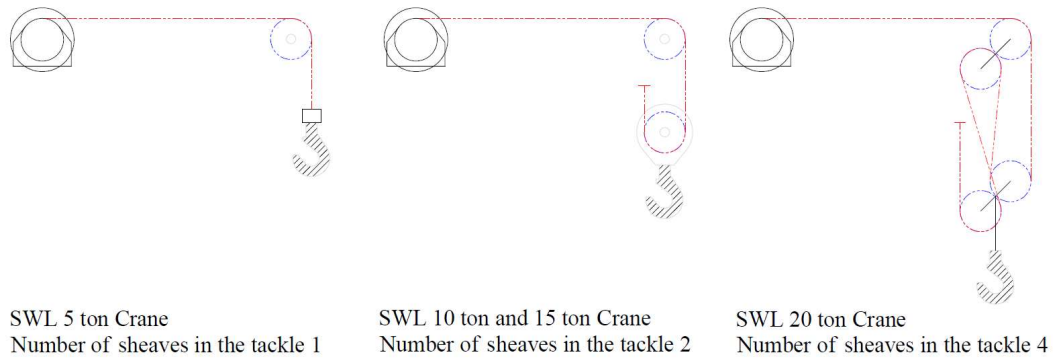


Figure 3.10: Cranes sheaves in the tackle

The reduction values for these sheaves can be calculated as follows

$$\eta_{swl5} = 1.02 \quad \eta_{swl10} = 0.516 \quad \eta_{swl15} = 0.516 \quad \eta_{swl20} = 0.263$$

Thus: Minimum breaking load of SWL5 ton crane rope = 5 tons $\times$ 1.02 = 5.1 tons. SWL5 ton crane rope minimum breaking load $5.1 \times 5 = 25.5$ tons = 250.155kN. TL Rules Chapter 50 Appendix A Round rope Warrington-steel wire rope and winding layout is 6x36 (1770N/mm² breaking rope Ø22 mm breaking load 283kN > 250.155kN. The required sheave block diameter according to this rope diameter is $22 \times 20d = \text{Ø}440$ mm. Minimum breaking load of SWL10 tons crane rope = 10 tons $\times$ 0.516 = 5.16 tons. SWL10 ton crane rope minimum breaking load $(10000 / (8.85 \times 10 + 1910)) \times 5.16 \text{ ton} = 25.81 \text{ ton} = 250.155\text{kN}$. TL Rules Chapter 50 Appendix A Round rope Warrington-steel wire rope and winding layout is 6x36 (1770N/mm² breaking rope Ø22 mm breaking load 283kN > 250.155 kN. The required sheave block diameter according to this rope diameter is $22 \times 20d = \text{Ø}440$ mm. Minimum breaking load of SWL15 ton crane rope = 15 tons $\times$ 0.516 = 7.74 tons SWL15 ton crane rope minimum breaking load $(10000 / (8.85 \times 15 + 1910)) \times 5.16 \text{ ton} = 37.889 \text{ ton} = 371.697\text{kN}$. TL Rules Chapter 50 Appendix A Round rope Warrington-steel wire rope and winding layout is 6x36 (1770N/mm² breaking rope Ø26 mm breaking load 283kN > 250.155 kN. The required sheave block diameter according to this rope diameter is $26 \times 20d = \text{Ø}520$ mm. Minimum breaking load of SWL20 tons crane rope = 20 tons $\times$ 0.263 = 5.26 tons. SWL10 ton crane rope minimum breaking load $(10000 / (8.85 \times 20 + 1910)) \times 5.16 \text{ ton} = 25.2 \text{ ton} = 247.212 \text{ kN}$. TL Rules Chapter 50 Appendix A Round rope Warrington-steel wire rope and winding layout is 6x36 (1770N/mm² breaking rope Ø22 mm breaking load 283kN > 247.212 kN. The required sheave block diameter according to this rope diameter is $22 \times 20d = \text{Ø}440$ mm. The diameter of the pulley to be selected according to the rope diameter is determined based on **Table 1.1**.

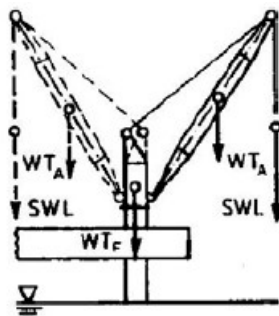
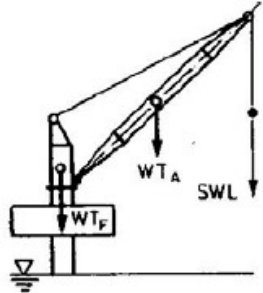
3.2.1.4. Simple Calculation of Crane Structural Parts

The main purpose of simply calculating the crane's boom is to provide the spring coefficient, and approximate weight of the crane's boom and to carry out the preliminary design of the crane with these values and then carry out preliminary work for the 3D model to be prepared. Additionally, by using these calculations, the

diameter of the turning gear and the tower will be determined. Our cranes will be calculated according to the load cases in TL rules.

We expect all the cranes we will design to operate offshore at 3-meter wave heights. The expected minimum hook speed at a wave height of 3 meters is specified in the standard as 37.8m/min [9]. In this regard, important loading parameters are shown in **Table 3.4**.

Table 3.4: Loading Parameters [9].

In Operation			
			
		No swell	Slight swell
Crane jib position (working range)			
Load Case	Possible Loads	Any	Over the water
		Load assumption	Load assumption
B A	Dead load	$WT_F \times WT_A \varphi$	$WT_F \times WT_A \varphi$
	Hoisting load	$SWL \times \psi$	$SWL \times \bar{\psi} \times f_e$
	Horizontal force from slewing	To be calculated if need be	To be calculated if need be
	Deck inclination	Static minimum inclination	Static minimum inclination
	Cargo fall angle of attack	Corresponding to total inclination	Corresponding to total inclination
	Swell	To be calculated or influence factor	To be calculated or influence factor
	Wind load	Lift. Appliances load $V=25\text{m/s}$	Lift. Appliances load $V=25\text{m/s}$
	Ice load	If to be considered	If to be considered
SWL available		SWL	$\overline{SWL} = \frac{SWL}{f_e} \times \frac{\psi}{\bar{\psi}}$

Crane dead load design coefficients are shown in **Table 3.5** and hoisting coefficients of the crane are shown in **Table 3.6**

Table 3.5: Crane dead load design coefficients [9].

Type and SWL of lifting appliances	Dead load coefficient ϕ
Derrick booms up to 10 t <i>SWL</i>	1.20
All other lifting appliances and derrick booms over 10 t <i>SWL</i>	1.10
Up to 60 t <i>SWL</i>	1.10
Over 60 t up to 100 t <i>SWL</i>	1.05
Over 100 t <i>SWL</i>	1.00

Table 3.6: Hoisting coefficients [9].

Type of crane	Hoist load coefficients ψ in dependence on hoisting speed V_h	
	$V_h \leq 90$ [m/min]	$V_h \geq 90$ [m/min]
Shipboard cranes type A	$\psi = 1.1 + 0.0022 \times V_h \geq 1.15$	$\psi = 1.3$
Shipboard cranes type B	$\psi = 1.2 + 0.0044 \times V_h \geq 1.3$	$\psi = 1.6$

Thus, for the case in this work, one can calculate the hoisting coefficient as:

$$\psi = 1.2 + 0.0044 \times 37.8 = 1.3663 \text{ m/min}$$

Moreover, the influence factor in calculating hoisting coefficient according to **Table 3.7**, is defined in the following table.

Table 3.7: Influence Factor [9].

Offshore Installation	f_e
Fixed platform	1.00
Semi-submersible	1.05
Ship	1.10
Pontoon	1.15

In this regard, 2D preliminary designs were made to design 16 different cranes with a lifting capacity of 5 tons to 20 tons at a frequently encountered reach distance of 7.5m

to 20m and to define the basic dimensions. The relevant preliminary designs are shown in **Figures 3.11 to 3.13**.

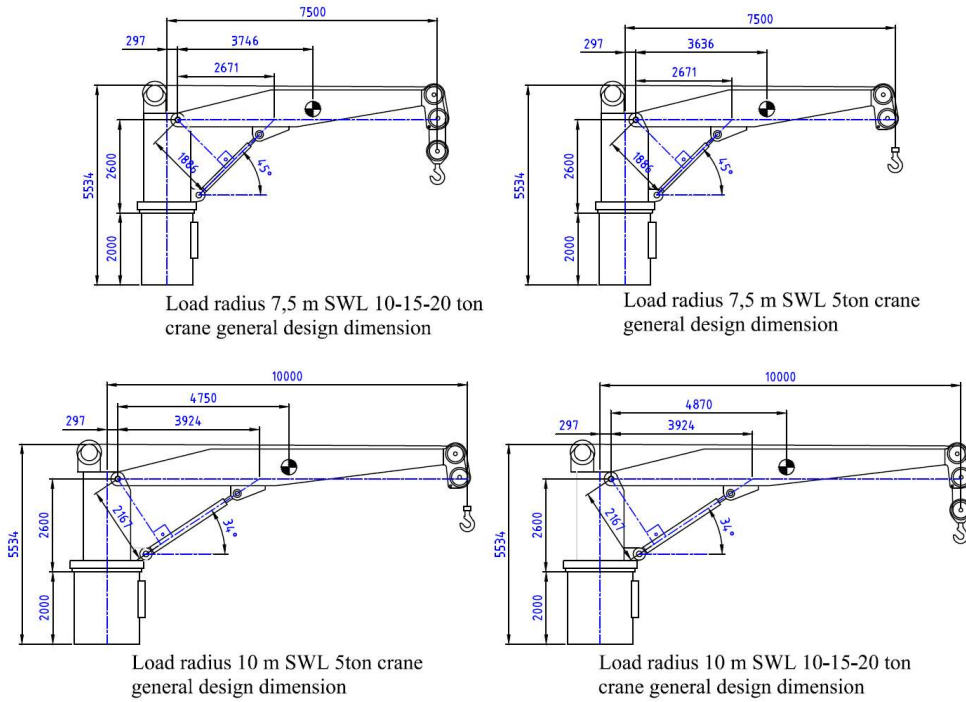


Figure 3.11: Predesign of the cranes 7.5 m and 10m radius.

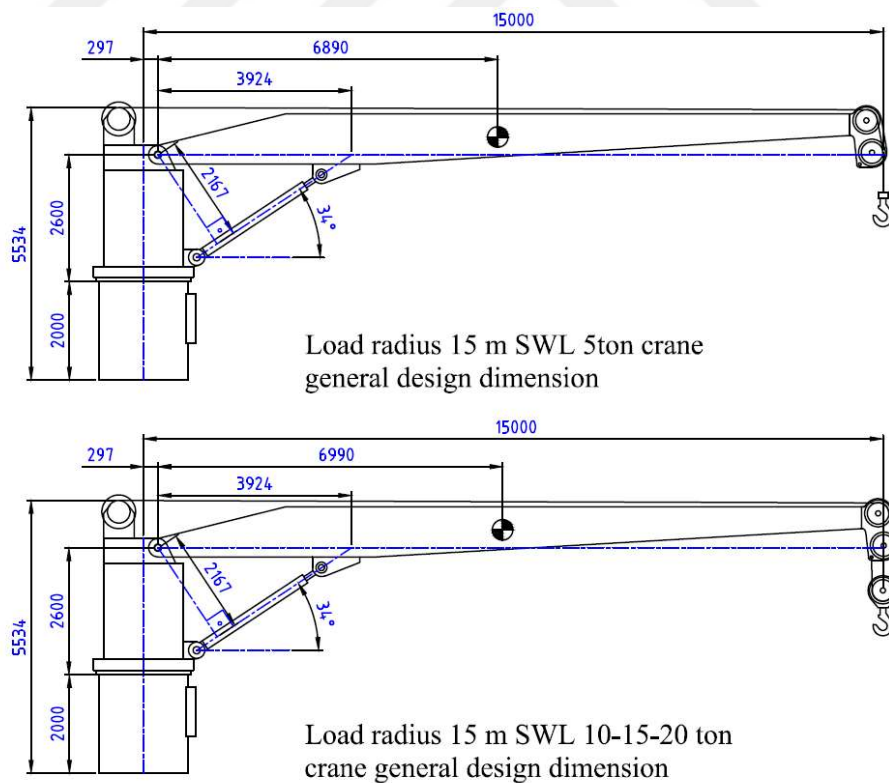


Figure 3.12: Predesign of the cranes 15m radius.

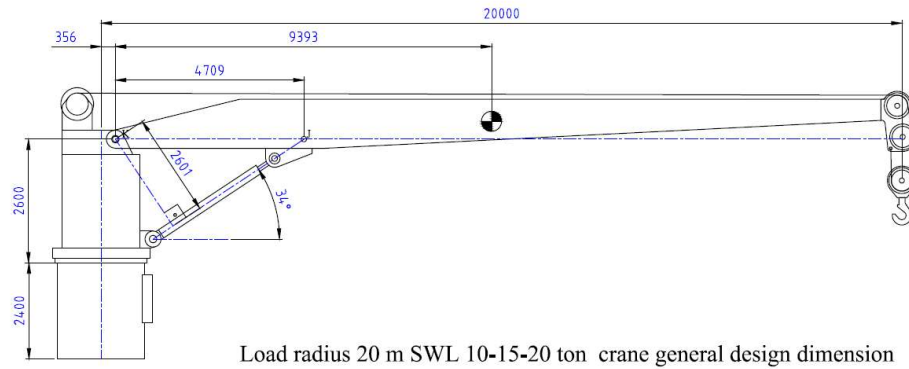
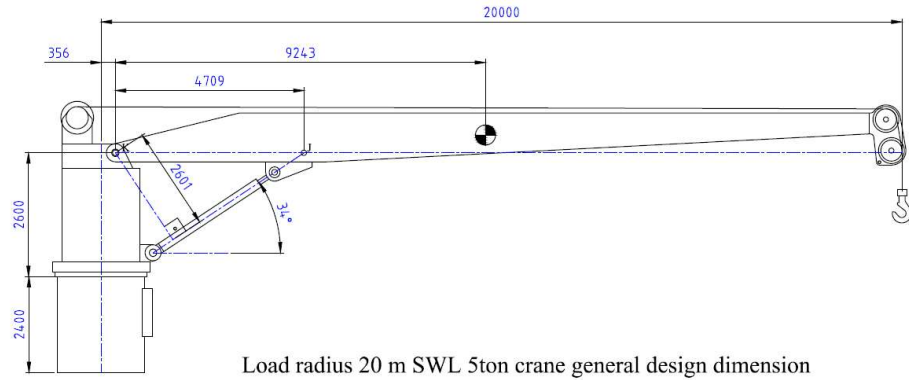


Figure 3.13: Predesign of the cranes 20m radius.

3.2.2. Processing the Loads Acting on the Crane on the Beam Element and Calculating the Main Loads with Simple Strength Equations

First, we will evaluate our crane as a rod element and define the loads on it. Since we have calculated the boom element of our crane, we will model the wind load and the effect of the heel and trim in a way that will create more tension on the boom. In this model, we will find the stresses obtained from all loads and then ensure that the von Misses stress remains below the safety limit. In this regard, the isometric images below show the design of the cranes as beam elements. Since a general formula is wanted to be obtained, critical measurements are indicated with letters. In the left image in **Figure 3.14**, the transformation of the general dimensions of the crane into the beam element can be seen. In the following image, the center of gravity of the boom is seen. In the next image, the pressure acting on the boom as a distributed load on the y-axis is seen. In the next image, the stresses on the boom due to wind load are seen. In the next image, the stresses are caused by the weight of the boom, and in the rightmost image, the stresses are caused by the boom lifting the load.

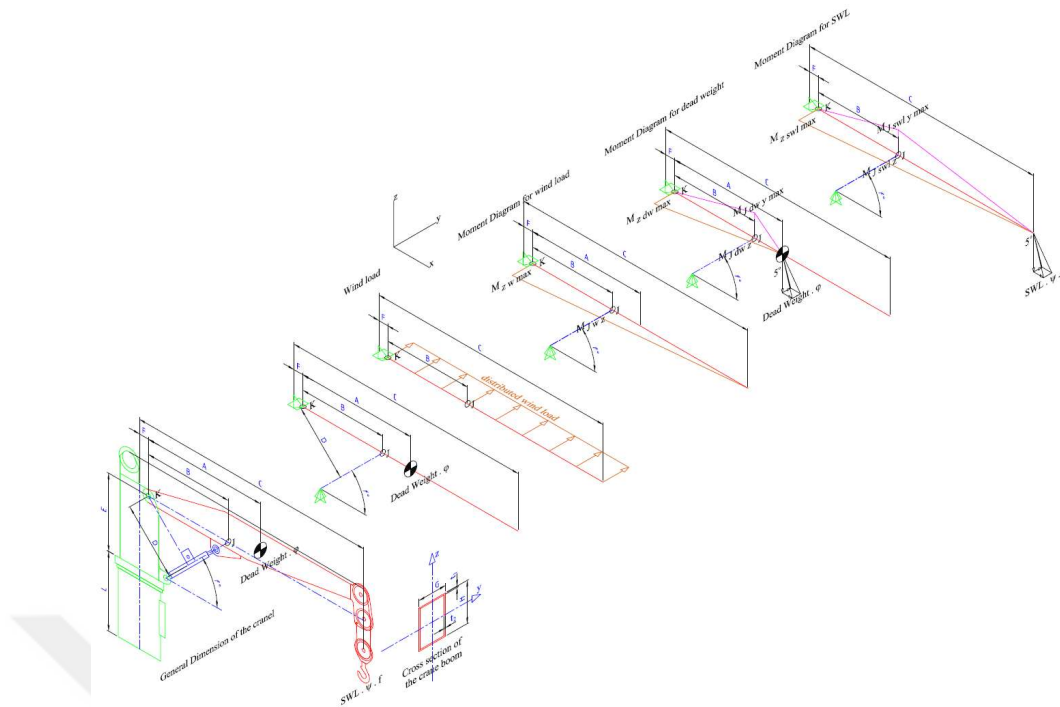


Figure 3.14: Converting the crane to a 2D simple model Isometric.

Also, in **Figure 3.15**, the 2d simple models' side view of the crane.

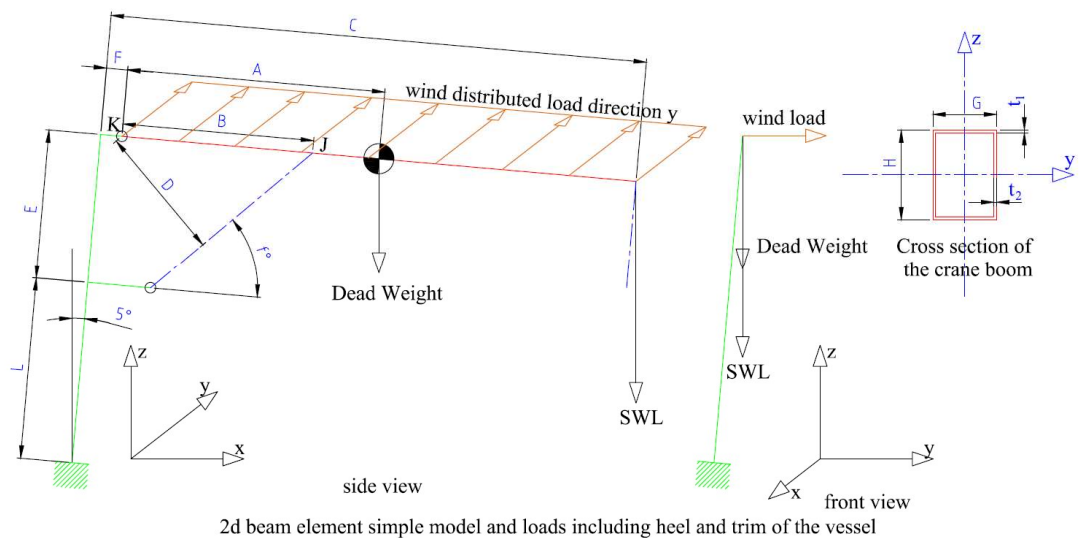


Figure 3.15: Converting the crane to a 2D simple model Side view.

Wind forces acting on cranes are specified in detail by F.E.M (European Federation of Material Handling). Since crane booms are generally box-shaped, the coefficient c was chosen as 1.6 and the wind speed was chosen as 25 m/s [9], [19].

The wind load on a structural member of a lifting appliance shall be deemed to act in the most unfavorable direction, and is to be calculated by the following formula:

Below is the wind force calculation formula from FEM standards. Wind force increases with the square of the speed.

$$F_w = \bar{c} q A_{\text{side}} \text{ (kN)} \quad (3.41)$$

$$A_{\text{side}} = H(C - F) \quad (3.42)$$

$$\bar{c} = 1.6 \text{ (constant square tube element)}$$

$$V = 25 \text{ m/s (wind speed according to T Rules)}$$

$$q = V^2 / 1600 \text{ (kN/m}^2\text{)} \quad (3.43)$$

$$\text{beam distributed load} = q \bar{c} H \quad (3.44)$$

$$\text{beam distributed load} = 0.625 H \quad (3.45)$$

$$M_{z \text{ w max}} = \frac{\text{beam distributed load} \times (C - F)^2}{2} \quad (3.46)$$

$$M_{J \text{ w z}} = \frac{\text{beam distributed load} \times ((C - F) - B)^2}{2} \quad (3.47)$$

To define the wind force, there must be areas that contact the wind surface perpendicularly. This area is related to the boom length of the crane and the height of the boom section.

$$F_{J \text{ wind}} = ((C - F) - B) 0.625 H \quad (3.48)$$

To find the stress of the wind on the boom, we need to find the moments of inertia on the weak axis of the crane boom and find the shear and bending stresses caused by the wind force. We can see the relevant formulas below.

$$W_z = \frac{H \cdot G^3 - (H - 2 t_1) (G - 2 t_2)^3}{6 G} \quad (3.49)$$

$$\sigma_{z \text{ wind max}} = \frac{M_{z \text{ w max}}}{W_z} \quad (3.50)$$

$$\sigma_{J \text{ z wind}} = \frac{M_{J \text{ w z}}}{W_z} \quad (3.51)$$

$$\tau_{J \text{ wind}} = \frac{F_{J \text{ wind}}}{2 \cdot G \cdot t_1} \quad (3.52)$$

The image below shows the general model of the crane boom created with the beam element. This image shows the distributed wind load acting on the crane boom and the stresses occurring on the crane boom due to this wind load. A , B , C , and F dimensions are the critical dimensions of the crane boom. $M_{z \text{ w max}}$ is the maximum stress caused

by the wind. Crane beam model, Isometric view wind loads and stresses are shown in **Figure 3.16**

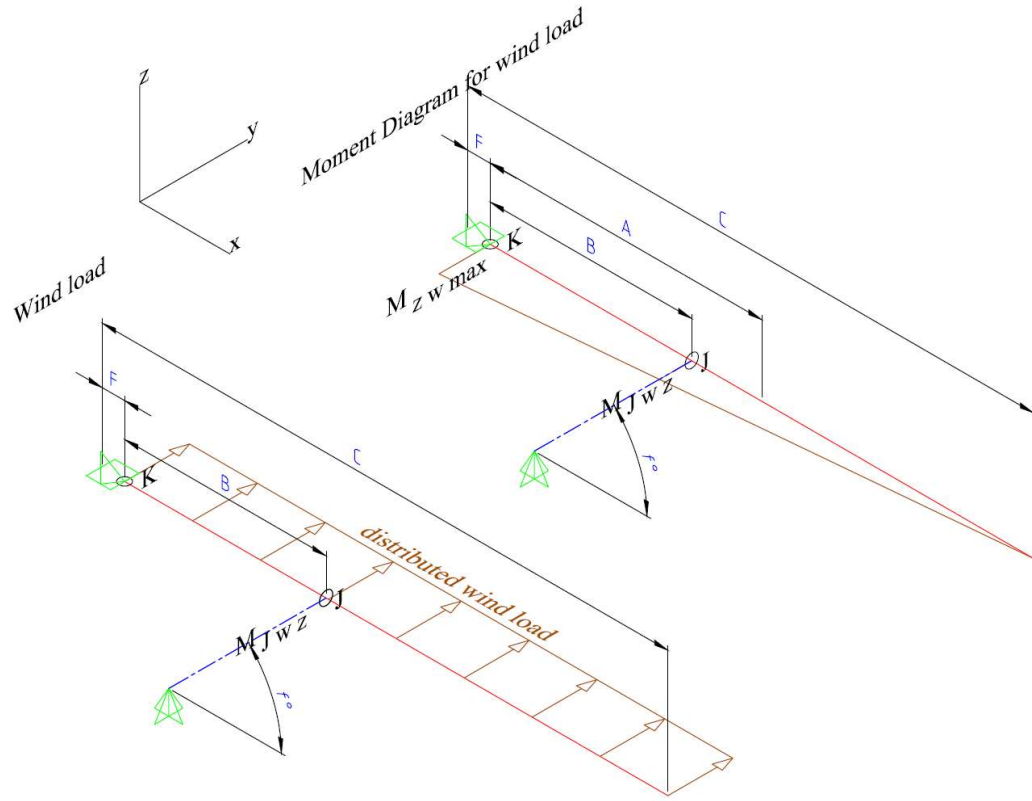


Figure 3.16: Crane beam model, Isometric view wind loads and stresses.

Approximate dead weight of the crane boom (include dead load coefficient):

$$G_{dw} = \rho_{steel}(GH - (G - 2t_2)(H - 2t_1))(C - F)\varphi \quad (3.53)$$

$$\varphi = 1.1 \text{ or } 1.2 \text{ according to table 2.2} \quad (3.54)$$

$$\rho_{steel} = 7.85 \text{ g/cm}^3$$

To find the crane boom weight, it is necessary to relate the boom cross-sectional area to the boom length. Since we found the approximate boom weight in the calculations, it was thought that it affected the heel and trim effect caused by ship motion by 5° , but there was no decrease in the dead weight. For this reason, additional force effects arise in the y and x directions [9].

The effect of these additional forces:

$$G_{dw y} = G_{dw} \sin(5^\circ) \varphi \quad G_{dw x} = G_{dw} \sin(5^\circ) \varphi \quad (3.55)$$

$$M_{J \text{ dw y max}} = (A - B) G_{dw} \quad (3.56)$$

$$W_y = \frac{G H^3 - (G - 2 t_2) (H - 2 t_1)^3}{6 H} \quad (3.57)$$

$$\sigma_{J \text{ dw y max}} = \frac{M_{J \text{ dw y max}}}{W_y} \quad (3.58)$$

$$M_{z \text{ dw max}} = A G_{dw y} \quad (3.59)$$

$$M_{J \text{ dw z}} = (A - B) G_{dw y} \quad (3.60)$$

$$\sigma_{J \text{ dw z}} = \frac{M_{J \text{ dw z}}}{W_z} \quad (3.61)$$

$$A_{\text{crane}} = (G H - (G - 2 t_2) (H - 2 t_1)) \quad (3.62)$$

$$\sigma_{x \text{ dw}} = \frac{G_{dw x}}{A_{\text{crane}}} \quad (3.63)$$

$$\tau_{J y \text{ dw}} = \frac{G_{dw y}}{2 G t_1} \quad (3.64)$$

$$\tau_{J z \text{ dw}} = \frac{G_{dw}}{2 H t_2} \quad (3.65)$$

The picture below shows the general model of the crane arm created with the beam element. This image shows the stress effects of the crane's weight on the crane boom. Dimensions A , B , C , and F are the critical dimensions of the crane arm. $M_{J \text{ dw z}}$ and $M_{z \text{ dw max}}$ are the maximum stresses caused by the weight of the boom. Crane beam model, Isometric view dead weight loads and stresses are shown in **Figure 3.17**

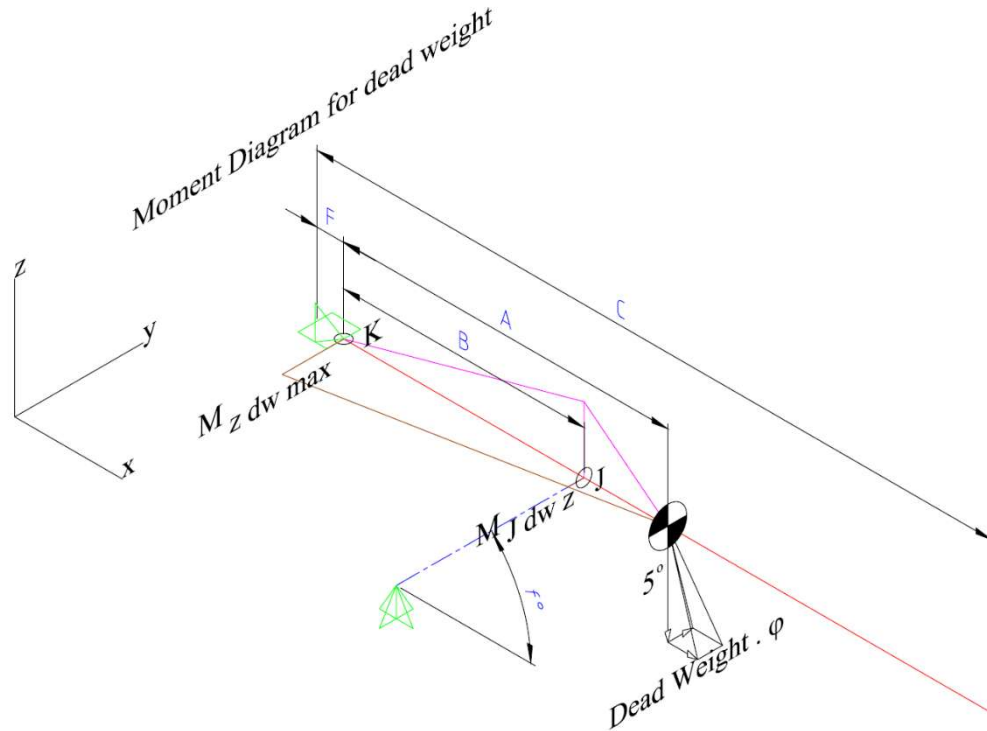


Figure 3.17: Crane beam model, Isometric view dead weight loads and stresses.

It is thought that the heel and trim effect caused by the movement of the ship has a 5° effect, but there is no decrease in SWL . For this reason, additional force effects arise in the y and x directions [9].

The effects of these additional forces are:

$$F_{swl\ y} = SWL \sin(5^\circ) \psi f_e \quad F_{swl\ x} = SWL \sin(5^\circ) \psi f_e \quad (3.66)$$

$$F_{swl\ z} = SWL \psi f_e \quad (3.67)$$

$$M_{J\ swl\ y\ max} = ((C - F) - B) SWL \psi f_e \quad (3.68)$$

$$\sigma_{J\ swl\ y\ max} = \frac{M_{J\ swl\ y\ max}}{W_y} \quad (3.69)$$

$$M_{z\ swl\ max} = (C - F) F_{swl\ y} \quad (3.70)$$

$$M_{J\ swl\ z} = (C - F - B) F_{swl\ y} \quad (3.71)$$

$$\sigma_{J\ swl\ z} = \frac{M_{J\ swl\ z}}{W_z} \quad (3.72)$$

$$A_{crane} = (GH - (G - 2t_2)(H - 2t_1)) \quad (3.73)$$

$$\sigma_{x\ swl} = \frac{F_{swl\ x}}{A_{crane}} \quad (3.74)$$

$$\tau_{J\ y\ swl} = \frac{F_{swl\ y}}{2.G.t_1} \quad (3.75)$$

$$\tau_{J\ z\ swl} = \frac{F_{swl\ z}}{2.H.t_2} \quad (3.76)$$

The picture below shows the general model of the crane arm created with the beam element. This image shows the stress effects of the crane's working load on the crane boom. Dimensions A , B , C , and F are the critical dimensions of the crane arm. $M_{J\ swl\ z}$ and $M_{z\ swl\ max}$ are the maximum stresses caused by the working load. Crane beam model, Isometric view SWL loads and stresses are shown in **Figure 3.18**

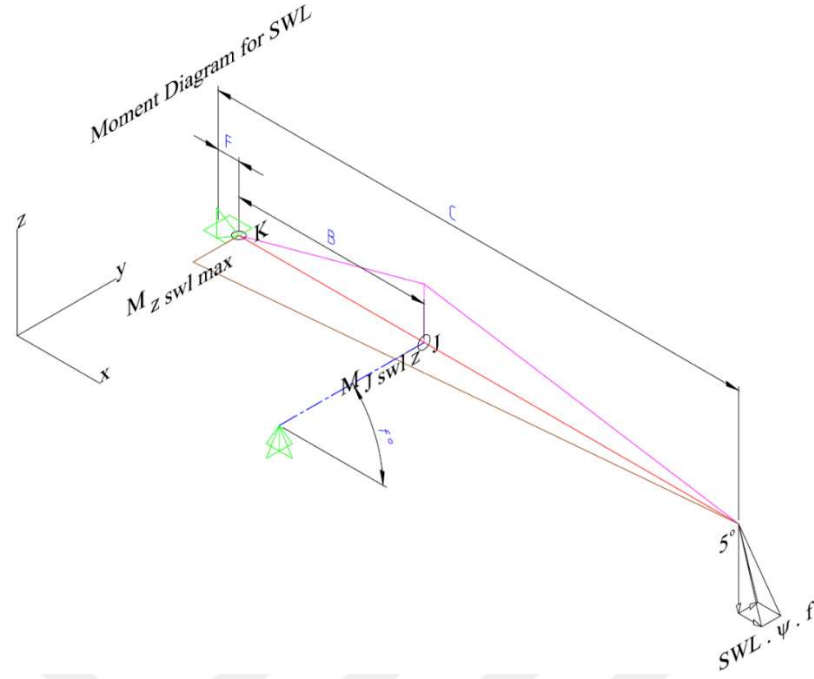


Figure 3.18: Crane beam model, Isometric view SWL loads and stresses.

The place where the critical stress on the boom occurs is at the location of the connection of the hydraulic cylinder. For this reason, the stresses at this location have been determined in the formulas above. The resultant von Misses stresses of the effects of these three different load groups, wind, dead weight, and *SWL*, have been found. This stress must be below the safe stress value.

$$\sigma_{\max} = \sqrt{\sigma^2 + 3\tau^2} \quad (3.77)$$

$$\sigma_{J \text{ boo}} = \sqrt{\left(\sigma_{Jz \text{ wind}} + \sigma_{J \text{ dw } y \text{ max}} + \sigma_{x \text{ dw}} + \sigma_{J \text{ swl } y \text{ max}} + \sigma_{x \text{ swl}} \right)^2 + 3 \cdot \left(\tau_{J \text{ wind}} + \tau_{Jy \text{ dw}} + \tau_{Jz \text{ dw}} + \tau_{Jy \text{ swl}} + \tau_{Jz \text{ swl}} \right)^2} \quad (3.78)$$

$$\sigma_{\max} < \sigma_{\text{per}} \quad (3.79)$$

3.2.3. Calculation of Slewing Ring of Cranes

Three main calculations are made in the calculation of the slewing ring of the cranes. Firstly, the ball bearings are evaluated in detail according to the hertz pressures on each ball according to the ISO / TS 16281 standard. The teeth of the slewing ring can be designed according to the 6336 standards. It is possible to make detailed calculations, especially with VDI 2230 for the loads on the bolts. Bolt calculations can

also be made with simplified bolt calculations in TL Section 50 Rules. In our work, we will choose Series 06 Single-row four-point bearing slewing rings produced by ThyssenKrupp-rothe-erde company. We will realize the load combinations, which are the main reference in the design of our crane, with the table in TL Rules Chapter 50 Section 4 Table 4.3 [9], [37], [38], [13].

In the calculation of the slewing ring, the moments and vertical loads caused by wind, dead weight, and *SWL* loads will affect the calculations, just as we did in the boom calculations. The maximum moment and maximum vertical forces acting on the return gear will be found.

Since the wind coming from behind the tower will increase the moment of the rotation gear, the diameter of the tower was 1.5 times the height of the boom H , which made it easier to calculate the wind in the preliminary calculations.

In addition, the weight of the tower is affected by very different elements such as hydraulic circuits, hydraulic cylinders, slip rings, hydro motors, planetary gearbox, hoisting drum, and operator cabinet, in addition to steel structural elements. To stay on the safe side in the calculations, it is approximately twice the weight of the boom, thus making calculations easier in the preliminary calculations. Since the movements of the tower do not cause dynamic accelerations, the ϕ coefficient is not included in the weight of the tower (Section 4 Table 4.3) [9].

Moments will act on the turning gear in the x and y directions. The resultant moment acting on the gear can be found by the formula below.

$$G_{dw\ mass} = \rho_{steel} (G\ H - (G - 2t_2) (H - 2t_1))\ E \quad (3.80)$$

$$F_{x\ wind\ sring} = \text{beam distributed load} \cdot 1,5\ E \quad (3.81)$$

$$M_{sring} = \sqrt{\sum M_{x\ sring}^2 + \sum M_{y\ sring}^2} \quad (3.82)$$

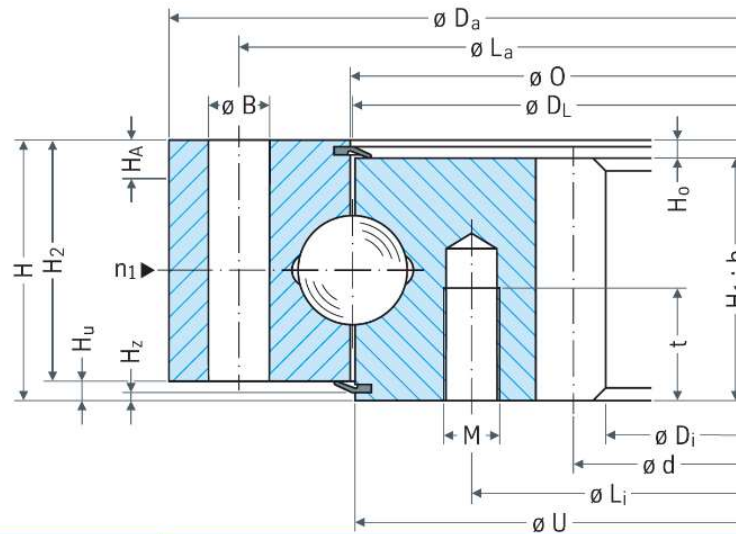
$$\sum M_{y\ sring} = F_{swl\ z} \cdot C + G_{dw} \cdot (A + F) + F_{swl\ x} \cdot E + G_{dw\ x} \cdot E + \frac{E}{2} F_{x\ wind\ sring} \quad (3.83)$$

$$\sum M_{x\ sring} = F_{swl\ y} \cdot E + G_{dw\ y} \cdot E \quad (3.84)$$

$$\sum F_{z\ sring} = F_{swl\ z} + G_{dw\ mass} + G_{dw} \quad (3.85)$$

The crane in boom design, the moments on the slewing ring, and the mathematical design of the joint at point K were made with an Excel program in which all our mathematical models run automatically. The slewing ring obtained according to the

table of ThyssenKrupp-rothe-erde company according to the found tilting moments and axial loads are selected in the images below [14].



Bearing with internal gearing		Geometry											
Drawing number		Ø D _L mm	Ø D _a mm	Ø D _i mm	H mm	Ø O mm	Ø U mm	H ₁ mm	H ₂ mm	H _u mm	H _o mm	Ø D _a * mm	Ø U* mm
1	062.25.0855.500.11.1503	855	955	710	80	854	856	71	54	26	9	953 - 0,23	855 - 0,23
2	062.25.0955.500.11.1503	955	1055	810	80	954	956	71	54	26	9	1053 - 0,26	955 - 0,23
3	062.25.1055.500.11.1503	1055	1155	910	80	1054	1056	71	54	26	9	1153 - 0,26	1055 - 0,26
4	062.25.1155.500.11.1503	1155	1255	1010	80	1154	1156	71	54	26	9	1253 - 0,31	1155 - 0,26
5	062.25.1255.500.11.1503	1255	1355	1110	80	1254	1256	71	54	26	9	1353 - 0,31	1255 - 0,31
6	062.25.1355.500.11.1503	1355	1455	1210	80	1354	1356	71	54	26	9	1453 - 0,31	1355 - 0,31
7	062.25.1455.500.11.1503	1455	1555	1310	80	1454	1456	71	54	26	9	1553 - 0,31	1455 - 0,31

Bearing with internal gearing		Geometry									
Drawing number		$\varnothing D_L$ mm	$\varnothing D_a$ mm	$\varnothing D_i$ mm	H mm	$\varnothing O$ mm	$\varnothing U$ mm	H_1 mm	H_2 mm	H_u mm	H_o mm
38	062.50.1800.001.49.1504	1800	1971	1554	109	1798	1802	100	99	10	9
39	062.50.2000.001.49.1504	2000	2171	1764	109	1998	2002	100	99	10	9
40	062.50.2240.001.49.1504	2240	2411	1984	109	2238	2242	100	99	10	9
41	062.50.2490.001.49.1504	2490	2661	2240	109	2488	2492	100	99	10	9
42	062.50.2800.001.49.1504	2800	2971	2544	109	2798	2802	100	99	10	9

Ring gear annealed

Figure 3.19: Slewing Ring Dimensions [14].

Crane Type		M_{spring} kNm	ΣF_{spring} kN
3	Load Radius 7,5m SWL 5 ton	609	98
6	Load Radius 7,5m SWL 10 ton	1.194	182
38	Load Radius 7,5m SWL 15 ton	1.778	265
38	Load Radius 7,5m SWL 20 ton	2.357	345
5	Load Radius 10m SWL 5 ton	829	111
38	Load Radius 10m SWL 10 ton	1.611	200
38	Load Radius 10m SWL 15 ton	2.397	290
39	Load Radius 10m SWL 20 ton	3.171	374
38	Load Radius 15m SWL 5 ton	1.362	159
38	Load Radius 15m SWL 10 ton	2.569	268
40	Load Radius 15m SWL 15 ton	3.761	368
41	Load Radius 15m SWL 20 ton	4.937	461
38	Load Radius 20m SWL 5 ton	2.008	213
40	Load Radius 20m SWL 10 ton	3.675	342
42	Load Radius 20m SWL 15 ton	5.361	471
42	Load Radius 20m SWL 20 ton	6.962	575

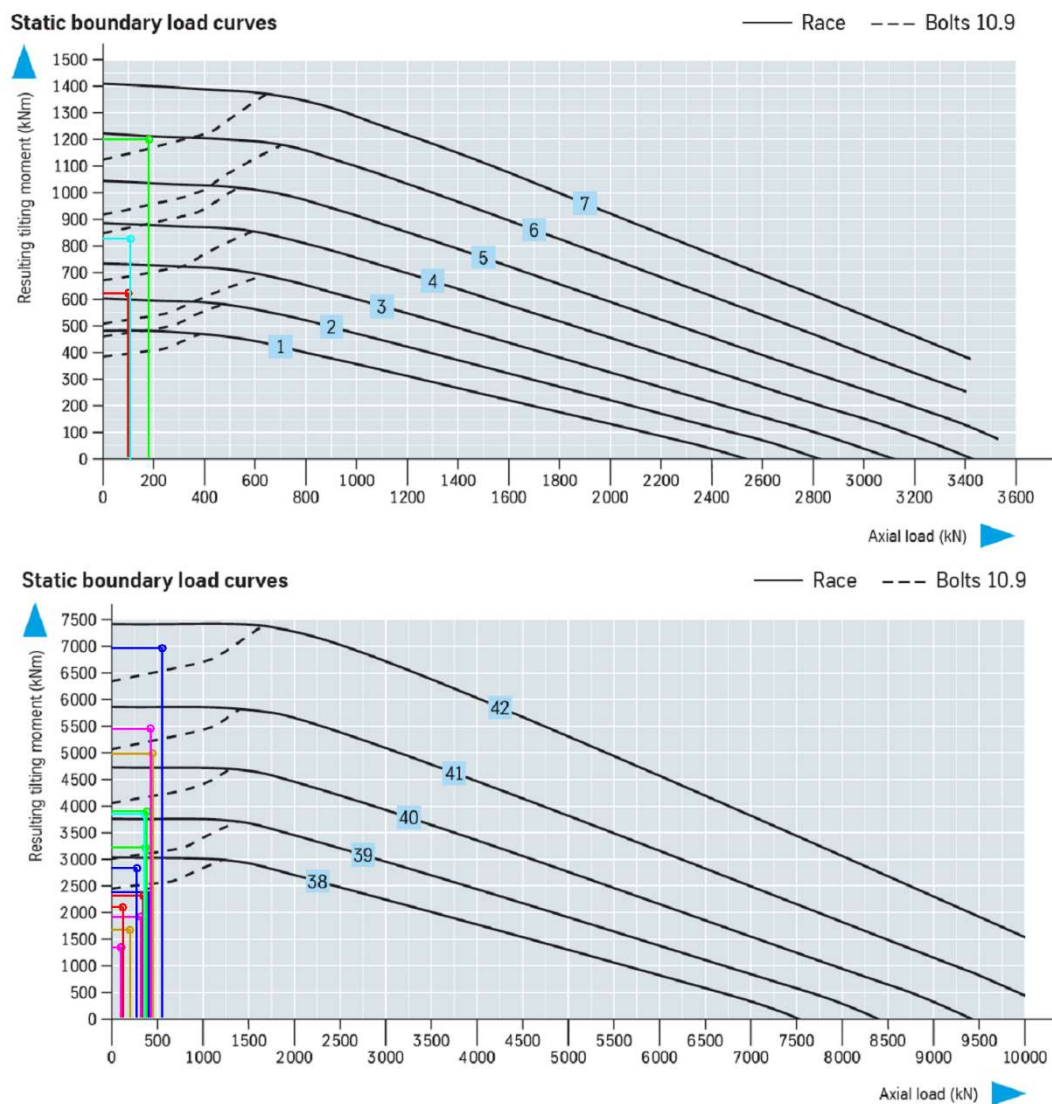


Figure 3.20: Slewing Ring Calculation Graphics [14].

3, 5, 6, 38, 39, 40, 41, 42 series slewing rings will be used in our designs. As can be seen from the image below, the top of the tower and the pedestal size of the crane are directly related to the slewing ring. Below, an example study has been carried out and a 2D general design of the lower and upper structure suitable for the 38 series slewing ring has been made. Other slewing rings are assembled similarly. As can be seen in the picture below, the cylindrical bodies of the pedestal are brought as close to the bolts as possible with the mass. A gap is created between the bolt and the pipe, which will prevent the bolt from hitting the weld and will not have a negative impact on the installation of the bolt. We can accept this gap as approximately 30mm. Slewing Ring Assembly drawing is shown in **Figure 3.21**.

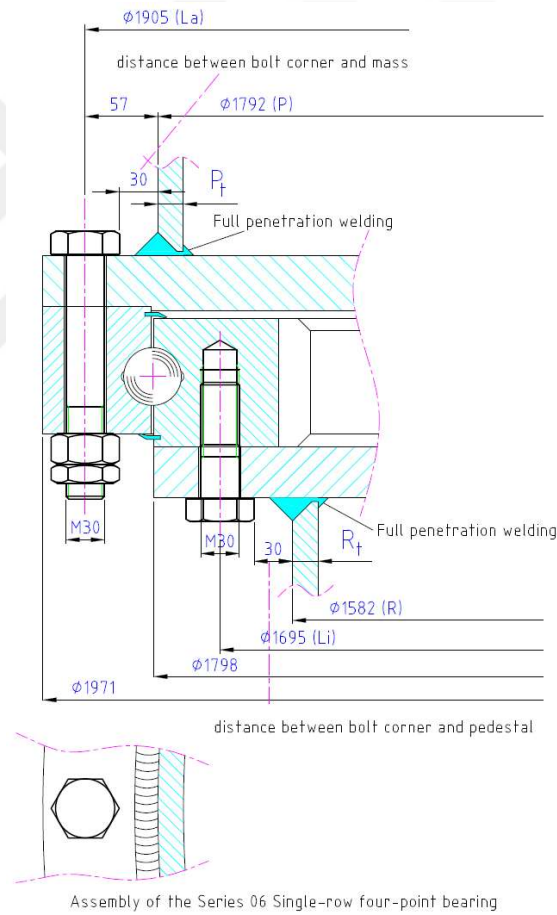


Figure 3.21: Slewing Ring Assembly Drawing.

A typical combination of Boom and mass is designed in the image below. There is a geometric relationship between the outer diameter P of the mass and the dimension M used in the connection of the boom. For this ratio, we can give the ratio $M/P = 0.77$. The measure of P can be found with the following equation.

$$P = \emptyset L_a - (\text{Bolt size according to } B) - 60\text{mm} \quad (3.86)$$

Similarly, the R measure can be found with the following formula

$$R = \emptyset L_i - (\text{Bolt size according to } B) - 60\text{mm} \quad (3.87)$$

The M , P and R values obtained according to the above formula and the ThyssenKrupp-rothe-erde company catalog are found in **Table 3.8**.

Table 3.8: Calculation of the pedestal and mass diameter.

Slewing Ring Model	$\emptyset L_a$ (mm)	$\emptyset L_i$ (mm)	P (mm)	R (mm)	M (mm)	$\emptyset B$ (mm)	Bolt Size DIN 931	Bolt Head Max (mm) DIN 931
3	1115	994	1013	892	1304	22	M24	42
5	1315	1194	1213	1092	1561	22	M24	42
6	1415	1294	1313	1192	1690	22	M24	42
38	1905	1695	1792	1582	2306	33	M30	53
39	2105	1895	1992	1782	2564	33	M30	53
40	2345	2135	2232	2022	2873	33	M30	53
41	2595	2385	2482	2272	3194	33	M30	53
42	2905	2695	2792	2582	3593	33	M30	53

3.2.4. Pedestal and Mass Thickness Calculation

The pedestal consists of a circular form. Since it is the basic calculation for the base and the mass, the stress calculations will be made with the same sheet thickness. As a separate detail, the allowable stress will change as there may be a local buckling effect on the plates in the base and mass.[9].

$$\sigma_{\text{per pedestal}} = \frac{0.8 \cdot 35.5 + 4.8}{1.6} = 207\text{N/mm}^2 (\text{Load Condition A}) \quad (3.88)$$

$$G_{\text{dw mass}} = \rho_{\text{steel}} (G H - (G - 2t_2) (H - 2t_1)) E 2 \quad (3.89)$$

$$G_{\text{dw pedestal}} = \rho_{\text{steel}} L (R^2 - (R - R_t)^2) \pi / 4 \quad (3.90)$$

$$F_{x \text{ wind pedestal}} = \text{beam distributed load } 1.5 (E + L) \quad (3.91)$$

$$M_{\text{pedestal}} = \sqrt{\sum M_x^2 \text{ pedestal} + \sum M_y^2 \text{ pedestal}} \quad (3.92)$$

$$\sum M_y \text{ pedestal} = F_{\text{swl } z} C + G_{\text{dw}} (A + F) + F_{\text{swl } x} (E + L) + G_{\text{dw } x} (E + L) + F_{x \text{ wind spring}} \frac{E+L}{2} \quad (3.93)$$

$$\sum M_x \text{ pedestal} = F_{\text{swl } y} (E + L) + G_{\text{dw } y} (E + L) \quad (3.94)$$

$$\sum F_z \text{ pedestal} = F_{\text{swl } z} + G_{\text{dw mass}} + G_{\text{dw}} + G_{\text{dw pedestal}} \quad (3.95)$$

$$W_{\text{pedestal}} = \frac{\pi (R^4 - (R - 2R_t)^4)}{32.R} \text{ mm}^3 \quad (3.96)$$

$$A_{\text{pedestal}} = \frac{\pi (R^2 - (R - R_t)^2)}{4} \text{ mm}^2 \quad (3.97)$$

$$\sigma_{\text{pedestal}} = \frac{M_{\text{pedestal}}}{W_{\text{pedestal}}} + \frac{\sum F_z \text{ pedestal}}{A_{\text{pedestal}}} \leq \sigma_{\text{per pedestal}} \quad (3.98)$$

3.2.5. Lifting Lug Calculation

While calculating the cross-sectional area of the boom in the formulas above, the calculation was made easier by considering point K as a built-in support in the Z direction. Now a more detailed calculation will be made for K .

The junction of the boom and the mass (Point K_1, K_2) is in the form of a pin under the influence of a distributed load, which has a free support feature. First, we will show the formulas that calculate the loads affecting the K_1, K_2 points, and then we will determine the stresses affecting the pins and ears. Boom and Mass connection point details are shown in **Figure 3.22**.

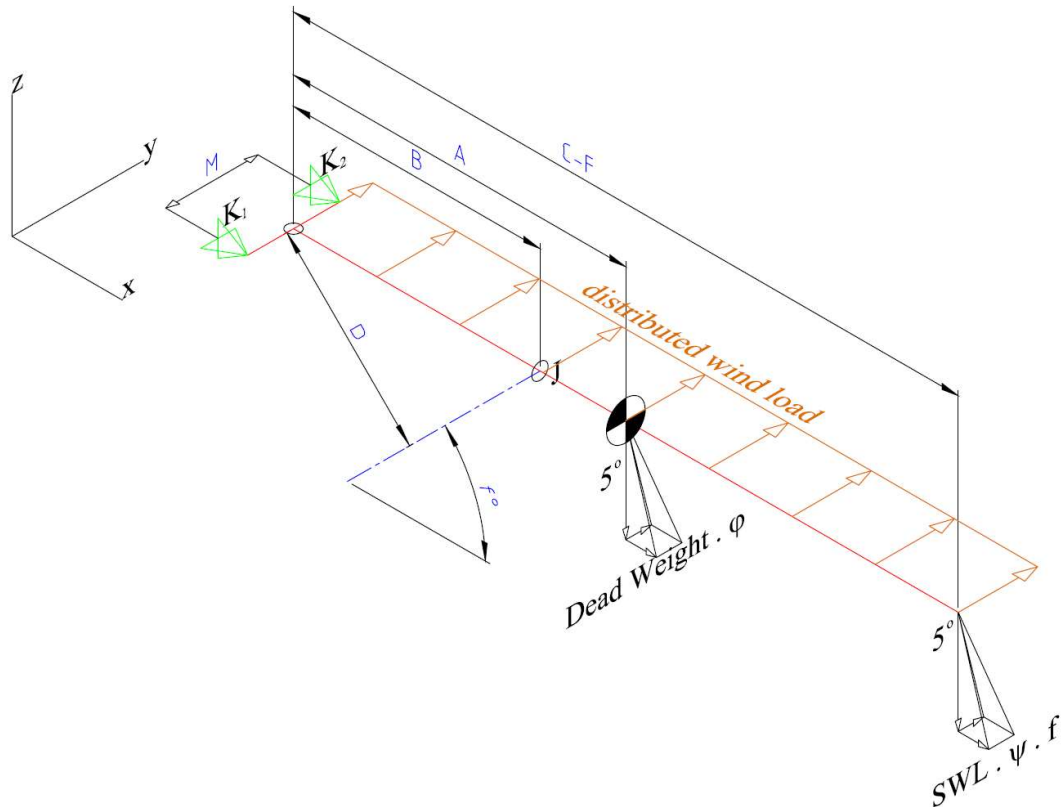


Figure 3.22: Boom and Mass connection point detail calculation

Point K_1, K_2 Calculation formulas of force caused by wind effect:

$$K_{1x} \cdot M = F_w \frac{(C-F)}{2} \quad (3.99)$$

$$K_{1xw} = F_w \cdot \frac{(C-F)}{2 \cdot M} \quad (3.100)$$

$$K_{1xw} = -K_{2xw} \quad (3.101)$$

Point K_1, K_2 Calculation formulas of force caused by SWL:

$$F_{swly} = SWL \sin(5^\circ) \psi f_e \quad F_{swlx} = SWL \sin(5^\circ) \psi f_e \quad (3.102)$$

$$F_{swlz} = SWL \psi f_e \quad (3.103)$$

$$K_{1xswl} = F_{swlz} \cdot \frac{C-F}{2 \cdot B} + \frac{F_{swlx}}{2} + F_{swly} \cdot \frac{C-F}{M} \quad (3.104)$$

$$K_{2xswl} = F_{swlz} \cdot \frac{C-F}{2 \cdot B} + \frac{F_{swlx}}{2} - F_{swly} \cdot \frac{C-F}{M} \quad (3.105)$$

$$K_{1zswl} = \frac{F_{swlz} \cdot \frac{C-F}{B} - F_{swlz}}{2} \quad (3.106)$$

$$K_{2zswl} = \frac{F_{swlz} \cdot \frac{C-F}{B} - F_{swlz}}{2} \quad (3.107)$$

Point K_1, K_2 Calculation formulas of force caused by Dead Weight:

$$G_{dwy} = G_{dw} \sin(5^\circ) \varphi \quad G_{dwx} = G_{dw} \sin(5^\circ) \varphi \quad (3.108)$$

$$G_{dw} = \rho_{steel} (G H - (G - 2t_2) (H - 2t_1)) (C - F) \varphi \quad (3.109)$$

$$K_{1xdw} = G_{dw} \frac{A}{2 \cdot B} + \frac{G_{dwx}}{2} + G_{dwy} \frac{A}{M} \quad (3.110)$$

$$K_{2xdw} = G_{dw} \frac{A}{2 \cdot B} + \frac{G_{dwx}}{2} - G_{dwy} \frac{A}{M} \quad (3.111)$$

$$K_{1zdw} = \frac{G_{dw} \frac{A}{B} - G_{dw}}{2} \quad (3.112)$$

$$K_{2zdw} = \frac{G_{dw} \frac{A}{B} - G_{dw}}{2} \quad (3.113)$$

The total forces occurring in lifting lugs can be calculated with the following formulas:

$$F_{llug} = K_{1F} = \sqrt{(K_{1xw} + K_{1xswl} + K_{1xdw})^2 + (K_{1zswl} + K_{1zdw})^2} \quad (3.114)$$

$$F_{llug} = K_{2F} = \sqrt{(K_{2xw} + K_{2xswl} + K_{2xdw})^2 + (K_{2zswl} + K_{2zdw})^2} \quad (3.115)$$

With the help of the formulas above, the force acting on the lifting lugs has been found. Now, the stresses affecting the lifting lug and the pin will be found. There are many alternative connection methods for the connections between the mass and the boom. Changes can be made in the designs according to many factors such as the dimensions of the crane, ease of manufacturing, accessibility to resources, homogeneous transfer

of stresses, and sudden structural transition problems. In this study, we prefer the connection of the tower as a single piece and the boom as two forks. Distributed loads on the pin in the lifting lug. It has an impact. In response to this distributed load, the reaction force from the ear of the mass acts. In this way, a certain amount of working tolerance must be left for the pins to move and be lubricated. In addition, wear will be observed on the bearing elements and pins due to years of heavy working conditions and corrosive sea effects. A common mistake made in this type of pin is to calculate only the shear loads. For this reason, the built-in support effect should not be considered in these pins and formulations should be made so that maximum stresses occur in the middle of the pin. Bending stresses can cause serious problems, especially when the pin length is high. In addition, since the lifting lug is a load acting through the circular hole, it creates a discontinuity that causes non-homogeneous notch effects. We need to include this effect in the calculation of Lifting Lug's strength (σ_{llug}) In addition, surface pressure effects affecting the structure should also be calculated [39]. Also, the calculation details of the lifting lug are shown in **Figure 3.23**.

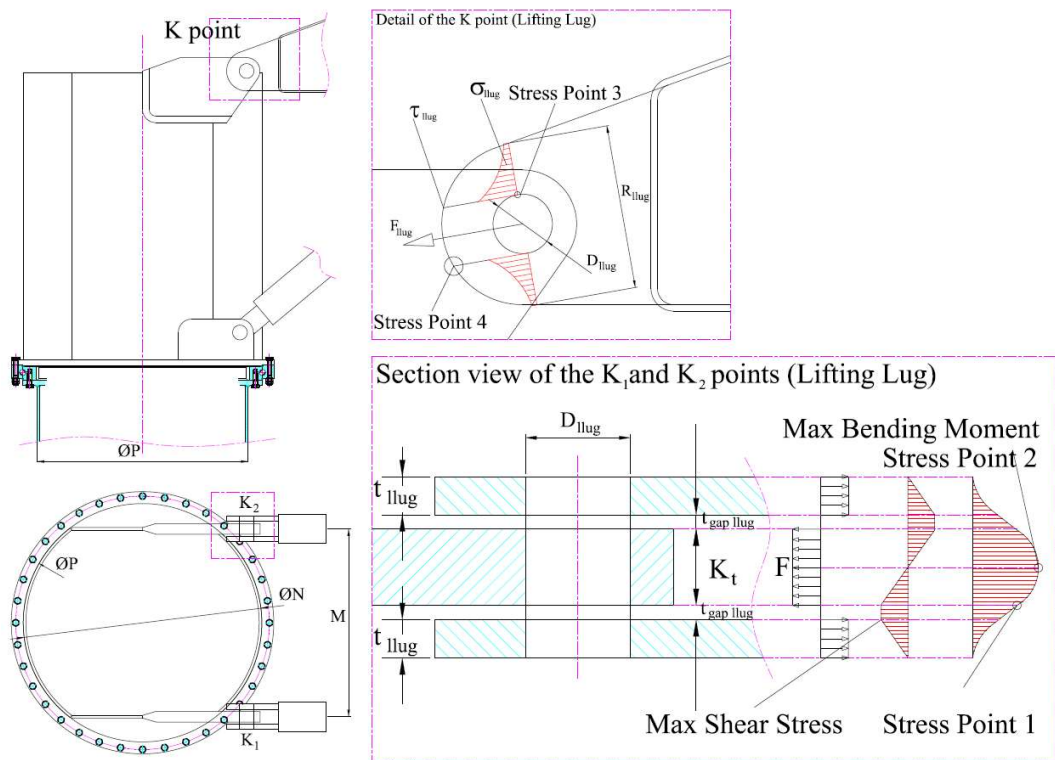


Figure 3.23: Calculation detail of the lifting lug

Permissible stress for lifting lug design:

$$\sigma_{per} = \frac{0.8 \cdot 35.5 + 4.8}{1.4} = 237 \text{ N/mm}^2 \quad (3.116)$$

Pin bending moment of inertia:

$$W_{lug \text{ pin}} = \frac{\pi d_{lug}^3}{32} \text{ mm}^3 \quad (3.117)$$

Pin Section area:

$$A_{lug \text{ pin}} = \frac{\pi d_{lug}^2}{4} \quad (3.118)$$

$$\sigma_{\text{stress point 1}} = \sqrt{\left(\frac{\frac{F_{lug} \cdot t_{lug}}{2} + \frac{F_{lug} \cdot t_{gap \text{ lug}}}{2}}{W_{lug \text{ pin}}} \right)^2 + 3 \left(\frac{F_{lug}}{2 A_{lug \text{ pin}}} \right)^2} \quad (3.119)$$

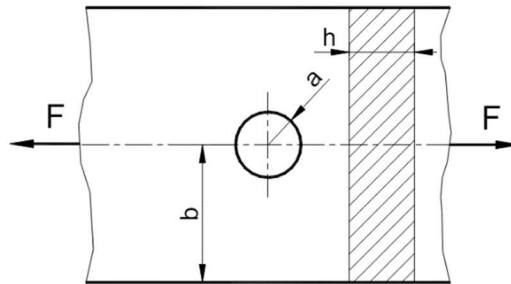
$$\sigma_{\text{stress point 2}} = \frac{\frac{F_{lug} \cdot K_t}{8} + \frac{F_{lug} \cdot t_{gap \text{ lug}}}{2} + \frac{F_{lug} \cdot t_{lug}}{4}}{W_{lug \text{ pin}}} \quad (3.120)$$

Stress point 3 for boom lifting lug:

$$\sigma_{\text{stress point 3}} = \frac{F_{lug}}{(R_{lug} - D_{lug}) \cdot t_{lug.2}} \alpha_{\zeta t} \quad (3.121)$$

Stress point 3 for Mass lifting lug:

$$\sigma_{\text{stress point 3}} = \frac{F_{lug}}{(R_{lug} - D_{lug}) K_t} \alpha_{\zeta t} \quad (3.122)$$



$$\alpha_{\zeta t} = \frac{3 + \left(\frac{a}{b} \right)^2}{1 + \frac{a}{b}}$$

Figure 3.24: Notch factor technical dimensions [39].

Surface pressure calculation:

$$\sigma_{Kt \text{ surf}} = \frac{F_{llug}}{D_{llug} \cdot K_t} \quad (3.123)$$

$$\sigma_{t \text{ llug surf}} = \frac{F_{llug}}{D_{llug} \cdot t_{llug}^2} \quad (3.124)$$

3.2.6. Design of the Hydraulic Cylinder

We will design according to DNV-ST-0194 for the design and calculation of hydraulic cylinders in offshore structures. The main purpose of our study is to determine the spring coefficient of the crane and we need the spring coefficient in the closed position of the hydraulic cylinder of the crane. Pipe back cover thickness, and eye bolt diameters We do not need to make calculations such as the strength of the teeth in threaded connections. Since there is no oil stuck in the rear area of the cylinder in the closed position, we will not need to calculate the compressibility of the oil. The most important factor in calculating the diameter of the rod in hydraulic cylinders is the buckling calculation. Since the buckling formula prepared by DNV includes the inertia of the cylinder pipe in the calculation, we will determine the lifting capacity of the cylinder, the tube diameter, and the open length of the cylinder, and with the evaluations, we obtain from these calculations, the rod diameter and therefore the spring coefficient of the hydraulic cylinder [33].

In offshore cranes, the double safety philosophy we call redundancy is applied to every possible point. Since hydraulic cylinders are vital components for the safety of life and property, they are used in pairs whenever possible in cranes. This application is preferred to reduce vital risks, as can be seen in the FMEA (Failure mode and effects analysis) studies prepared especially before the crane is designed [40].

In one important detail, load holding or hose burst valves are connected to the hydraulic cylinders, which are connected to steel pipes (hoses are unacceptable) [9].

Generally, in the hydraulic systems of cranes, variable displacement proportional pumps with a working capacity between 200 and 350 bar, IP protection, and corrosive-resistant electromechanical valves, installation pipes, and components are preferred. The components used will need to be specifically certified. In this study, we will use 250 bar working pressure as a reference. We will choose the cylinder tube as cold-

drawn honed St52 tube rod material and the Ck45 chrome-coated solid cylindrical material. Hydraulic lifting capacity of the hydraulic cylinder [12], [18], [33].

The weight of the boom, SWL and the vertical distance of the hydraulic cylinder to the center of rotation (point K) affect the load on the cylinder.

$$F_{\text{cylinder}} = (F_{\text{swl}} z (C - F) + G_{\text{dw}} B) / (D/2) \quad (3.125)$$

Bore inside diameter of the hydraulic cylinder:

$$F_{\text{cylinder}} = \frac{\pi \cdot d_{\text{cylinder}}^2}{4} P_{\text{hydraulic}} \quad (3.126)$$

$$d_{\text{cylinder}} = \sqrt{\frac{4 F_{\text{cylinder}}}{\pi \cdot 25}} \quad (3.127)$$

$$t = \frac{P \cdot D_o}{20 \cdot \sigma_t \cdot e + p} + \bar{c} \quad (3.128)$$

$$R_{eH} (\text{st52}) = 355 \text{N/mm}^2 \quad (3.129)$$

$$R_m (\text{st52}) = 520 \text{N/mm}^2 \quad (3.130)$$

$$R_{eH} (\text{Ck45}) = 330 \text{N/mm}^2 \quad (3.131)$$

$$R_m (\text{Ck45}) = 580 \text{N/mm}^2 \quad (3.132)$$

$$\sigma_t (\text{St52}) = \frac{355}{1.7} = 208 \text{N/mm}^2 \text{ or } \sigma_t (\text{St52}) = \frac{520}{2.7} = 192 \text{N/mm}^2 \quad (3.133)$$

$$\sigma_t (\text{Ck45}) = \frac{330}{1.7} = 194 \text{N/mm}^2 \text{ or } \sigma_t (\text{Ck45}) = \frac{580}{2.7} = 215 \text{N/mm}^2 \quad (3.134)$$

$$t_{\text{cylinder}} = \frac{250 (d_{\text{cylinder}} + 2 t_{\text{cylinder}})}{20 \cdot 192 + 250} + 0.3 \quad (3.135)$$

$$t_{\text{cylinder}} = \frac{250 \cdot d_{\text{cylinder}} + 1227}{3590} \quad (3.136)$$

For the buckling calculation of the hydraulic cylinder, facilitate the calculations, we give a ratio of 0.7 between the L_2 and L_1 dimensions of our hydraulic cylinders and a ratio of 0.9 between the closed length of the cylinder and the L_2 dimension, giving details such as kinematic calculations arising from the level luffing motion of the hydraulic cylinders of the cranes. We will not go into this study. Below we see the buckling calculation formulas from DNV [33].

I_1 and I_2 shall be calculated as:

$$I_1 = \frac{\pi (D_0^4 - D_1^4)}{64} \quad (3.137)$$

$$I_2 = \frac{\pi (d_0^4 - d_1^4)}{64} \quad (3.138)$$

A Parameter, called Z shall be determined as:

$$z = \frac{L_1}{I_1} + \frac{L_2}{I_2} + \left(\frac{1}{I_2} - \frac{1}{I_1} \right) \frac{L}{2\pi} \sin \left(2\pi \frac{L_1}{L} \right) \quad (3.139)$$

Buckling load is equal to:

$$F_E = \frac{E \pi^2}{L Z} \quad (3.140)$$

Buckling safety factor resulting from method in shall not be less than 4, therefore

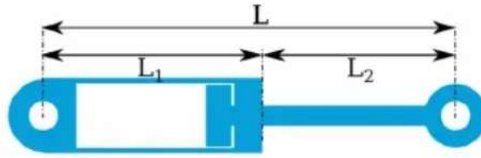


Figure 3.25: Hydraulic Cylinder Buckling Dimensions [33].

Cylinder pipe diameter, thickness and rod dimensions obtained according to the calculation program prepared according to the formulas from DNV CLASS GUIDELINE are shown in the **Table 3.9**.

Buckling Safety Criteria of the cylinder is:

$$\frac{F_E}{F_{\text{cylinder}}} \geq 4 \quad (3.141)$$

Table 3.9: Buckling Calculation

Hydraulic Cylinder Buckling and Tickness Calculation											
	Ørod diameter (mm)	Cylinder Closed L (mm)	$F_{cylinder}$ N	$d_{cylinder}$ (mm)	$t_{cylinder}$ (mm)	L1 (mm)	L2 (mm)	I1 mm^4	I2 mm^4	F_E (N)	Buckling Safety
Load Radius 7,5m SWL 5 ton	85	2315	149447,9	90	7	2084	1458	2521906	2562392	628804	4,2
Load Radius 7,5m SWL 10 ton	85	2315	293653,1	130	10	2084	1458	10830641	2562392	1259992	4,3
Load Radius 7,5m SWL 15 ton	95	2315	437698,3	150	11	2084	1458	18111432	3998198	2006931	4,6
Load Radius 7,5m SWL 20 ton	95	2315	580652,8	180	13	2084	1458	36867282	3998198	2354554	4,1
Load Radius 10m SWL 5 ton	120	3050	180215,4	100	8	2745	1922	3979217	10178760	722775	4,0
Load Radius 10m SWL 10 ton	110	3050	351038,6	140	11	2745	1922	14951406	7186884	1549016	4,4
Load Radius 10m SWL 15 ton	115	3050	522745,4	170	13	2745	1922	31444351	8585414	2323085	4,4
Load Radius 10m SWL 20 ton	120	3050	691851,3	190	14	2745	1922	46894189	10178760	2957886	4,3
Load Radius 15m SWL 5 ton	105	3035	300909,9	130	10	2732	1912	10830641	5966602	1215146	4,0
Load Radius 15m SWL 10 ton	115	3035	569086,4	180	13	2732	1912	36867282	8585414	2468028	4,3
Load Radius 15m SWL 15 ton	125	3035	834040,2	210	15	2732	1912	67394526	11984225	3709312	4,4
Load Radius 15m SWL 20 ton	130	3035	1095378,4	240	18	2732	1912	121983280	14019848	4756097	4,3
Load Radius 20m SWL 5 ton	135	3645	371376,2	140	11	3281	2296	14951406	16304406	1540232	4,1
Load Radius 20m SWL 10 ton	135	3645	681464,5	190	14	3281	2296	46894189	16304406	2829182	4,2
Load Radius 20m SWL 15 ton	140	3645	995046,7	230	17	3281	2296	101076935	18857410	3997878	4,0
Load Radius 20m SWL 20 ton	150	3645	1293196,9	260	19	3281	2296	162792934	24850489	5526427	4,3

Calculation program has been prepared to generalize the calculation formulas for the fixed boom and drum deck crane moved by the bottom hydraulic cylinder that we found above and to use them in future studies. Below **Table 3.10**, the input and output values of the calculation program prepared for 16 cranes are shown. Also, **Table 3.11** and **Table 3.12** the Crane stress calculations are to be shown:

Table 3.10: Crane calculation program's input and result columns

Crane Calculation Program												
Crane Type	SWL (ton)	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)	F (mm)	G (mm)	H (mm)	L (mm)	t1 (mm)	t2 (mm)
Load Radius 7.5m SWL 5 ton	5	3636	2671	7500	1886	2600	297	250	400	2000	12	10
Load Radius 7.5m SWL 10 ton	10	3746	2671	7500	1886	2600	297	250	550	2000	15	12
Load Radius 7.5m SWL 15 ton	15	3746	2671	7500	1886	2600	297	350	650	2000	15	12
Load Radius 7.5m SWL 20 ton	20	3746	2671	7500	1886	2600	297	350	800	2000	15	12
Load Radius 10m SWL 5 ton	5	4750	3924	10000	2167	2600	297	250	450	2000	14	10
Load Radius 10m SWL 10 ton	10	4870	3924	10000	2167	2600	297	320	600	2000	14	12
Load Radius 10m SWL 15 ton	15	4870	3924	10000	2167	2600	297	350	700	2000	16	14
Load Radius 10m SWL 20 ton	20	4870	3924	10000	2167	2600	297	400	800	2000	16	14
Load Radius 15m SWL 5 ton	5	6890	3924	15000	2167	2600	297	350	600	2000	14	12
Load Radius 15m SWL 10 ton	10	6990	3924	15000	2167	2600	297	400	800	2000	16	14
Load Radius 15m SWL 15 ton	15	6990	3924	15000	2167	2600	297	500	950	2000	16	14
Load Radius 15m SWL 20 ton	20	6990	3924	15000	2167	2600	297	500	1150	2000	16	14
Load Radius 20m SWL 5 ton	5	9243	4709	20000	2601	2600	356	400	700	2000	16	12
Load Radius 20m SWL 10 ton	10	9393	4709	20000	2601	2600	356	450	1000	2000	16	14
Load Radius 20m SWL 15 ton	15	9393	4709	20000	2601	2600	356	550	1150	2000	18	15
Load Radius 20m SWL 20 ton	20	9393	4709	20000	2601	2600	356	550	1300	2000	20	15
	ØP (mm)	R (mm)	D_{lug} (mm)	K_t (mm)	t_{lug} (mm)	$t_{gap lug}$ (mm)	R_{lug} (mm)	Rt (mm)	ϕ	ψ	f_e	
Load Radius 7.5m SWL 5 ton	1013	892	60	30	10	5	150	8	1,2	1,3663	1,1	
Load Radius 7.5m SWL 10 ton	1313	1192	70	40	20	5	150	8	1,1	1,3663	1,1	
Load Radius 7.5m SWL 15 ton	1792	1582	80	40	20	5	200	8	1,1	1,3663	1,1	
Load Radius 7.5m SWL 20 ton	1792	1582	90	60	30	5	200	8	1,1	1,3663	1,1	
Load Radius 10m SWL 5 ton	1213	1092	60	30	10	5	200	8	1,2	1,3663	1,1	
Load Radius 10m SWL 10 ton	1792	1582	70	25	10	5	250	8	1,1	1,3663	1,1	
Load Radius 10m SWL 15 ton	1792	1582	80	40	20	5	200	8	1,1	1,3663	1,1	
Load Radius 10m SWL 20 ton	1992	1782	90	60	25	5	250	8	1,1	1,3663	1,1	
Load Radius 15m SWL 5 ton	1792	1582	60	50	20	5	200	8	1,2	1,3663	1,1	
Load Radius 15m SWL 10 ton	1792	1582	80	50	20	5	250	8	1,1	1,3663	1,1	
Load Radius 15m SWL 15 ton	2232	2022	100	70	30	5	250	8	1,1	1,3663	1,1	
Load Radius 15m SWL 20 ton	2482	2272	110	80	40	5	250	8	1,1	1,3663	1,1	
Load Radius 20m SWL 5 ton	1792	1582	80	40	20	5	200	8	1,2	1,3663	1,1	
Load Radius 20m SWL 10 ton	2232	2022	90	40	20	5	300	8	1,1	1,3663	1,1	
Load Radius 20m SWL 15 ton	2792	2582	100	40	20	5	350	8	1,1	1,3663	1,1	
Load Radius 20m SWL 20 ton	2792	2582	110	50	25	5	400	10	1,1	1,3663	1,1	
	σ_{boom} N/mm ²	$\sigma_{pedestal}$ N/mm ²	$\sigma_{stress point 1}$ N/mm ²	$\sigma_{stress point 2}$ N/mm ²	$\sigma_{stress point 3}$ boom N/mm ²	$\sigma_{stress point 3}$ mass N/mm ²	$\sigma_{Kt surf}$ N/mm ²	$\sigma_{t lug surf}$ N/mm ²				
Load Radius 7.5m SWL 5 ton	221	133	120	78	236	158	105	157				
Load Radius 7.5m SWL 10 ton	233	146	183	126	233	233	121	121				
Load Radius 7.5m SWL 15 ton	219	125	182	117	221	221	147	147				
Load Radius 7.5m SWL 20 ton	218	165	211	152	208	208	115	115				
Load Radius 10m SWL 5 ton	220	120	123	80	164	109	107	161				
Load Radius 10m SWL 10 ton	232	111	146	78	217	173	185	232				
Load Radius 10m SWL 15 ton	231	165	187	120	226	226	150	150				
Load Radius 10m SWL 20 ton	233	173	194	140	177	148	114	137				
Load Radius 15m SWL 5 ton	228	93	236	193	126	101	99	124				
Load Radius 15m SWL 10 ton	234	175	215	151	191	153	138	173				
Load Radius 15m SWL 15 ton	234	158	198	144	190	163	108	126				
Load Radius 15m SWL 20 ton	236	164	226	166	191	191	110	110				
Load Radius 20m SWL 5 ton	237	137	157	101	190	190	127	127				
Load Radius 20m SWL 10 ton	236	153	198	118	191	191	187	187				
Load Radius 20m SWL 15 ton	227	138	209	116	219	219	228	228				
Load Radius 20m SWL 20 ton	235	143	230	135	196	196	214	214				

Table 3.11: Crane calculation program's stress calculation columns

	σ_{per} N/mm ²	$\sigma_{per\ pedestal}$ N/mm ²	Wy (mm ³)	Wz (mm ³)	A_{crane} mm ²	Aside (mm ²)	Beam Dist Load (N/mm)	Total Wind Force (N)	$F_{j\ wind}$ (N)	$M_{zw\ max}$ Nmm	$M_{j\ wz}$ Nmm	$\sigma_{zw\ wind\ max}$ N/mm ²
Load Radius 7.5m SWL 5 ton	237	208	1.572.418	1.116.805	13.520	2.881.200	0	1.801	1.133	6.485.401	2.567.378	6
Load Radius 7.5m SWL 10 ton	237	208	2.974.649	1.727.532	19.980	3.961.650	0	2.476	1.558	8.917.427	3.530.145	5
Load Radius 7.5m SWL 15 ton	237	208	4.724.057	3.042.021	25.380	4.681.950	0	2.926	1.841	10.538.777	4.171.989	3
Load Radius 7.5m SWL 20 ton	237	208	6.327.134	3.629.809	28.980	5.762.400	1	3.602	2.266	12.970.802	5.134.756	4
Load Radius 10m SWL 5 ton	237	208	2.035.710	1.264.517	15.440	4.366.350	0	2.729	1.625	13.239.592	4.696.431	10
Load Radius 10m SWL 10 ton	237	208	3.812.173	2.513.729	22.688	5.821.800	0	3.639	2.167	17.652.789	6.261.908	7
Load Radius 10m SWL 15 ton	237	208	5.730.715	3.671.660	29.904	6.792.100	0	4.245	2.528	20.594.921	7.305.559	6
Load Radius 10m SWL 20 ton	237	208	7.560.342	4.860.102	34.304	7.762.400	1	4.852	2.890	23.537.052	8.349.210	5
Load Radius 15m SWL 5 ton	237	208	4.052.596	2.813.096	23.528	8.821.800	0	5.514	4.042	40.533.414	21.785.033	14
Load Radius 15m SWL 10 ton	237	208	7.560.342	4.860.102	34.304	11.762.400	1	7.352	5.390	54.044.552	29.046.710	11
Load Radius 15m SWL 15 ton	237	208	11.147.116	7.406.195	41.704	13.967.850	1	8.730	6.400	64.177.906	34.492.968	9
Load Radius 15m SWL 20 ton	237	208	14.617.044	8.729.258	47.304	16.908.450	1	10.568	7.747	77.689.044	41.754.646	9
Load Radius 20m SWL 5 ton	237	208	5.981.622	3.871.197	28.832	13.750.800	0	8.594	6.534	84.412.724	48.793.112	22
Load Radius 20m SWL 10 ton	237	208	11.204.907	6.806.814	41.504	19.644.000	1	12.278	9.334	120.589.605	69.704.445	18
Load Radius 20m SWL 15 ton	237	208	17.043.056	10.513.315	53.220	22.590.600	1	14.119	10.735	138.678.046	80.160.112	13
Load Radius 20m SWL 20 ton	237	208	21.558.267	11.854.976	59.800	25.537.200	1	15.961	12.135	156.766.487	90.615.779	13
	$\sigma_{jz\ wind}$ N/mm ²	$\tau_{j\ wind}$ N/mm ²	G_{dw} N	$G_{dw\ y}$ N	$G_{dw\ x}$ N	$M_{j\ dw\ y\ max}$ Nmm	$\sigma_{j\ dw\ y\ max}$ N/mm ²	$\sigma_{j\ dw\ z}$ N/mm ²	$M_{j\ dw\ z}$ Nmm	$\sigma_{x\ dw}$ N/mm ²	$\sigma_{j\ dw\ z}$ N/mm ²	$\tau_{j\ y\ dw}$ N/mm ²
Load Radius 7.5m SWL 5 ton	2	0	8.999	784	784	8.684.350	6	2.851.871	756.891	0	1	0
Load Radius 7.5m SWL 10 ton	2	0	12.191	1.063	1.063	13.105.353	4	3.980.192	1.142.207	0	1	0
Load Radius 7.5m SWL 15 ton	1	0	15.486	1.350	1.350	16.647.340	4	5.055.920	1.450.911	0	0	0
Load Radius 7.5m SWL 20 ton	1	0	17.682	1.541	1.541	19.008.665	3	5.773.071	1.656.714	0	0	0
Load Radius 10m SWL 5 ton	4	0	13.844	1.207	1.207	11.435.451	6	5.731.428	996.665	0	1	0
Load Radius 10m SWL 10 ton	2	0	18.648	1.625	1.625	17.641.062	5	7.915.139	1.537.520	0	1	0
Load Radius 10m SWL 15 ton	2	0	24.579	2.142	2.142	23.251.865	4	10.432.578	2.026.534	0	1	0
Load Radius 10m SWL 20 ton	2	0	28.196	2.457	2.457	26.673.087	4	11.967.601	2.324.713	0	0	0
Load Radius 15m SWL 5 ton	8	0	31.968	2.786	2.786	94.816.086	23	19.196.679	8.263.766	0	3	0
Load Radius 15m SWL 10 ton	6	0	42.725	3.724	3.724	130.994.850	17	26.028.866	11.416.953	0	2	0
Load Radius 15m SWL 15 ton	5	0	51.942	4.527	4.527	159.252.834	14	31.643.769	13.879.799	0	2	0
Load Radius 15m SWL 20 ton	5	0	58.916	5.135	5.135	180.637.254	12	35.892.884	15.743.574	0	2	0
Load Radius 20m SWL 5 ton	13	1	52.339	4.562	4.562	237.304.580	40	42.163.200	20.682.457	0	5	0
Load Radius 20m SWL 10 ton	10	1	69.064	6.019	6.019	323.495.404	29	56.539.447	28.194.482	0	4	0
Load Radius 20m SWL 15 ton	8	1	88.560	7.718	7.718	414.813.643	24	72.499.744	36.153.391	0	3	0
Load Radius 20m SWL 20 ton	8	1	99.509	8.673	8.673	466.100.260	22	81.463.448	40.623.314	0	3	0

Table 3.12: Crane calculation program's stress calculation columns

	$\tau_{j \pm dw}$ N/mm ²	$M_{j \pm dw \max}$ Nmm	$\sigma_{j \pm dw \max}$ N/mm ²	$F_{j \pm dw}$ N	$F_{j \pm dw}$ N	$F_{j \pm dw}$ N	$F_{j \pm dw}$ N	$\Sigma M_{j \pm dw}$ Nmm	$\tau_{j \pm dw}$ N	$F_{j \pm dw}$ N	$F_{j \pm dw}$ N	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm dw}$ Nmm	$\Sigma M_{j \pm 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3.3. Finding Spring Constants of Cranes Using FEA Calculations

16 crane shell elements were modeled with the help of a 3D model prepared parametrically. In structural steel calculations, the shell model is preferred because it gives fast results, gives more realistic results in plate type structures, and lastly creates a mesh as quadrilateral elements. The image below shows the general dimensions of the first of these 16 cranes.

We can find the spring coefficients of cranes with engineering formulas, as well as with FEA calculations with the help of a parametrically modeled 3D model. In this part of the study, we will find the spring coefficients of the cranes in more detail regarding the measurements we found in the calculation program. In addition, by comparing the results we found with engineering formulas with FEA results, we will prove the accuracy of the results we found with engineering formulas [36]. As you can see **Figure 3.26** dimensions have equations with the other dimensions. Basic parameters of the FEA analysis data are shown in **Table 3.13**.

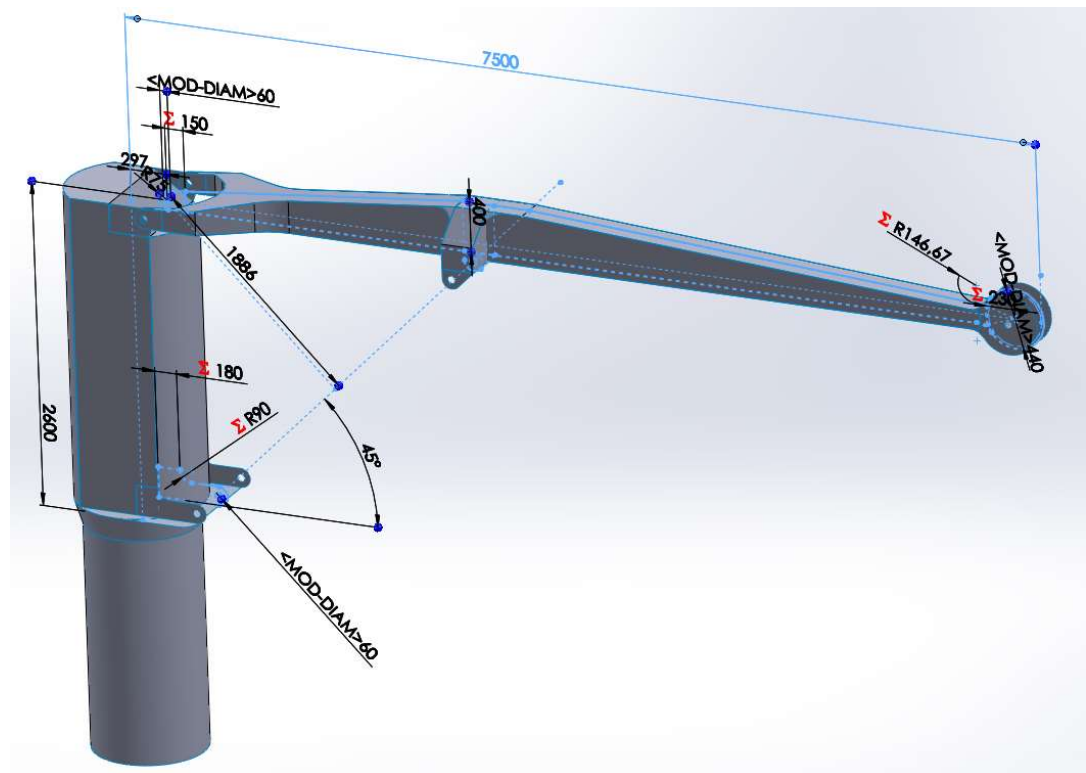


Figure 3.26: Sample Crane 3d shell model Parametric Modeling 7.5m 5 ton

Table 3.13: FEA Calculation Steel Material Data.

Property	Value
Density	7.85e-06 kg/mm ³
Young's Modulus	2e+05 MPa
Poisson's Ratio	0.30000
Bulk Modulus	1.6667e+05 MPa
Shear Modulus	76923 MPa

The thickness of the shell elements is indicated by the colors in the upper left corner of the program in **Figure 3.27**

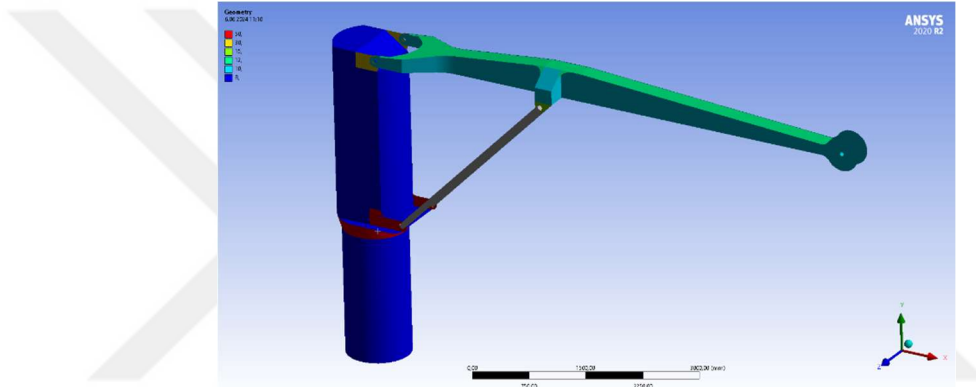


Figure 3.27: Thickness of the crane parts (Shell model)

Hydraulic cylinders are defined as beam elements. The rod thickness value in the buckling calculation has been transferred to the program, and it is shown in **Figure 3.28**.

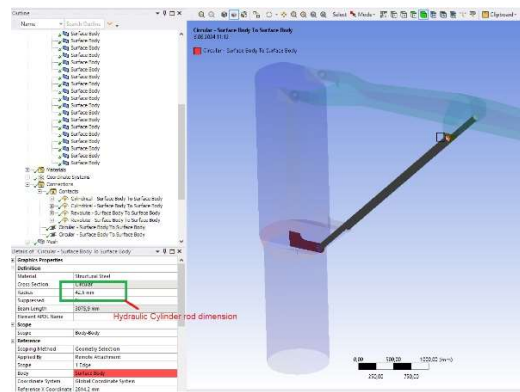


Figure 3.28: Definition the hydraulic cylinder section [33].

The input and output data of the FEA program made for the verification of the prepared calculation formula are shown in **Figure 3.29**, **Figure 3.30**, **Figure 3.31**, **Figure 3.32**, **Figure 3.33**, **Figure 3.34**, **Figure 3.35**, **Figure 3.36** and **Figure 3.37** below.

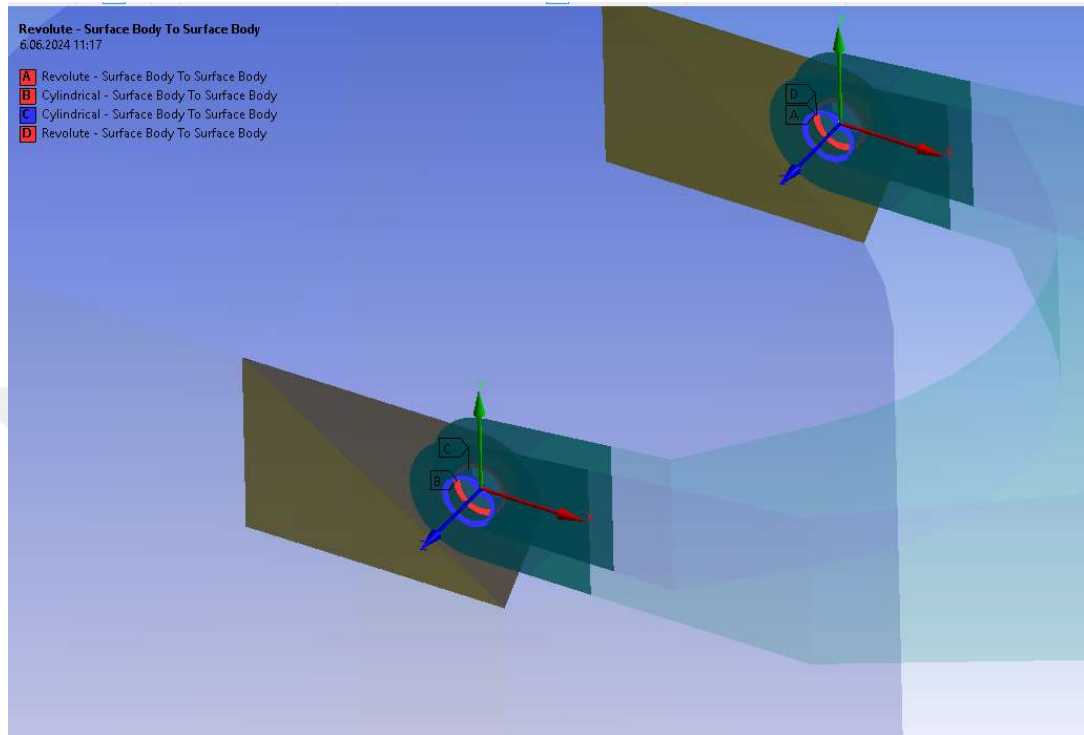


Figure 3.29: Rotational pin definition

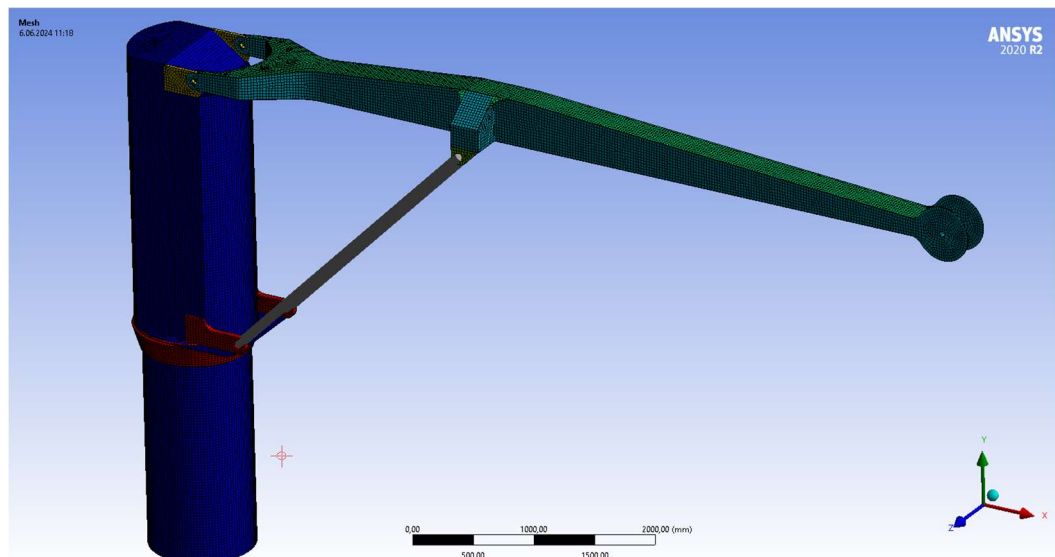


Figure 3.30: Shell mesh view

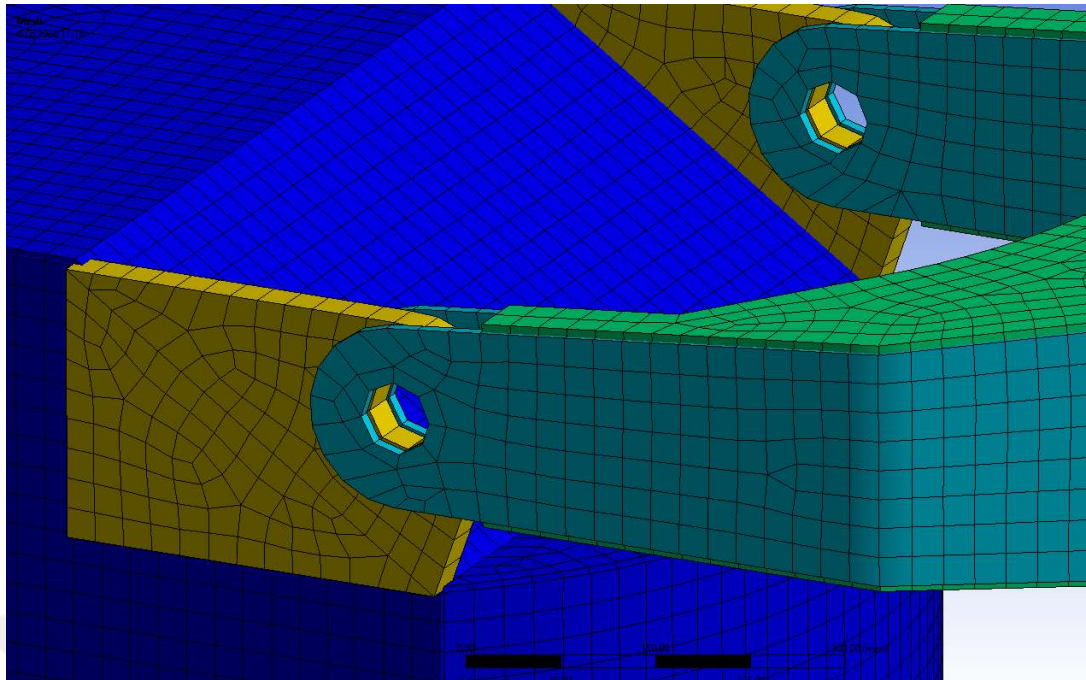


Figure 3.31: Lifting lug shell mesh detail

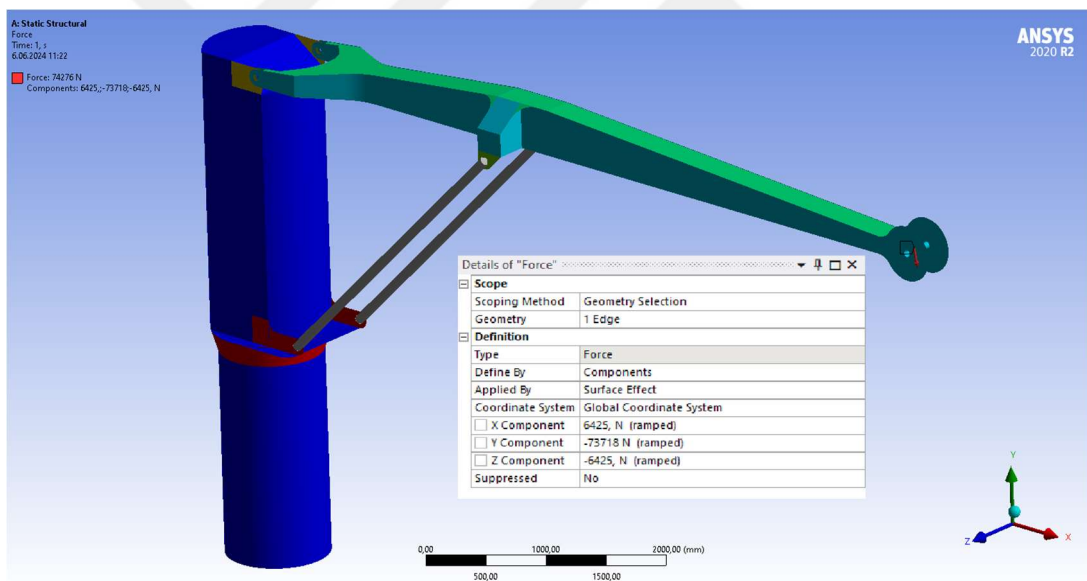


Figure 3.32: SWL Load Definition

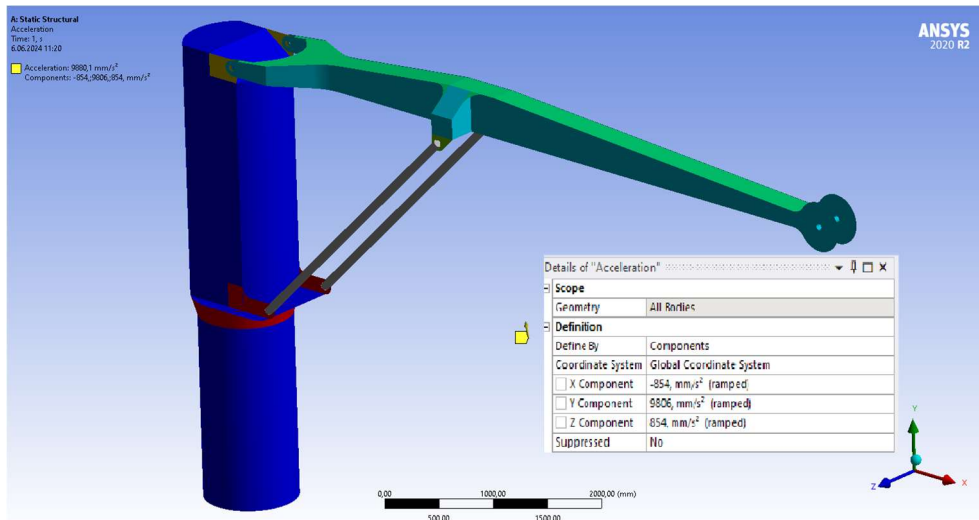


Figure 3.33: Definition of the Dead weight

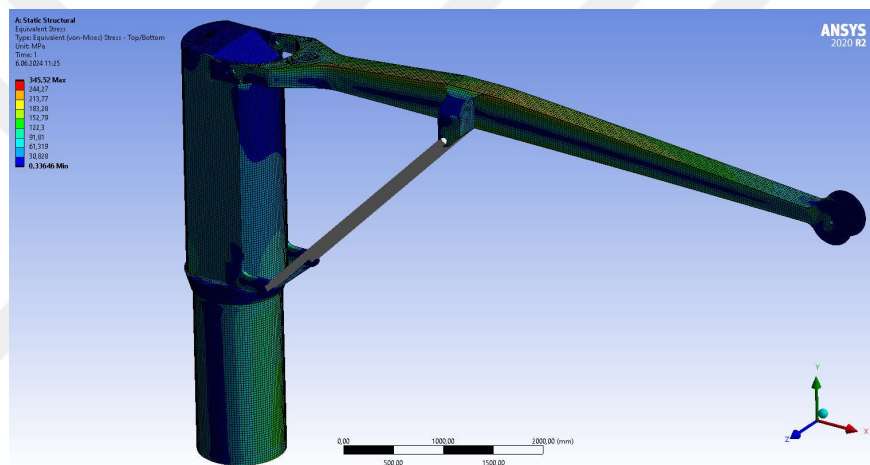


Figure 3.34: Von Misses Stress Result

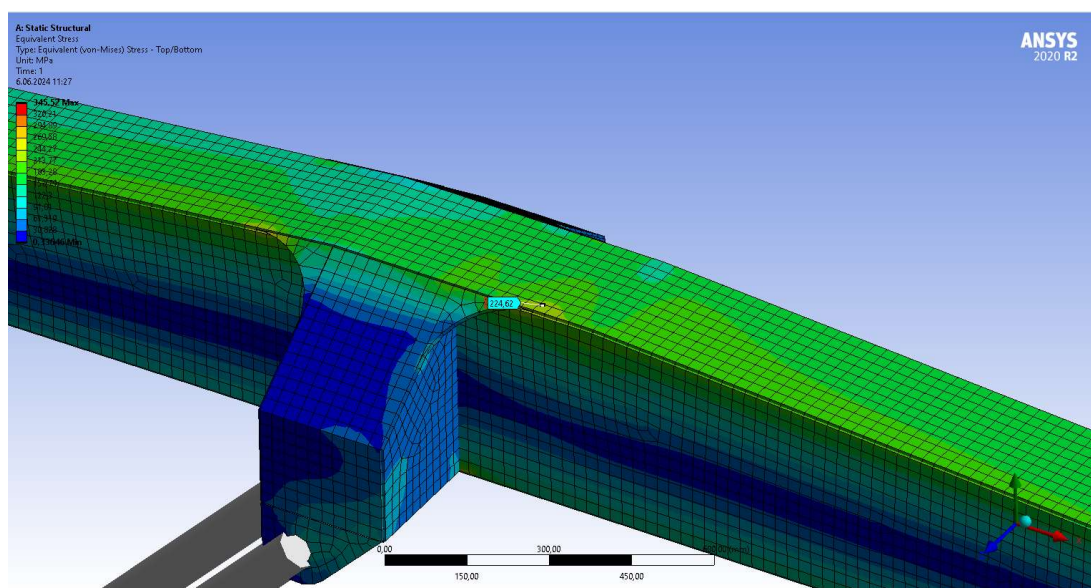


Figure 3.35: Boom critical stress point 224N/mm²

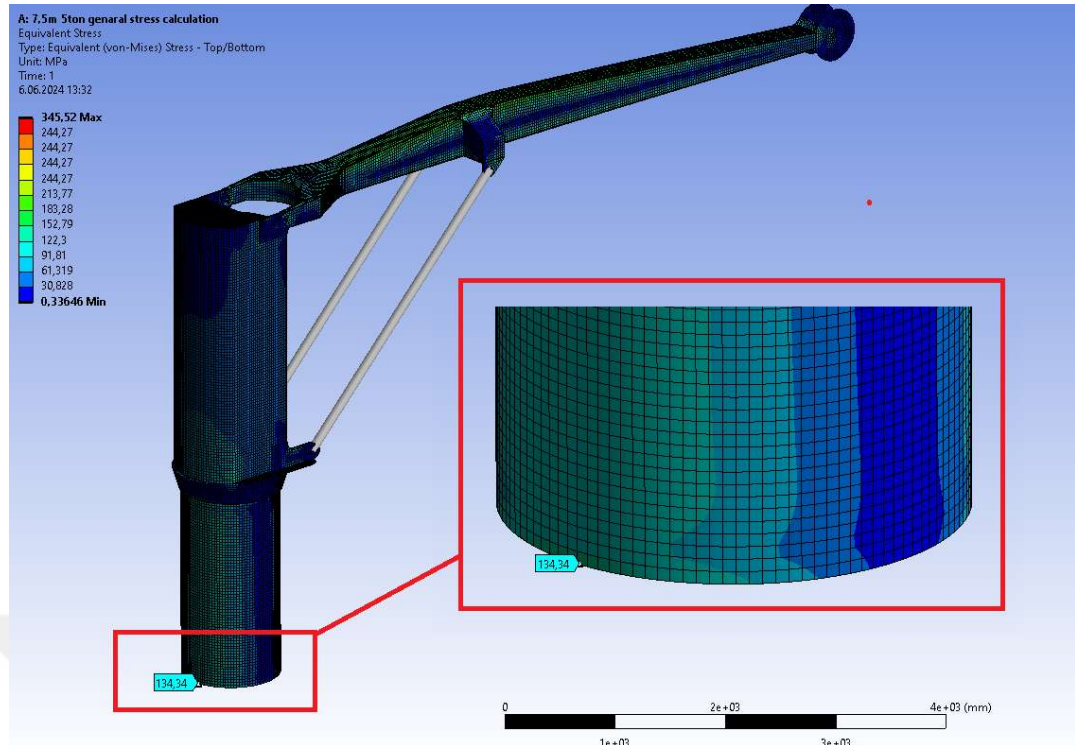


Figure 3.36: Pedestal critical stress point 134,34N/mm²

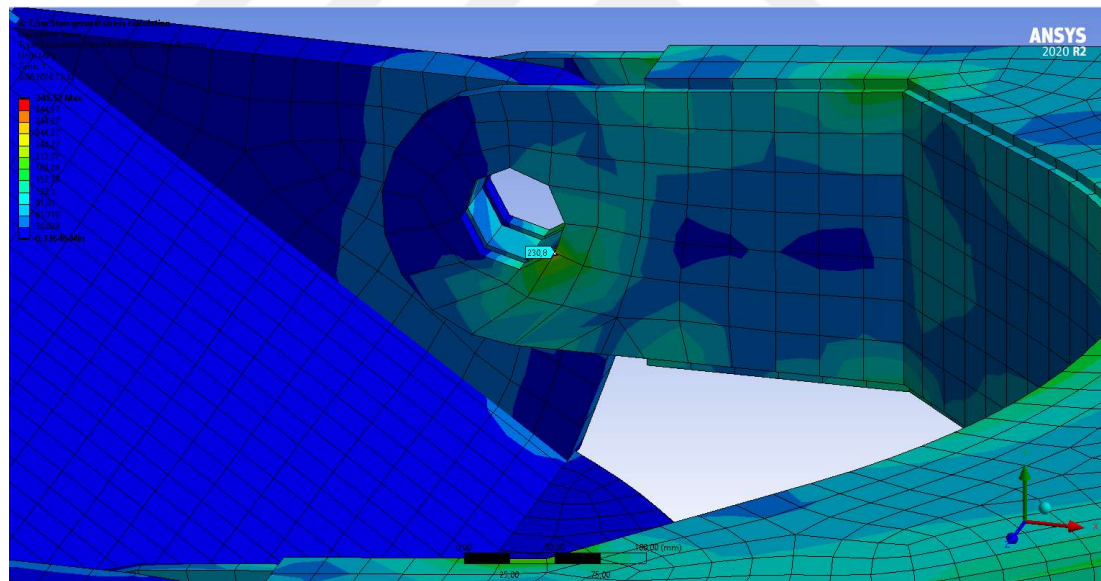


Figure 3.37: Critical stress point 3 lifting lug 230.8N/mm²

As can be seen in **Table 3.14** below, the deviation between the results we obtained in the program where we made the calculations at the critical points of the crane and the results in the FEA calculations is minimal. TL Rules accept 10% stress changes in FEA calculations. The maximum value difference we found is around 2%. For this reason, the program results we have designed are acceptable for the preliminary designs of hydraulic bottom cylinder cranes [9].

Since our main purpose in these calculations is to determine the spring coefficients of the cranes, the force-displacement relationship in the hook's law region, where the elastic behavior of the steel shows, will be examined in the FEA calculations for the next cranes.

Table 3.14: Stress Deviation Between Calculation Software and FEA Result

Deviation Calculation Between FEA and Crane Calculation Excel		σ_{boom} N/mm ²	$\sigma_{pedestal}$ N/mm ²	$\sigma_{stress\ point\ 3}$ boom N/mm ²
Load Radius 7,5m SWL 5 ton	Stress Results according to crane program	221,4	133	236,5
Load Radius 7,5m SWL 5 ton	Stress Results according to FEA calculation	224	134,34	230,8
	Deviation	1%	1%	-2%

The displacements in all cranes in **Figures 3.38, to 3.54** below are the displacements obtained from 5 tons of vertical force.

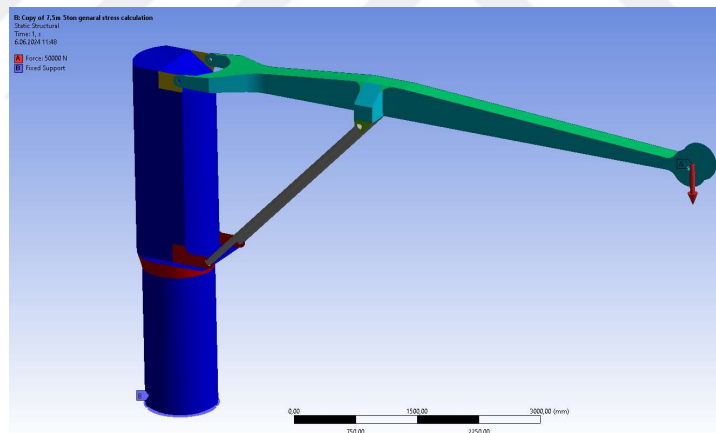


Figure 3.38: Spring coefficient of the 7.5m-5-ton capacity crane

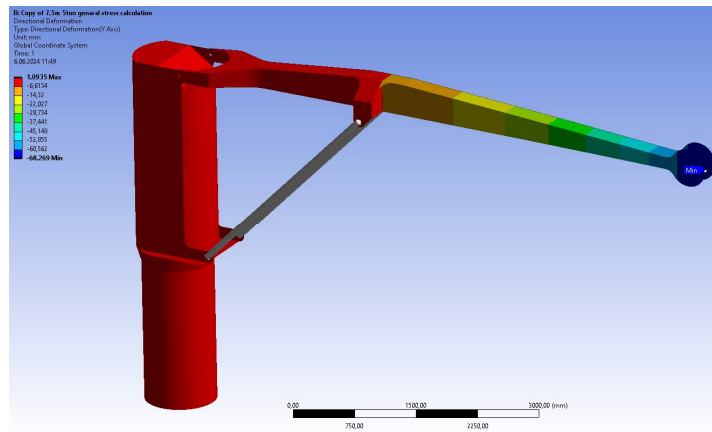


Figure 3.39: Vertical displacement in 7.5m 5-ton capacity crane

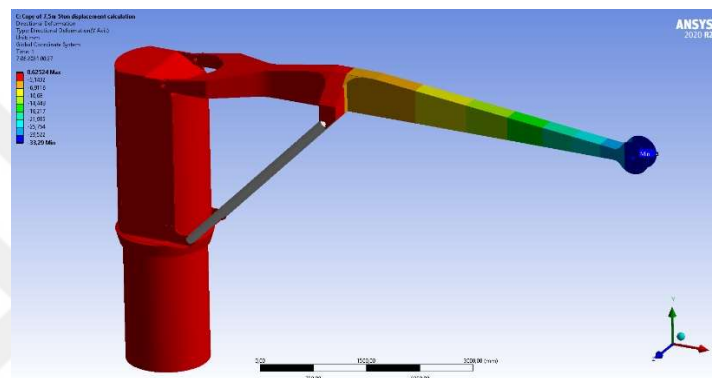


Figure 3.40: Vertical displacement in 7.5m 10-ton capacity crane

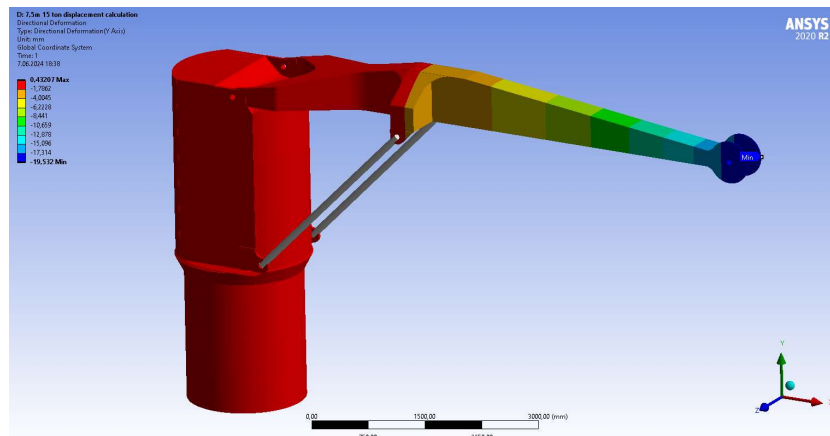


Figure 3.41: Vertical displacement in 7.5m 15-ton capacity crane

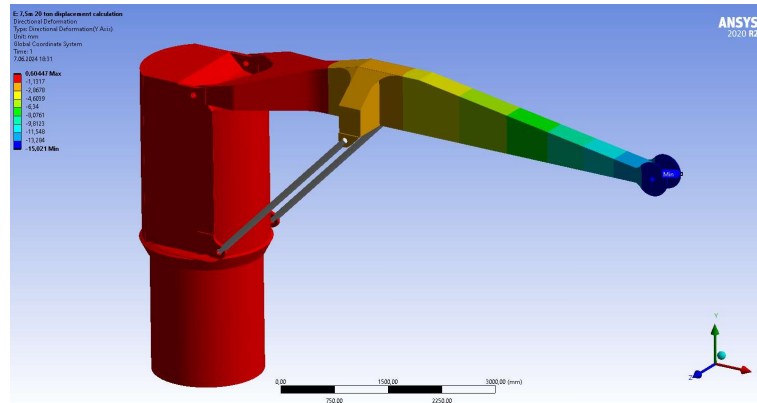


Figure 3.42: Vertical displacement in 7.5m 20-ton capacity crane

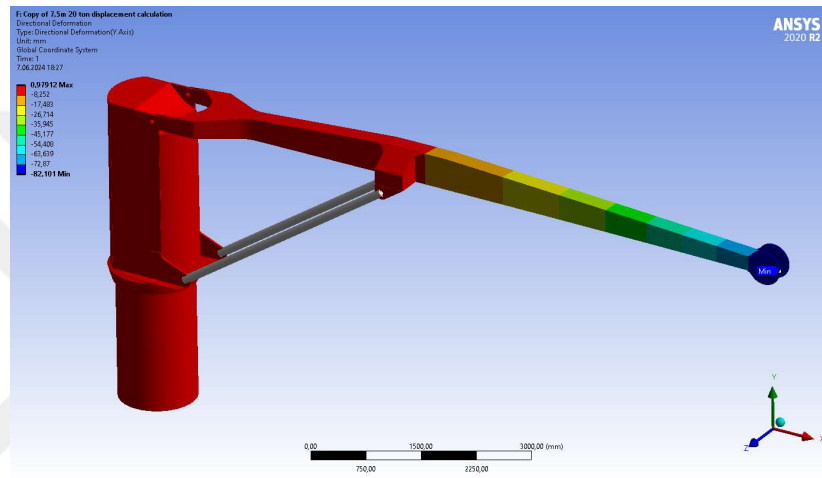


Figure 3.43: Vertical displacement in 10m 5-ton capacity crane

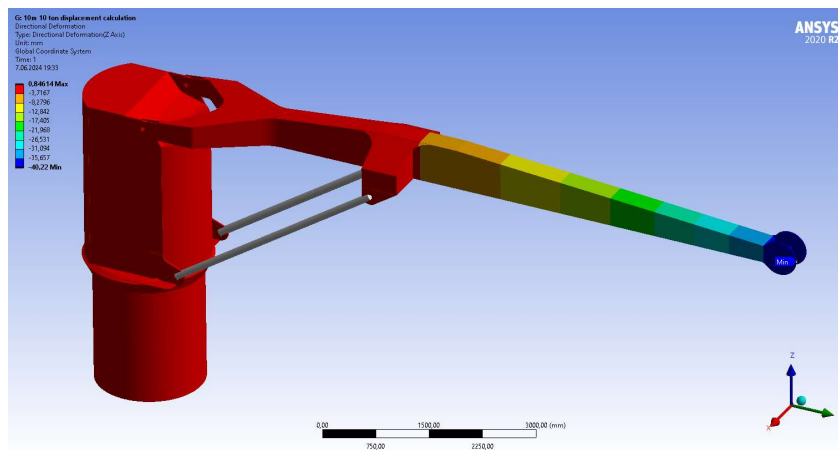


Figure 3.44: Vertical displacement in 10m 10-ton capacity crane

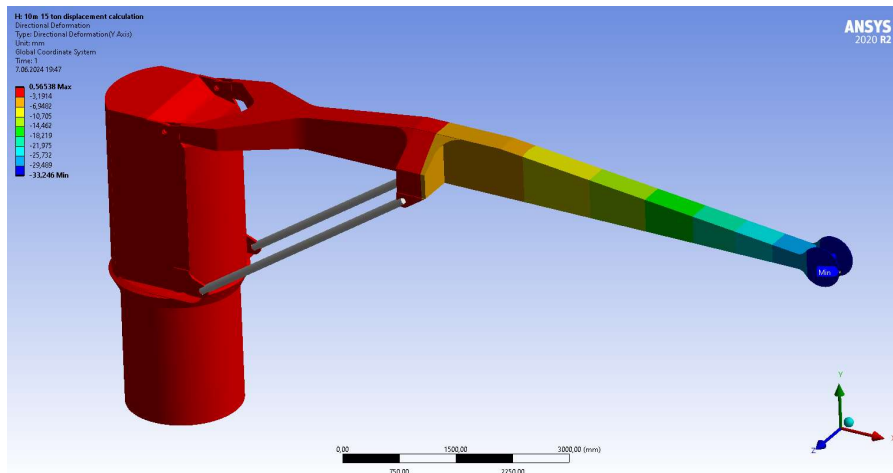
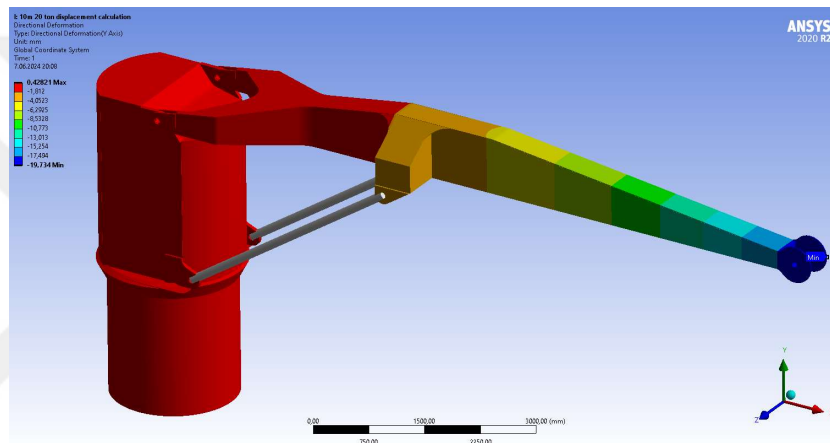


Figure 3.45: Vertical displacement in 10m 15-ton capacity crane



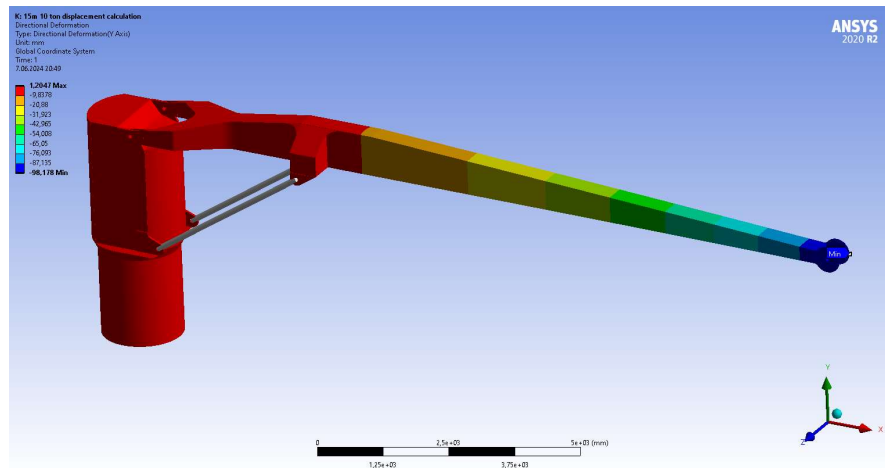


Figure 3.48: Vertical displacement in 15m 10-ton capacity crane

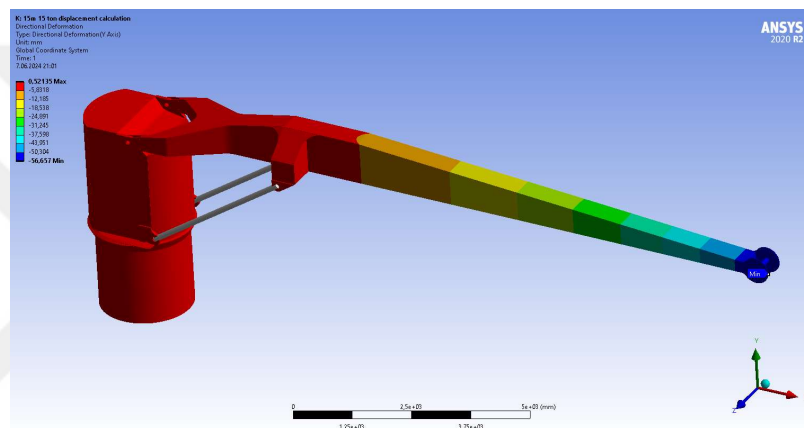


Figure 3.49: Vertical displacement in 15m 15-ton capacity crane

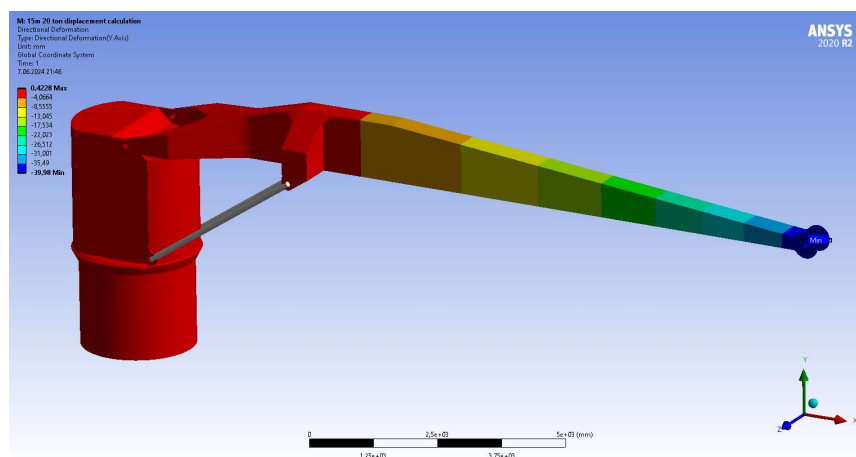


Figure 3.50: Vertical displacement in 15m 20-ton capacity crane

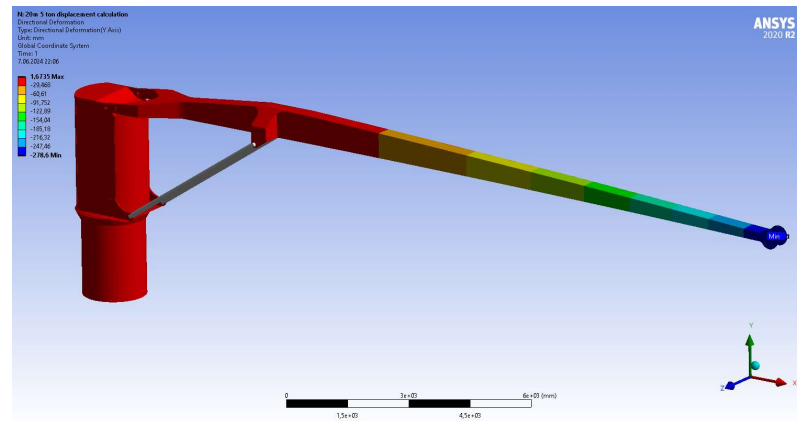


Figure3.51: Vertical displacement in 20m 5-ton capacity crane

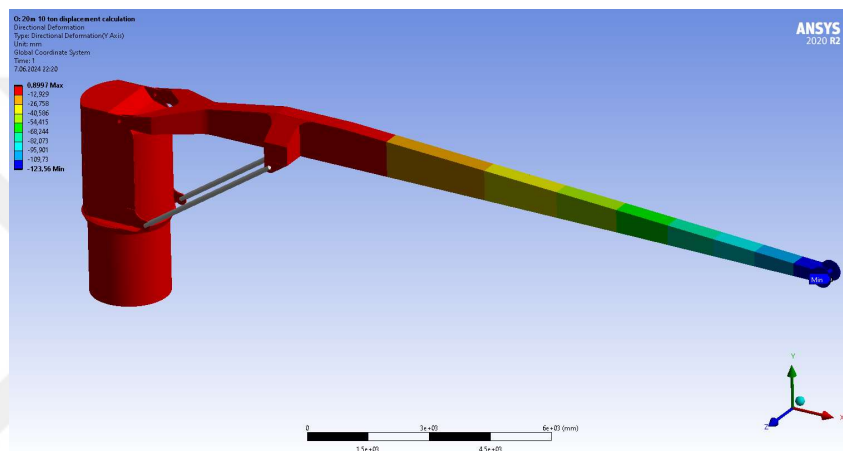


Figure 3.52: Vertical displacement in 20m 10-ton capacity crane

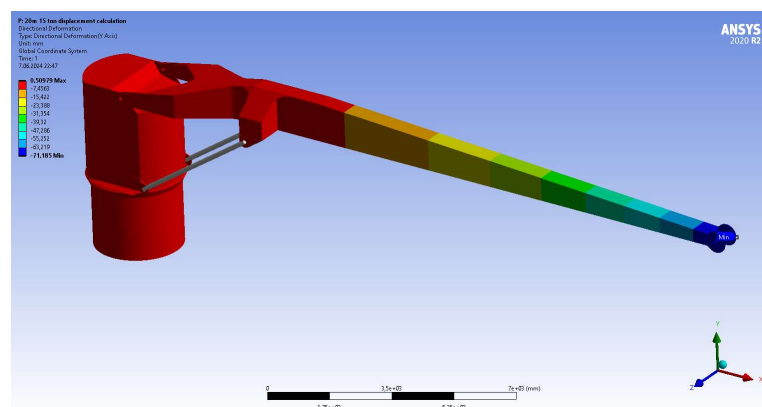


Figure 3.53: Vertical displacement in 20m 15-ton capacity

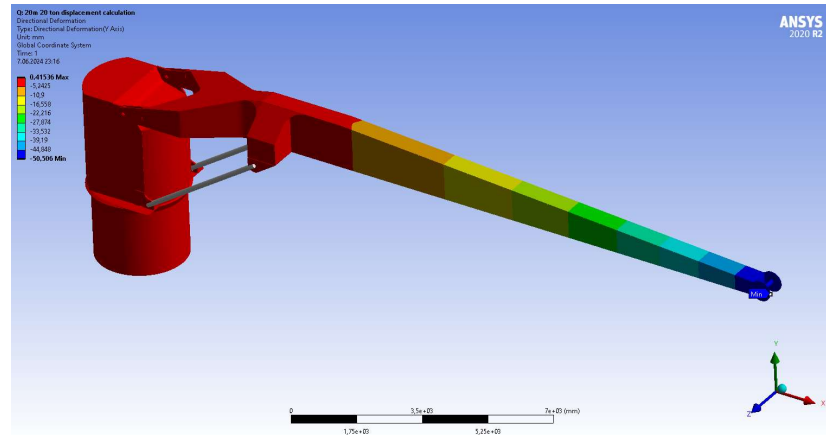


Figure3.54: Vertical displacement in 20m 20-ton capacity crane

Our work on the pages above aims to find the structural spring coefficients of typical cranes built according to classification society rules. TL Rules (Section 4 C 3.2.6) expects the spring constants affected by the ropes to be included in the calculation, taking as reference the situation where the hook is 6 m above sea level. The evaluation will be made assuming the Young's-modulus of the ropes is 7500N/cm^2 [9].

If it is assumed that the crane's rope winding drum is at the center of the crane's slewing ring, that is, the lifting radius of the crane will go as a single wire from the rope drum to the pulley at the end of the boom, and then, if there is rigging in the hook block, the rope will become multi-section. It is assumed that the tip of the crane boom is 11m above sea level. Therefore, the Hook hangs 5m below the boom tip. In **Figure 3.55** the spring constant of the crane is shown.

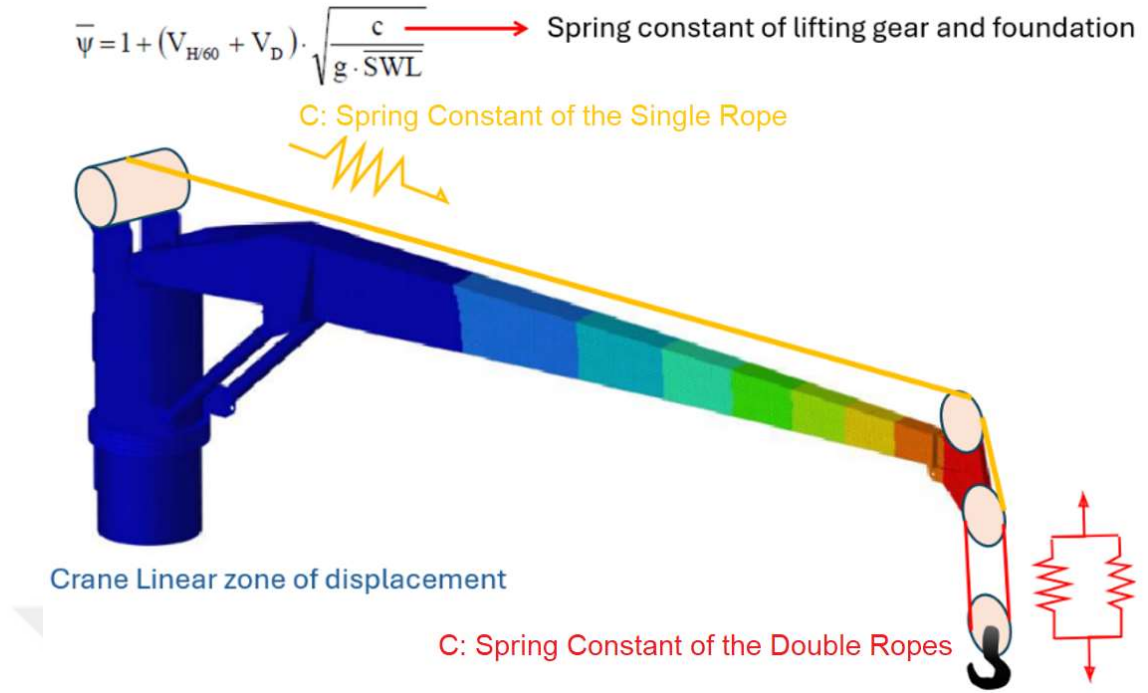


Figure 3.55: Displacement and tension figure

$$\frac{1}{c} = \frac{1}{c_{\text{Crane Structure}}} + \frac{1}{c_{\text{above the boom}}} + \frac{1}{c_{\text{Double Rope}}} \quad (3.142)$$

In the FEA studies above, 50000N was specifically applied to all cranes. The reason for this is to determine the spring coefficients in a practical way for all cranes.

$$c_{\text{Crane Structure}} = F / \text{displacement}_z \quad (3.143)$$

$$c_{\text{above the boom}} = \frac{d_{\text{Rope}}^2 \cdot \pi \cdot \text{Young's-modulus}}{c \cdot 4} \quad (3.144)$$

$$c_{\text{between boom tip and hook}} = \frac{d_{\text{Rope}}^2 \cdot \pi \cdot N_{\text{hanging rope}} \cdot \text{Young's-modulus}}{5 \text{ meters} \cdot 4} \quad (3.145)$$

The program has been prepared to find the combined spring coefficients. Below you can see the spring coefficient calculation obtained for 16 cranes. In **Table 3.15** Spring Coefficients of the Cranes are shown.

Table 3.15: Spring Coefficient of the Cranes

Spring Coefficient Crane Ropes (Drum to Hook)											
	C (mm)	Ø Rope Diameter (mm)	Rope's Youngs Modulus N/mm²	Hook hangs (mm)	Number of the hanging rope pieces	Crane Vertical Displacement According to FEA Result (mm)	Spring coefficient of the crane (N/mm)	Spring coefficient of the rope above the boom (N/mm)	Rope spring coefficient between boom tip and hook (N/mm)	Combine Spring Coefficient (N/mm)	
	Load Radius 7,5m SWL 5 ton	7500	22	75000	5000	1	68,3	732	3801	5702	554
	Load Radius 7,5m SWL 10 ton	7500	22	75000	5000	2	33,3	1502	3801	11404	984
	Load Radius 7,5m SWL 15 ton	7500	26	75000	5000	2	19,5	2560	5309	15928	1558
	Load Radius 7,5m SWL 20 ton	7500	22	75000	5000	4	15,0	3329	3801	22808	1647
	Load Radius 10m SWL 5 ton	10000	22	75000	5000	1	82,1	609	2851	5702	461
	Load Radius 10m SWL 10 ton	10000	22	75000	5000	2	40,2	1243	2851	11404	805
	Load Radius 10m SWL 15 ton	10000	26	75000	5000	2	33,2	1504	3982	15928	1022
	Load Radius 10m SWL 20 ton	10000	22	75000	5000	4	19,7	2534	2851	22808	1267
	Load Radius 15m SWL 5 ton	15000	22	75000	5000	1	179,2	279	1901	5702	233
	Load Radius 15m SWL 10 ton	15000	22	75000	5000	2	98,2	509	1901	11404	388
	Load Radius 15m SWL 15 ton	15000	26	75000	5000	2	56,7	882	2655	15928	636
	Load Radius 15m SWL 20 ton	15000	22	75000	5000	4	40,0	1251	1901	22808	730
	Load Radius 20m SWL 5 ton	20000	22	75000	5000	1	278,6	179	1425	5702	155
	Load Radius 20m SWL 10 ton	20000	22	75000	5000	2	123,6	405	1425	11404	307
	Load Radius 20m SWL 15 ton	20000	26	75000	5000	2	71,2	702	1991	15928	503
	Load Radius 20m SWL 20 ton	20000	22	75000	5000	4	50,5	990	1425	22808	570

In the 3D graphic below, the table regarding the crane length, lifting capacity, and stiffness of the crane can be seen. This table includes the stiffness of the crane but does not include the stiffness of the ropes. In **Figure 3.56** 3D Graph of the crane spring coefficient according to load radius and SWL values is shown.

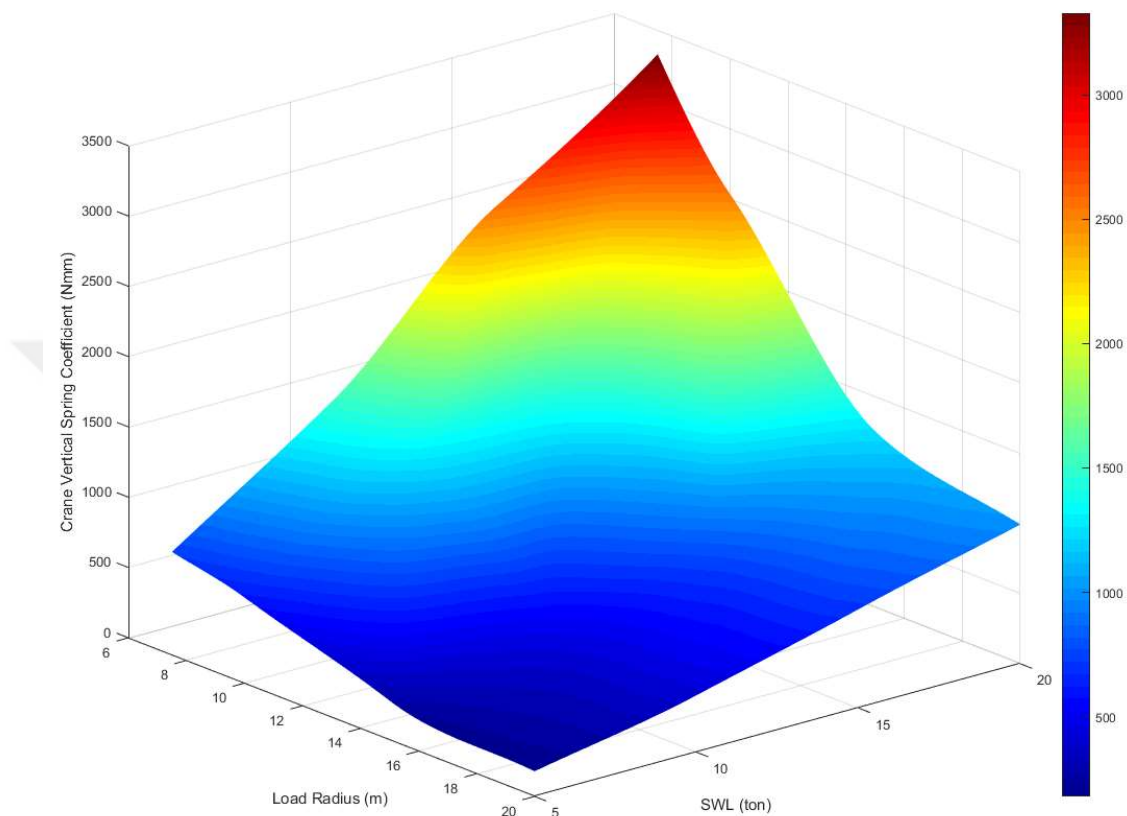


Figure 3.56: 3D graph of crane structure spring

In the 3D graphic below, the table regarding the crane length, lifting capacity and stiffness of the crane can be seen. This table includes both the stiffness of the crane and the stiffness of the ropes. In **Figure 3.57** 3D Graph of the crane spring coefficient according to load radius and SWL values, including the stiffness of the rope values, is shown.

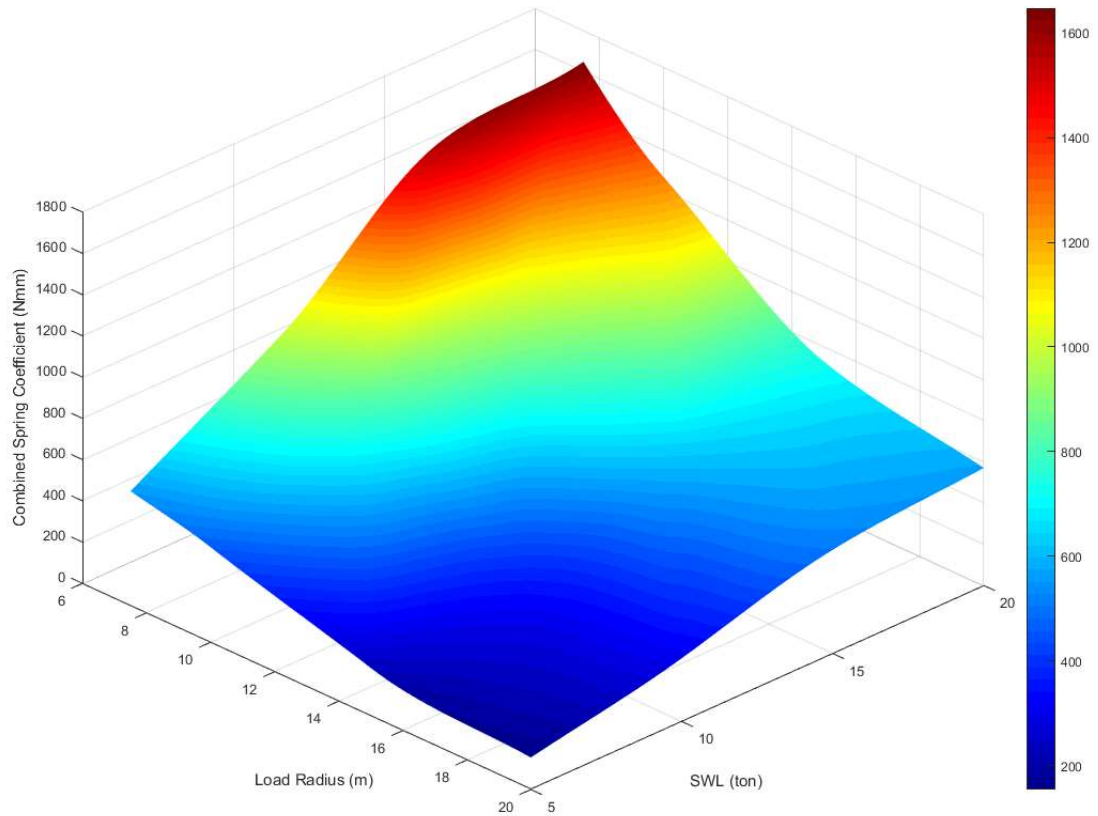


Figure 3.57: 3D graph of crane spring coefficients (include rope stiffness)

3.4. Determining the Spring Coefficient of the Lifting Ropes Used Between the Shock Absorber and the SWL

We will determine the ropes used in this region using a different method. Standard and simple tables are prepared to prevent special engineering calculations from being made each time when lifting loads and to prevent riggers performing operations from being put at risk. This subject is covered in the EN 13414-1 standard. We will determine the diameters of these ropes according to this standard and then find the spring coefficients.

As seen in the figure below, we will use the 4-arm sling with a beta angle of up to 45 degrees as a reference. In this case, the K_L coefficient is 2.1 according to EN 13414-1. Additionally, ferruled secured hard eye ends of the ropes were preferred. For this reason, the K_T coefficient was taken as 0.9. Unless special engineering calculations are made, the safety coefficient of the ropes is never reduced below 5, therefore the Z_P

coefficient is taken as 5. As additional information, since there will be a plane at 3 points and millimetric errors in the length of the 4 ropes, it is aimed to further reduce the risks by evaluating the standard 3 ropes as carrying the load.[42].

$$K_T = 0.90 \quad (3.146)$$

$$K_L = 2.1 \quad (3.147)$$

$$Z_P = 5 \quad (3.148)$$

$$g = 9.81\text{m/s}^2$$

$$\text{Working Load Limit} = \frac{F_{\text{rope breaking load}} K_T K_L}{Z_p g} \quad (3.148)$$

$F_{\text{rope breaking load}}$ for SWL 5 ton = 129.76 kN minimum requirement
 Ø16 rope size steel core rope 6x36 class Breaking load 161.3 kN > 129.76 ok

$F_{\text{rope breaking load}}$ for SWL 10 ton = 259.52 kN minimum requirement
 Ø22 rope size steel core rope 6x36 class 305 kN > 259.52 ok

$F_{\text{rope breaking load}}$ for SWL 15 ton = 389.28 kN minimum requirement
 Ø26 rope size steel core rope 6x36 class 426 kN > 389.28 ok

$F_{\text{rope breaking load}}$ for SWL 20 ton = 519.04 kN minimum requirement
 Ø32 rope size steel core rope 6x36 class 645.2 kN > 519.04 ok

The design criteria and dimensions of the lifting rope system are defined in **Figure 3.58** below. Spring coefficient of the lifting ropes are defined in **Table 3.16**. The Graph between SWL and Spring coefficient of the General Lifting Rope is described **Table 3.17**

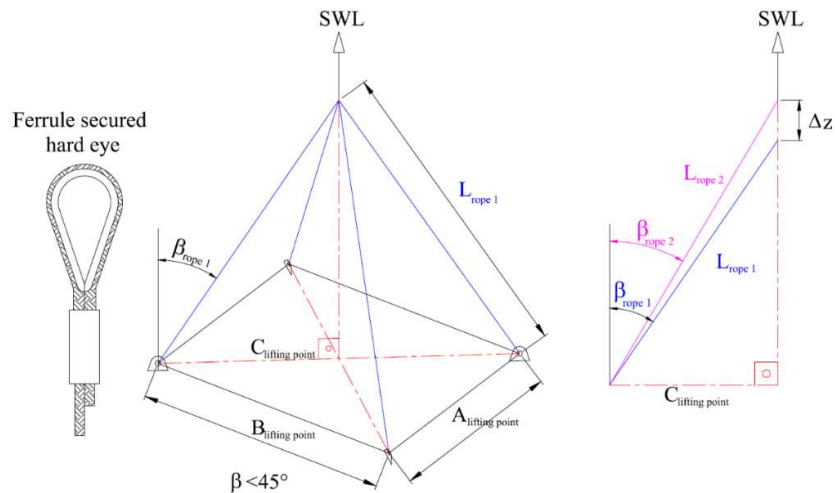


Figure 3.58: Lifting Rope System and Design Dimensions.

$$F_{\text{Rope } 1} = 0 \text{ N} \quad (3.149)$$

$$F_{\text{Rope } 2} \cdot \cos(\beta_{\text{Rope } 2}) = \frac{\text{SWL}}{4} \quad (3.150)$$

$$L_{\text{Rope } 2} = L_{\text{Rope } 1} + F_{\text{Rope } 2} \cdot k_{\text{Rope}} \quad (3.151)$$

$$k_{\text{single rope}} = \frac{d_{\text{Rope}}^2 \cdot \pi \cdot \text{Youn 's-modulu}}{L_{\text{Rope } 1} \cdot 4} \quad (3.152)$$

$$2 \cdot L_{\text{Rope } 1} \cdot \sin(\beta_{\text{Rope } 1}) = \sqrt{A_{\text{working load}}^2 + B_{\text{working load}}^2} \quad (3.153)$$

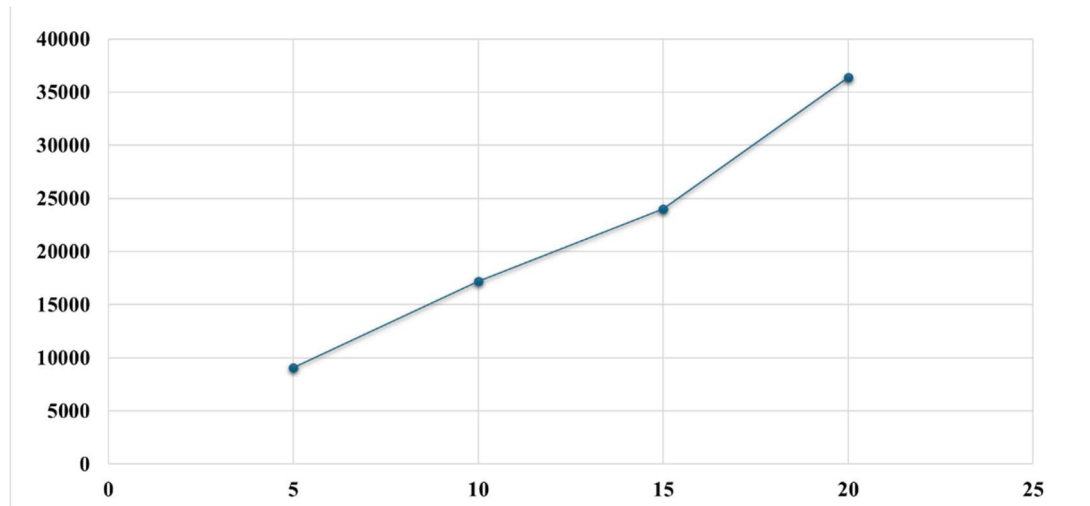
$$k_R = \frac{\text{SWL}}{L_{\text{Rope } 2} \cdot \cos(\beta_{\text{Rope } 2}) - L_{\text{Rope } 1} \cdot \cos(\beta_{\text{Rope } 1})} \quad (3.154)$$

$$k_R = \frac{\text{SWL} \cdot g}{\Delta z} \text{ (N/mm)} \quad (3.155)$$

Table 3.16: Spring coefficient of the lifting rope

Spring Coefficient Lifting Rope (Rope Design According to EN 13414-1)									
SWL (ton)	Rope Diameter (mm)	A lifting point (m)	B lifting point (m)	L rope1 (m)	k single rope (N/mm)	D L (mm)	b rope1	b rope2	k R (N/mm)
5	16	2	3	6	2513	5,396	17,485	17,470	9091
10	22	2	3	6	4752	5,707	17,485	17,469	17189
15	26	2	3	6	6637	6,129	17,485	17,468	24009
20	32	2	3	6	10053	5,396	17,485	17,470	36360

Table 3.17: The Graph Between SWL and Spring Coefficient



3.5. Calculation of the Shock Absorber

Passive Shock Absorbing Elements are elements used to absorb shocks when additional energy is needed, providing a spring feature with the pressure and volume relationship due to the compressibility of gases. Since the wave periods of the ship vary between 5.4 and 8.3 seconds in high wave situations according to TL lifting appliances rule, the movements are very slow and the gas is considered to exhibit adiabatic properties, ignoring heat exchange with the external environment [9], [34].

$$\frac{P.V^\kappa}{R.T} = \text{Constant} \quad \text{and} \quad \Delta V = V_1 \left[1 - \left(\frac{P_1}{P_2} \right)^{\frac{1}{\kappa}} \right] \quad (3.156)$$

The adiabatic gas constant of nitrogen can be chosen as 1.4 at low pressures, but these values differ at high pressures and temperatures. For a detailed analysis, the study is described in **Figure 3.59** below.

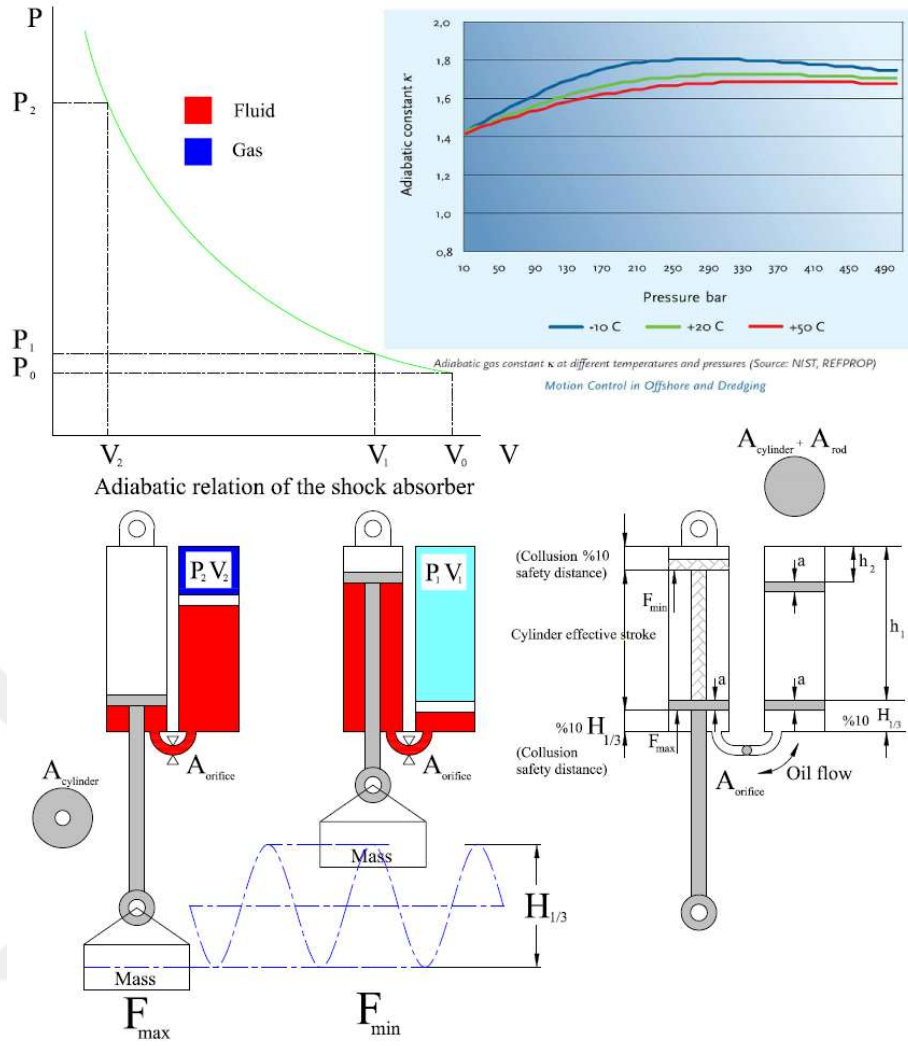


Figure 3.59: Shock absorber calculation parameters

$$P_{max} = \frac{F_{max}}{A_{cylinder}} \quad (3.157)$$

$$P_{max} = 150 \text{ Mpa (pre design value)} \quad (3.158)$$

The inner diameter of the cylinder is increased according to the standard inner diameter of the fabricated pipe

$$F_{max} = F_{swl z} \cdot 1.1 \quad (3.159)$$

Shock absorber height %10 more than significant wave height (protect the cylinder head collision to the bottom end)

$$\Delta V = A_{\text{Cylinder}} \cdot \text{stroke} \quad (3.160)$$

$$P_1 \cdot V_1^\kappa = P_2 \cdot V_2^\kappa \quad (3.161)$$

3.5.1. Calculation of Weight of the Shock Absorber

Calculating the weight of the shock absorber is close to the calculations of the hydraulic cylinder. In order to find the weight, it is necessary to know the diameter of the rod and therefore the stresses on the rod, the diameter of the pipe and the pipe thickness of the pipe. For pipe calculations, thickness calculations should be made with reference to the formulas from DNV according to the working pressure [33].

Shock absorber tube thickness calculation:

$$t = \frac{p D_0}{20 \sigma_t e + p} + \bar{c} \quad (3.162)$$

For the shock absorber rod thickness calculation: required structural safety stress 5 times for breaking strength Shock absorber rod material tensile strength:

$$R_{m \text{ (Ck45)}} = 580 \text{ N/mm}^2$$

$$580 / 5 \text{ (safety factor)} = 116 \text{ N/mm}^2 \text{ (design stress of the shock absorber rod)}$$

$$F_{\max} / A_{\text{rod}} = \max 116 \text{ N/mm}^2$$

$$F_{\min} = (\text{weight of the shock absorber} + \text{lifting ropes} + \text{shackles}) \times 1.2$$

$$F_{\min} = (m_s + \text{lifting ropes} + \text{schakels}) \cdot 1.2$$

$$m_s = \left[\frac{((D_0^2 + 2 t) - D_0^2) \pi \text{ stroke}}{2} + A_{\text{rod}} \text{ stroke} \right] \rho_{\text{steel}} 1.2 \quad (3.163)$$

$$P_{\min} = \frac{F_{\min}}{A_{\text{Cylinder}}} \quad (3.164)$$

$$\Delta V = V_1 \left[1 - \left(\frac{P_1}{P_2} \right)^{\frac{1}{\kappa}} \right] \quad (3.165)$$

$$V_1 = \frac{\Delta V}{1 - \left(\frac{P_1}{P_2} \right)^{\frac{1}{\kappa}}} \quad (3.166)$$

$$Tube\ diameter = \sqrt{\frac{4 \cdot (A_{rod} + A_{cylinder})}{\pi}} \quad (3.167)$$

Shock Absorber Stiffness Calculation

$$k_s = \frac{F_{max} - F_{min}}{\text{stroke (according to significant wave height)}} \quad (3.168)$$

The following **Table 3.18** show the input and output tables of the program that calculates the shock absorber cylinder.

Table 3.18: Shock Absorber Calculation

Shock Absorber Calculation ks calculation (absorber stroke 3m)									
	F_{swl} N	Pmax (bar) for Pre- design	Shock Absorber Effective Stroke (m)	Rod Diameter (mm)	A cylinder mm² for pre- design	Diameter of orifice (mm)	Tube Inside Diameter (mm)	Pmax (bar)	Tube Thickness
Load Radius 7,5m SWL 5 ton	73719	150	3	30	4914,6	30	90	130,4	5
Load Radius 7,5m SWL 10 ton	147437	150	3	50	9829,2	30	130	130,4	6
Load Radius 7,5m SWL 15 ton	221156	150	3	50	14743,7	30	150	140,8	7
Load Radius 7,5m SWL 20 ton	294875	150	3	60	19658,3	30	170	148,4	8
Load Radius 10m SWL 5 ton	73719	150	3	30	4914,6	30	90	130,4	5
Load Radius 10m SWL 10 ton	147437	150	3	50	9829,2	30	130	130,4	6
Load Radius 10m SWL 15 ton	221156	150	3	50	14743,7	30	150	140,8	7
Load Radius 10m SWL 20 ton	294875	150	3	60	19658,3	30	170	148,4	8
Load Radius 15m SWL 5 ton	73719	150	3	30	4914,6	30	90	130,4	5
Load Radius 15m SWL 10 ton	147437	150	3	50	9829,2	30	130	130,4	6
Load Radius 15m SWL 15 ton	221156	150	3	50	14743,7	30	150	140,8	7
Load Radius 15m SWL 20 ton	294875	150	3	60	19658,3	30	170	148,4	8
Load Radius 20m SWL 5 ton	73719	150	3	30	4914,6	30	90	130,4	5
Load Radius 20m SWL 10 ton	147437	150	3	50	9829,2	30	130	130,4	6
Load Radius 20m SWL 15 ton	221156	150	3	50	14743,7	30	150	140,8	7
Load Radius 20m SWL 20 ton	294875	150	3	60	19658,3	30	170	148,4	8

	D V (liter)	(Ms)Mass Shock Absorber (kg)	F_min (N)	Pmin (bar)	V1 (l)	V2 (l)	k (70 bar)	ks stiffness of shock absorber (N/mm)
Load Radius 7,5m SWL 5 ton	17,0	86,9	1234,1	2,2	17,0	0,0	1,6	24,2
Load Radius 7,5m SWL 10 ton	33,9	167,0	2176,5	1,9	34,0	0,0	1,6	48,4
Load Radius 7,5m SWL 15 ton	47,1	208,9	2753,1	1,8	47,2	0,1	1,6	72,8
Load Radius 7,5m SWL 20 ton	59,6	277,3	3475,1	1,7	59,7	0,1	1,6	97,1
Load Radius 10m SWL 5 ton	17,0	86,9	1234,1	2,2	17,0	0,0	1,6	24,2
Load Radius 10m SWL 10 ton	33,9	167,0	2176,5	1,9	34,0	0,0	1,6	48,4
Load Radius 10m SWL 15 ton	47,1	208,9	2753,1	1,8	47,2	0,1	1,6	72,8
Load Radius 10m SWL 20 ton	59,6	277,3	3475,1	1,7	59,7	0,1	1,6	97,1
Load Radius 15m SWL 5 ton	17,0	86,9	1234,1	2,2	17,0	0,0	1,6	24,2
Load Radius 15m SWL 10 ton	33,9	167,0	2176,5	1,9	34,0	0,0	1,6	48,4
Load Radius 15m SWL 15 ton	47,1	208,9	2753,1	1,8	47,2	0,1	1,6	72,8
Load Radius 15m SWL 20 ton	59,6	277,3	3475,1	1,7	59,7	0,1	1,6	97,1
Load Radius 20m SWL 5 ton	17,0	86,9	1234,1	2,2	17,0	0,0	1,6	24,2
Load Radius 20m SWL 10 ton	33,9	167,0	2176,5	1,9	34,0	0,0	1,6	48,4
Load Radius 20m SWL 15 ton	47,1	208,9	2753,1	1,8	47,2	0,1	1,6	72,8
Load Radius 20m SWL 20 ton	59,6	277,3	3475,1	1,7	59,7	0,1	1,6	97,1

As stated in Peter Albers's book Motion Control in Offshore and Dredging. Due to the huge diameters of the cylinders, pipes, and valves in passive lifting systems, the damping coefficient is not very large. This coefficient can be taken as 5800N.s/m [34].



4. ROUGH SEA SWL REDUCTION CALCULATIONS

This part of the study examined the load reduction of offshore cranes in rough seas. While examining this, the results and differences of the situations with and without shock absorbers in different sea states were evaluated by taking the formulas given by the class rules as reference. In addition, the effect of this load reduction was considered with a different methodology, and the equations obtained from the dynamic system model were compared with the matrix solutions prepared in the MATLAB program and the simulations obtained in the Working Model 2D program.

4.1. Rough Sea SWL Reduction According to Classification Society Rules

The lifting capacity in calm sea conditions is first calculated when designing a crane according to the classification rules. Then the SWL is reduced according to the effects of the rough sea conditions. here are two main formulas for these calculations, which we previously mentioned in Chapter 3 (Equations (3.1) and (3.2)).

4.1.1. Case without shock absorber

In this calculation, the calculation has been made considering the situation where the shock absorber is not included in the system. The spring coefficient of the crane and the spring coefficients of the ropes will affect the spring coefficient c [9].

In these circumstances. Below, we can see the Sample Calculation for Load Radius 7.5m SWL 5-ton Significant wave height 2 meters and 3 meters. System spring coefficient “ c ”

$$\frac{1}{c} = \frac{1}{k_C} + \frac{1}{k_R} \quad (4.1)$$

$$\frac{1}{c} = \frac{1}{554} + \frac{1}{9091}$$

$$c = 523 \text{ kN/m}$$

$$V_H \text{ (For } H_{1/3} \text{ 2m)} = 0.5 \text{ [m/s]}$$

$$V_H \text{ (For } H_{1/3} \text{ 3m)} = 0.63 \text{ [m/s]}$$

$$V_D \text{ (For } H_{1/3} \text{ 2m)} = 1.09 \text{ [m/s]}$$

$$V_D \text{ (For } H_{1/3} \text{ 3m)} = 1.53 \text{ [m/s]}$$

$$\Psi = 1.332 \text{ significant wave height 2m}$$

$$\Psi = 1.3663 \text{ significant wave height 3m}$$

The unknowns of the 2 formulas overlap each other. $\overline{SWL}$ and $\bar{\psi}$ values that give the common result by interpolation have been found.

$$\bar{\psi} = 1 + (0.5 + 1.09) \cdot \sqrt{\frac{523}{9.81 \cdot 14.78}} \text{ (significant wave height 2m according to equation 3.2)}$$

$$\bar{\psi} = 4.019$$

$$\overline{SWL} = SWL \cdot \frac{\psi}{\bar{\psi} \cdot f_e} \text{ (significant wave height 2m)}$$

$$14.78 \cdot 4.019 \cdot 1.1 = 1.332 \cdot 49.05 \text{ (significant wave height 2m according to equation 3.1)}$$

$$65.34 = 65.34 \text{ both of formulation found the same result}$$

$$\overline{SWL} = 14.78 \text{ kN (significant wave height 2m)}$$

$$\bar{\psi} = 1 + (0.63 + 1.53) \cdot \sqrt{\frac{523}{9.81 \cdot 10.31}} \text{ (significant wave height 3m according to equation 3.2)}$$

$$\bar{\psi} = 5.910$$

$$10.31 \cdot 5.910 \cdot 1.1 = 1.366 \cdot 49.05 \text{ (significant wave height 3m according to equation 3.1)}$$

$$67 = 67 \text{ both of formulation found the same result}$$

$$\overline{SWL} = 10.31 \text{ kN (significant wave height 3m)}$$

4.1.2. Case with Shock Absorber

It has reduced load coefficients compared to 3-meter and 2-meter-long shock absorbers, which are specially designed for each crane to be used in 16 different cranes and therefore have customized spring coefficients. The formulas of the study in this section have been constructed according to the reference of the class rules. If you pay attention, Class rules do not include the dumping coefficients of cranes and shock

absorbers in the calculations. The rules have developed formulations according to the worst-case scenario in unpredictable sea states of the crane, but they allow load reduction even if shock absorbers are used, leaving the responsibility to the approval body. On the other hand, when we look carefully to equation (3.2), the part of the formula in the square root is the natural frequency formula of a simple mass-spring system. Here we have found SWL reducing according to the basic formulas [9].

In the next part of our study, we carried out a detailed study on load reductions in the reference time-dependent model prepared by considering the wave movement and damping effects.

Below, we can see the Sample Calculation for Load Radius 7.5m SWL 5-ton Significant wave height 2 meters and 3 meters, and with shock absorber.

System spring coefficient “ c ”

$$\begin{aligned}\frac{1}{c} &= \frac{1}{k_C} + \frac{1}{k_R} + \frac{1}{k_S} \\ \frac{1}{c} &= \frac{1}{554} + \frac{1}{9091} + \frac{1}{24,2}\end{aligned}\tag{4.2}$$

$$c = 23 \text{ kN/m}$$

$$\overline{SWL} = 39.98 \text{ kN (significant wave height 3m with 3m stroke shock absorber)}$$

$$\overline{SWL} = 43.32 \text{ kN (significant wave height 2m with 3m stroke shock absorber)}$$

The calculations above are just an example to explain how the formulas work. 192 calculations were made to show the effect of all these cranes, sea states and shock absorbers of different capacities on the lifting capacity. Instead of writing repetitive calculations, the following graphs in **Figure.4.1**, **Figure 4.2**, **Figure 4.3** and **Figure4.4** were prepared to make the subject more understandable.

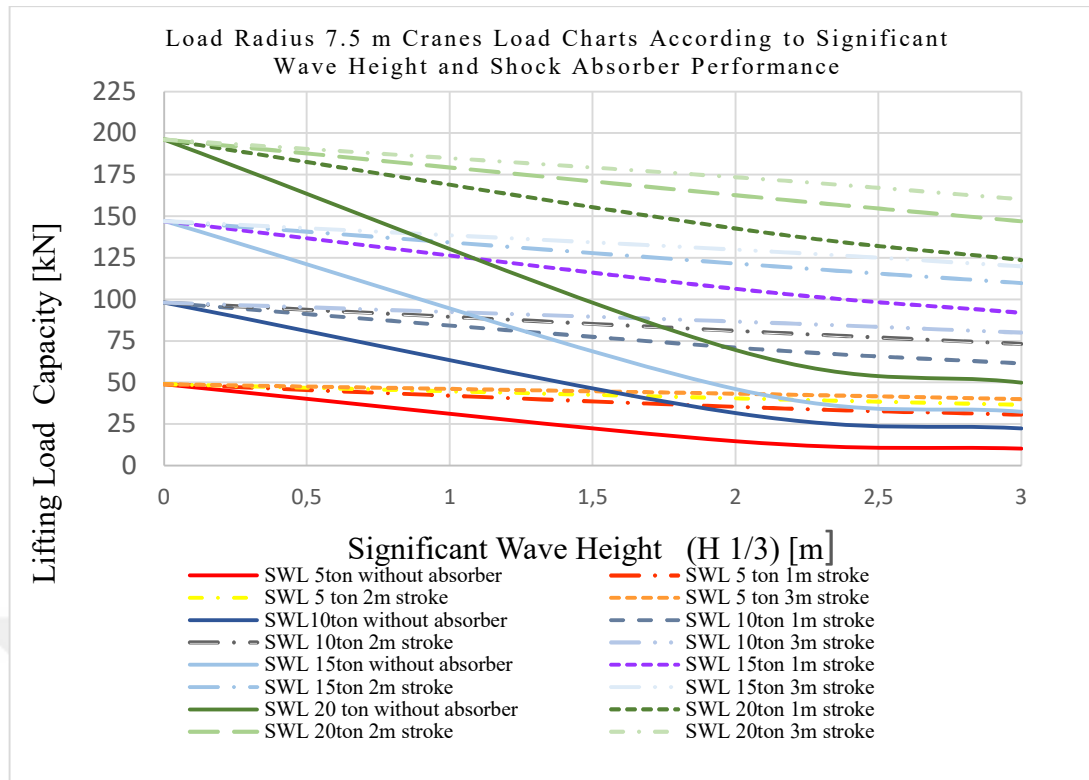


Figure 4.1: Load Radius 7.5m Cranes Load Charts

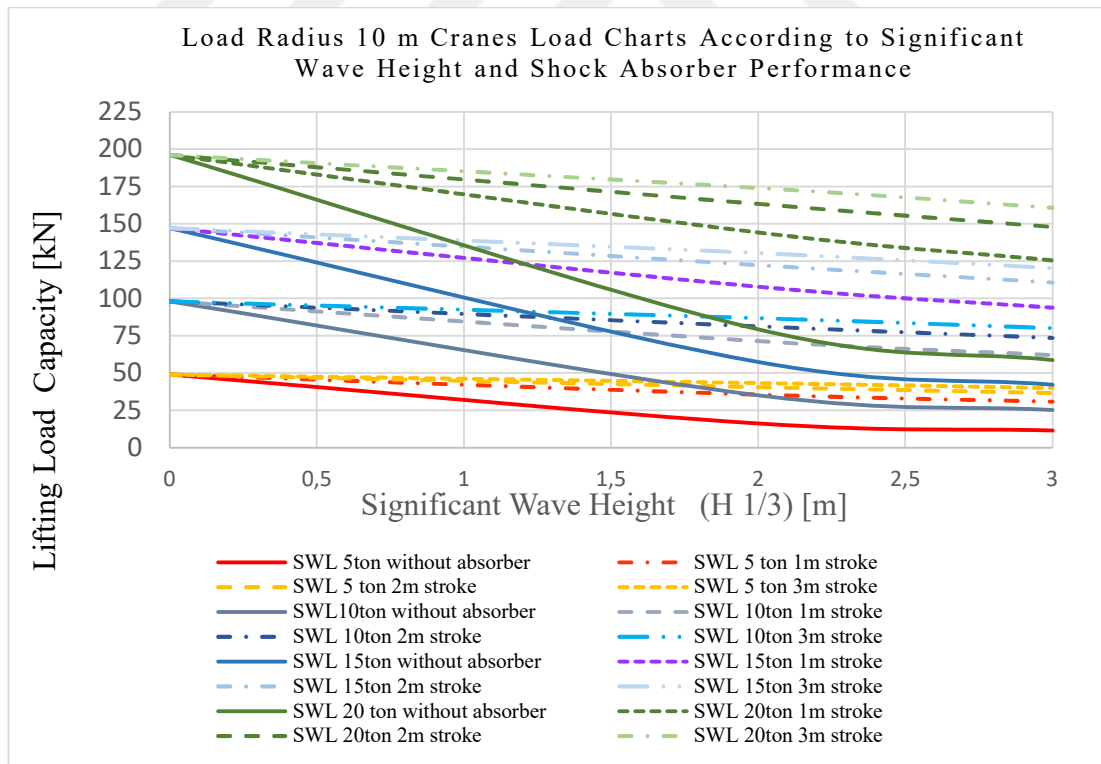


Figure 4.2: Load Radius 10m Cranes Load Charts

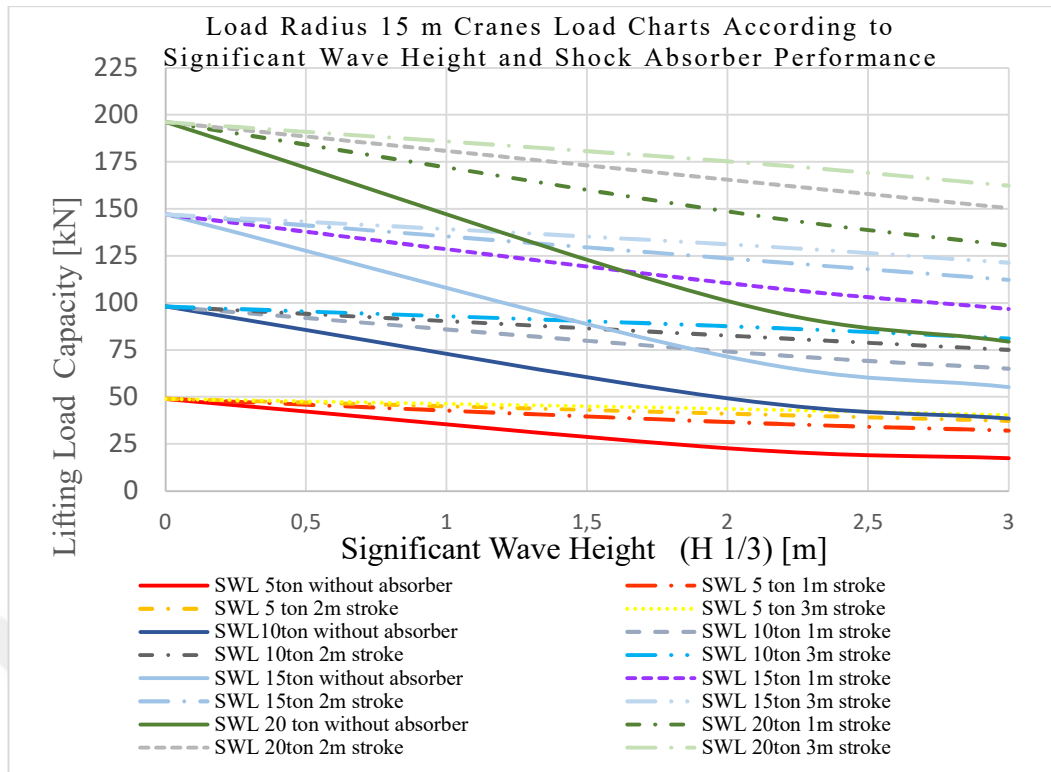


Figure 4.3: Load Radius 15m Cranes Load Charts

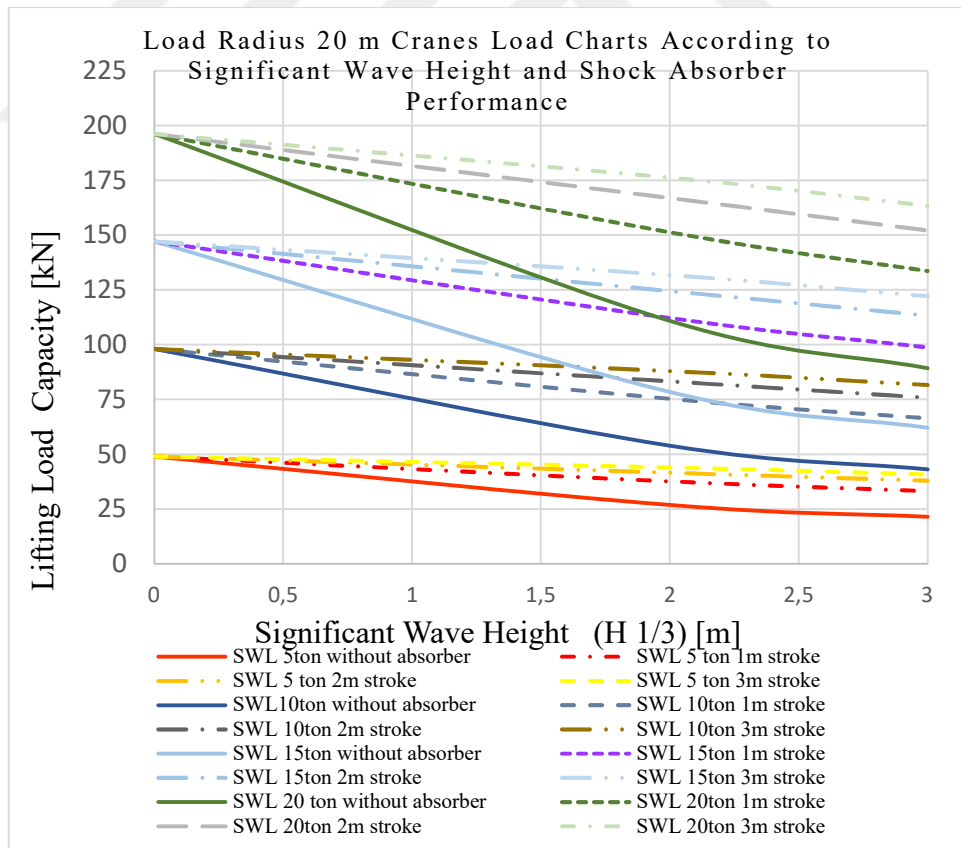


Figure 4.4: Load Radius 20m Cranes Load Charts

4.1.3. Rough Sea SWL Reduction According to Simplified Model by Using MATLAB

The analysis was performed using the sine curve in the MATLAB program. Phase was performed to prevent sudden and unrealistic force increases, especially at the beginning of the simulation. You can see the main input values of the equation in the table below. Calculations for wave heights of 2 meters and 3 meters for all cranes from 7.5 m to 20 m and from 5 tons to 20 tons. has been made. The load on the crane was found from the load on the spring element, which indicates the k_c crane elasticity, and reduced load values were determined according to the allowable stress values.

Below you can see the MATLAB input and matrix values for the crane, which will operate in 3m waves with a 7.5 m lifting radius, 5 tons lifting capacity, 3 meters of shock absorber.

To understand the accuracy of the analysis more clearly, special codes were written in MATLAB and a 2D system simulation was prepared. Detailed MATLAB codes are included in the Appendix section. Input and output values are shown in **Figure 4.4** and **Figure 4.5** below.

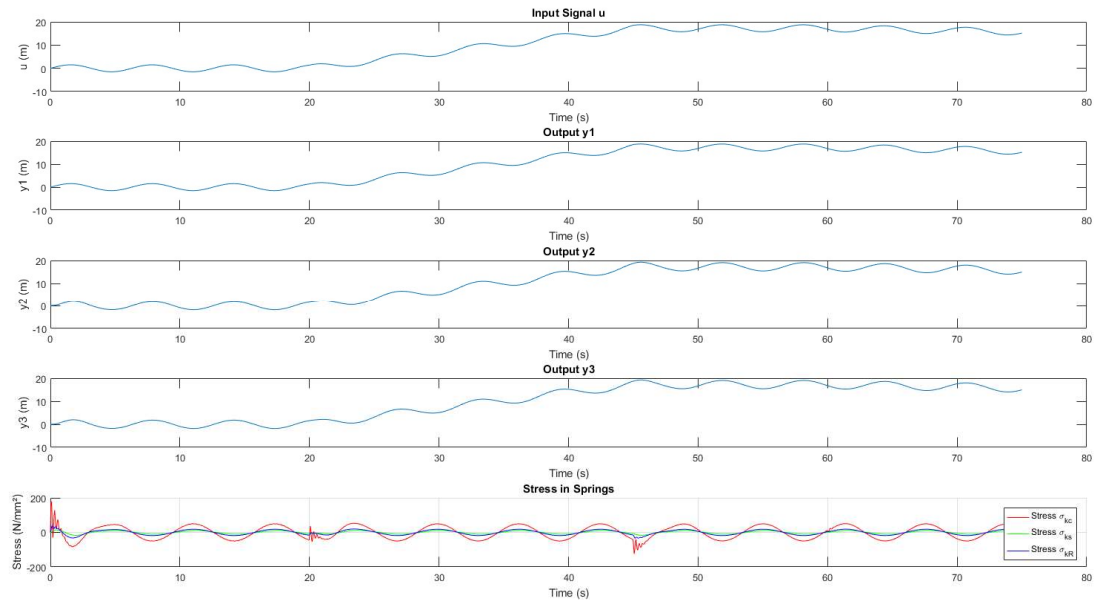


Figure 4.5: Load Radius 7.5 m Cranes Load Charts MATLAB Simulation

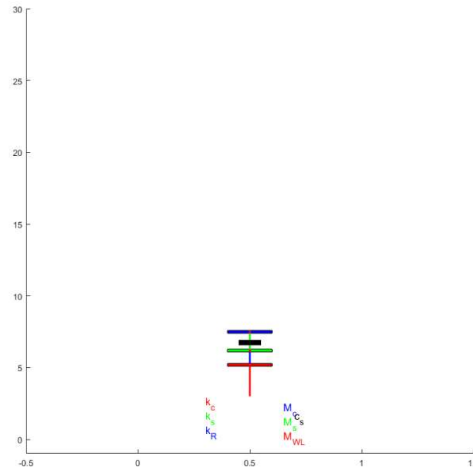


Figure 4.6: 7.5 m, Crane MATLAB model

For a lifting radius of 7.5 m, a lifting capacity of 5 tons, and a 3-meter shock absorber, the maximum load achieved according to the class rules was found to be 4075 kg. This load is determined according to the maximum safe stress value in the boom, which is 237 N/mm². As a result of the MATLAB code designed according to the same values, the stress on the boom was obtained as 231.5 N/mm², but as the boom length gets longer and the spring coefficients increase, differences occur in the shock absorbing behavior of the system. Even if there is a shock absorber, there is a reduction in lifting capacity under the class rules. MATLAB parameters for 7.5m radius 10-ton crane MATLAB inputs are shown in **Table 4.1** below.

Table 4.1: MATLAB System Parameters.

System Parameter	Value
Mass of the crane boom (kg)	1243
Mass of the shock absorber (kg)	167
Mass of the working load (kg)	8800
Spring constant of the crane + crane hook ropes (N/m)	983715
Spring constant of the shock absorber system (N/m)	48420
Spring constant of the lifting beam and lower rope (N/m)	17189417
Damping constant of the shock absorber system (Ns/m)	5800
Amplitude of input signal (m)	2,16
Angular frequency of input signal (rad/s)	1
Diameter of spring k _c (m)	0,0284
Diameter of spring k _s (m)	0,05
Diameter of spring k _R (m)	0,0368

4.1.4. Rough Sea SWL Reduction According to MATLAB Working Model 2D

Figure 4.7, to **Figure 4.12** are the results obtained from the working model 2d program. The program performed a dynamic simulation according to our input signal depending on time. The stress equation can be obtained in the Working Model 2D program. Only the forces acting on the boom are seen. According to these stresses, a relationship is established between our reference load value and the maximum lifting capacity is determined. More detailed and more realistic results can be obtained. Solutions have been found for all cranes, but the results obtained for only a few cranes are shared in the images. **Figures 4.13** to **Figure 4.20**, simulation studies of different combinations of cranes were made and their comparisons were made according to the class rules.

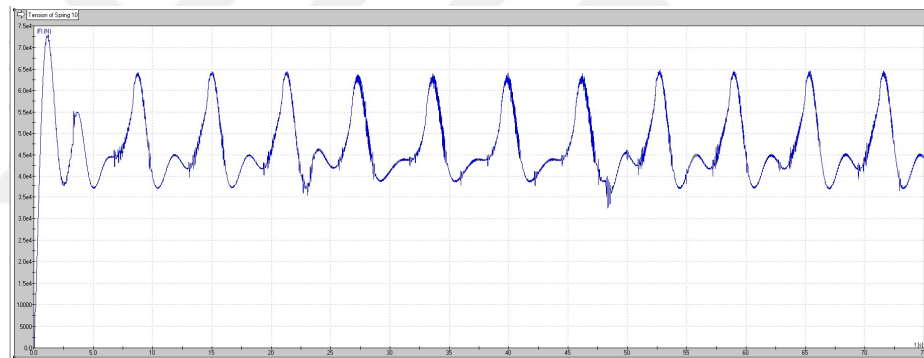


Figure 4.7: 7.5 m, Crane with absorber 3m wave height

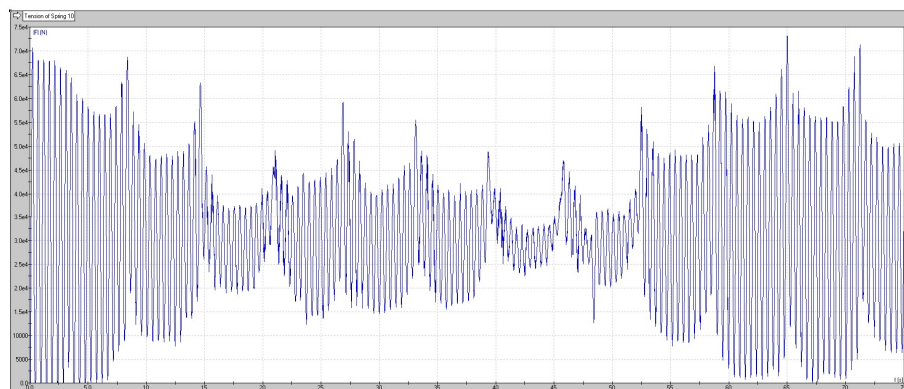


Figure 4.8: 7.5 m, Crane without shock absorber 3m wave height

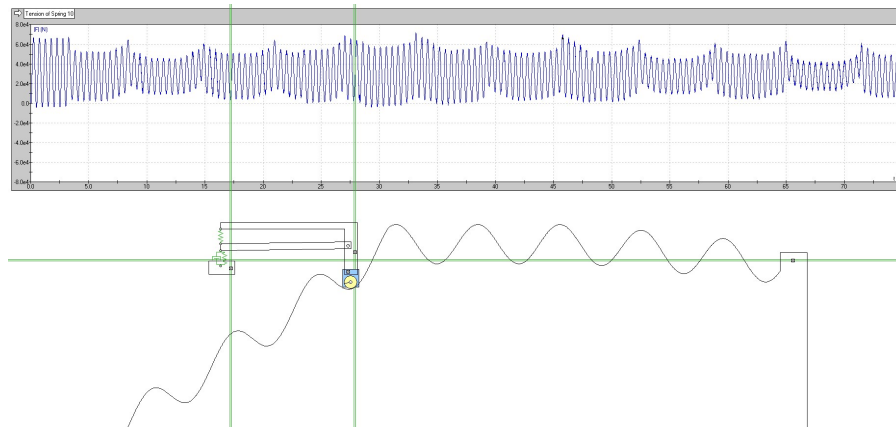


Figure 4.9: 10 m, Crane without shock absorber 3m wave height

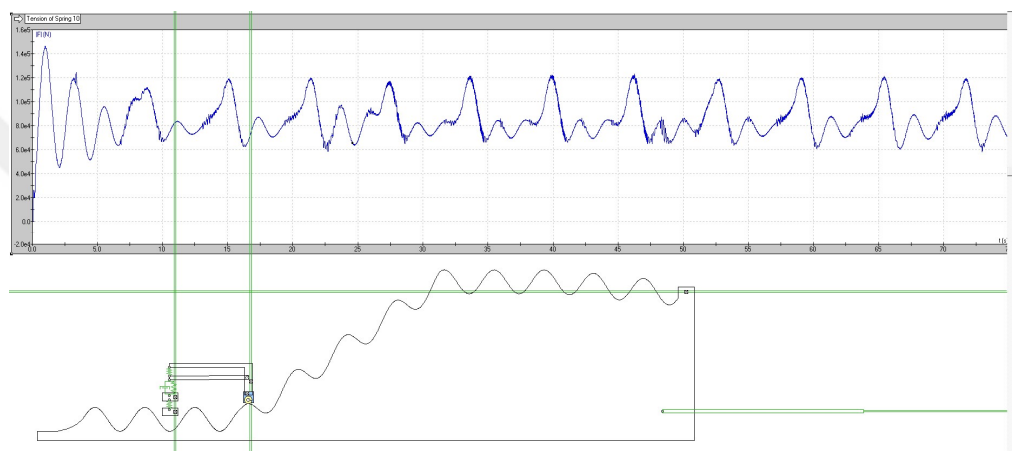


Figure 4.10: 10 m, Crane with shock absorber 3m wave height

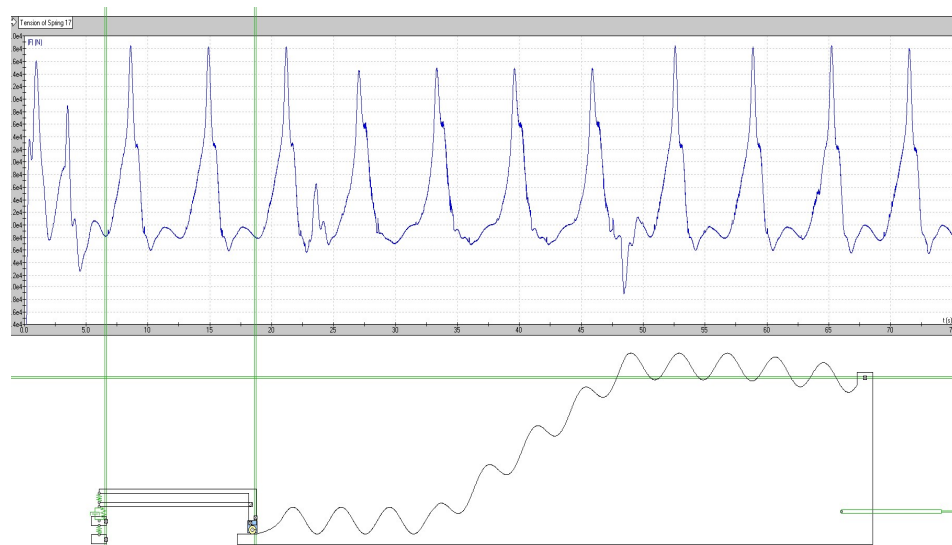


Figure 4.11: 20 m, Crane with shock absorber

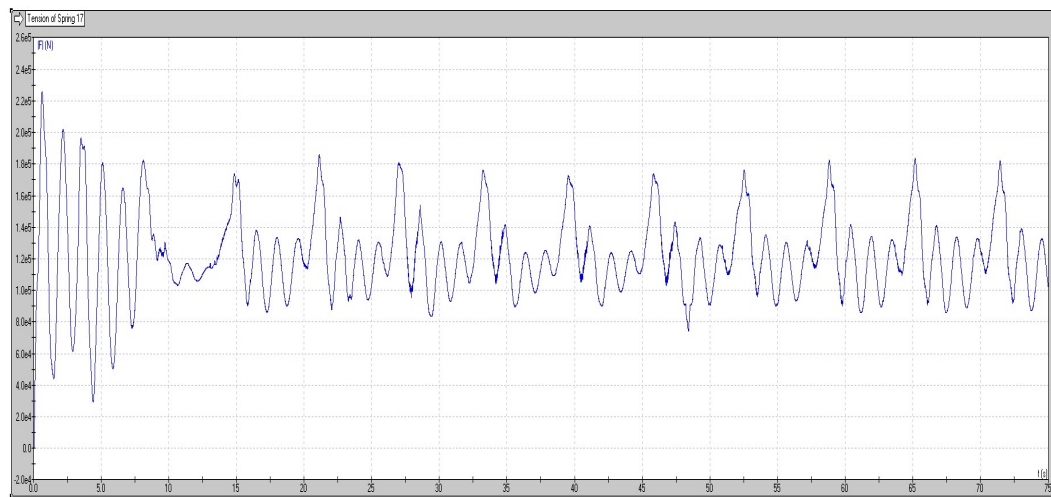


Figure 4.12: 20 m, Crane without shock absorber

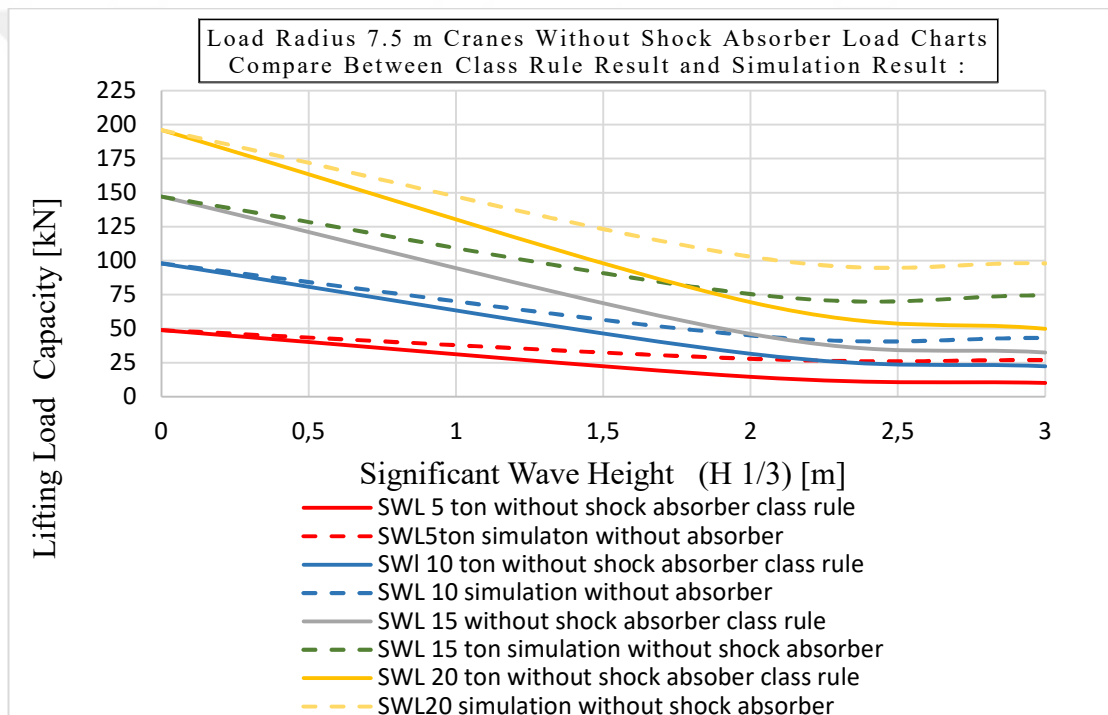


Figure 4.13: 7.5m, Cranes Without Shock Absorber Load Charts

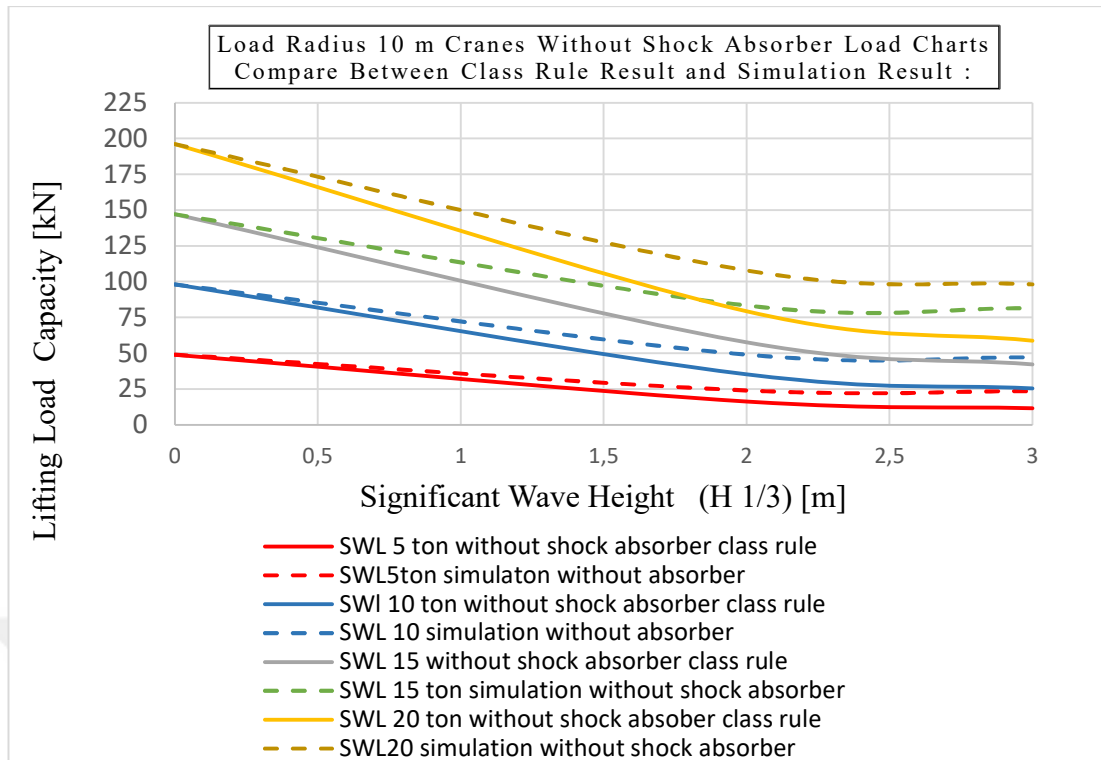


Figure 4.14: 10m, Cranes Without Shock Absorber Load Charts

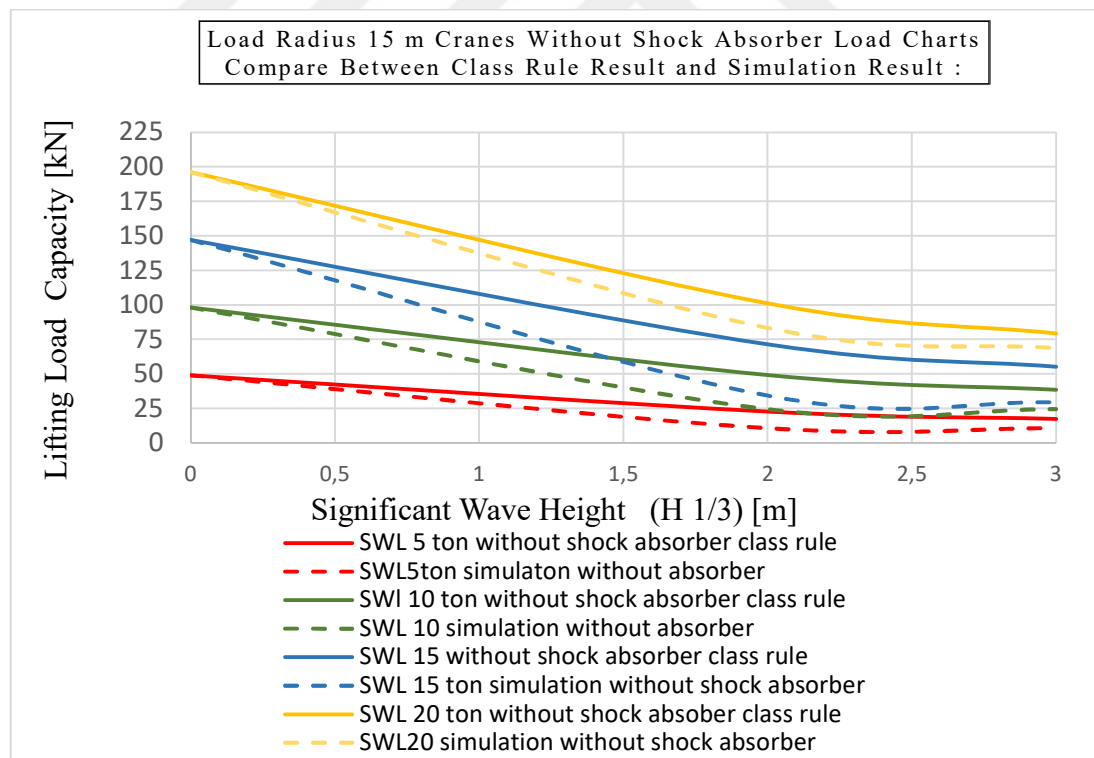


Figure 4.15: 15 m, Cranes Without Shock Absorber Load Charts

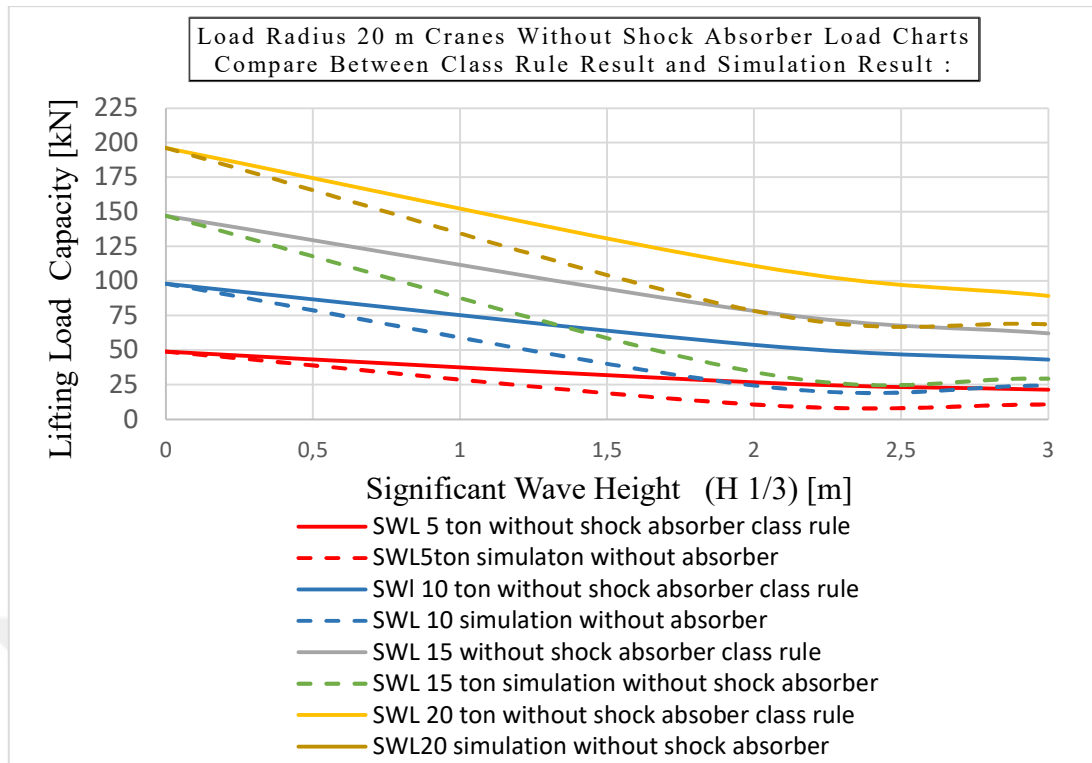


Figure 4.16: 20 m, Cranes Without Shock Absorber Load Charts

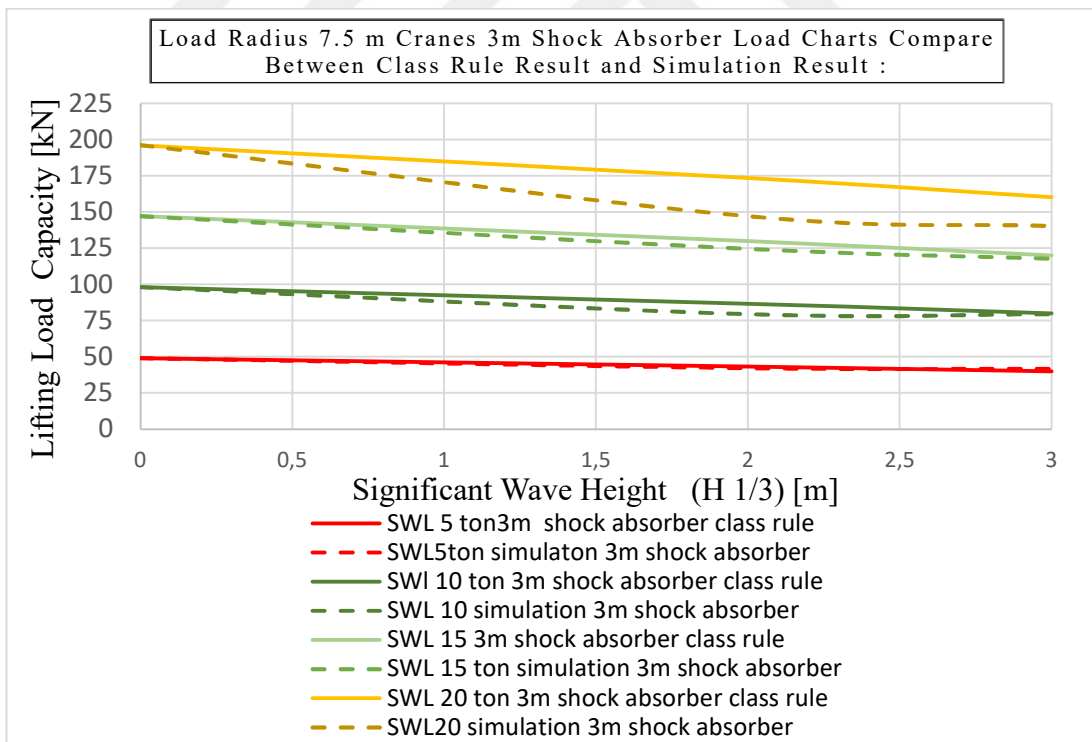


Figure 4.17: 7.5 m, Cranes 3m Shock Absorber Load Charts

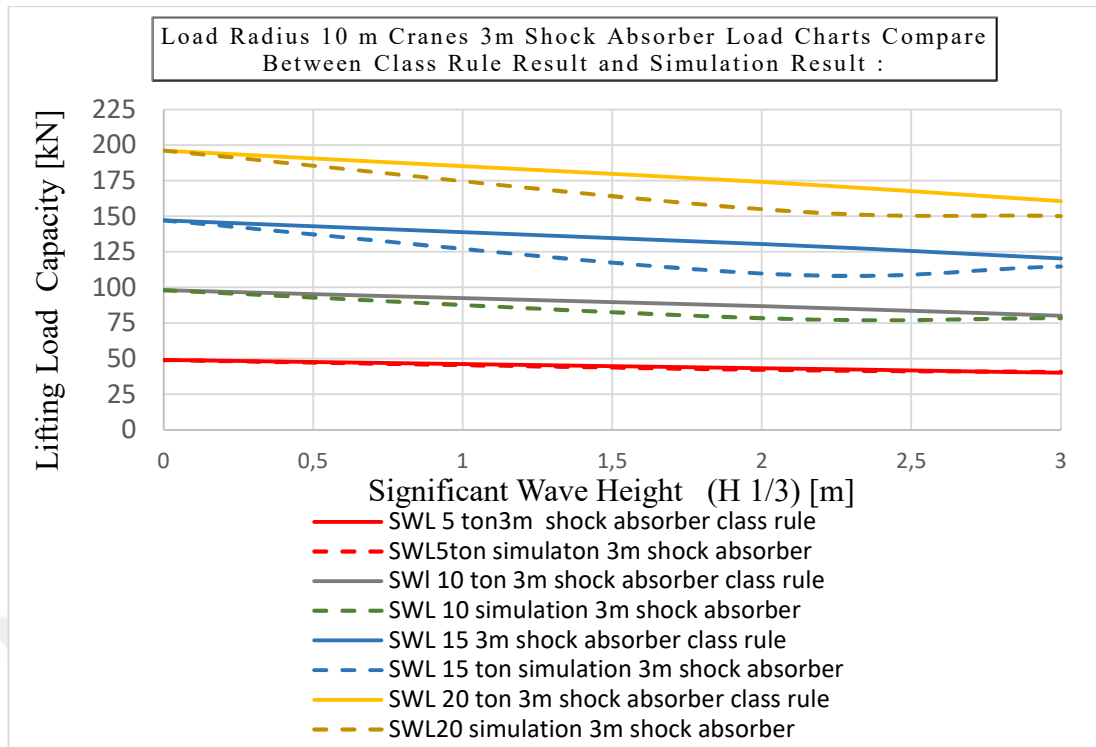


Figure 4.18: 10 m, Cranes 3m Shock Absorber Load Charts

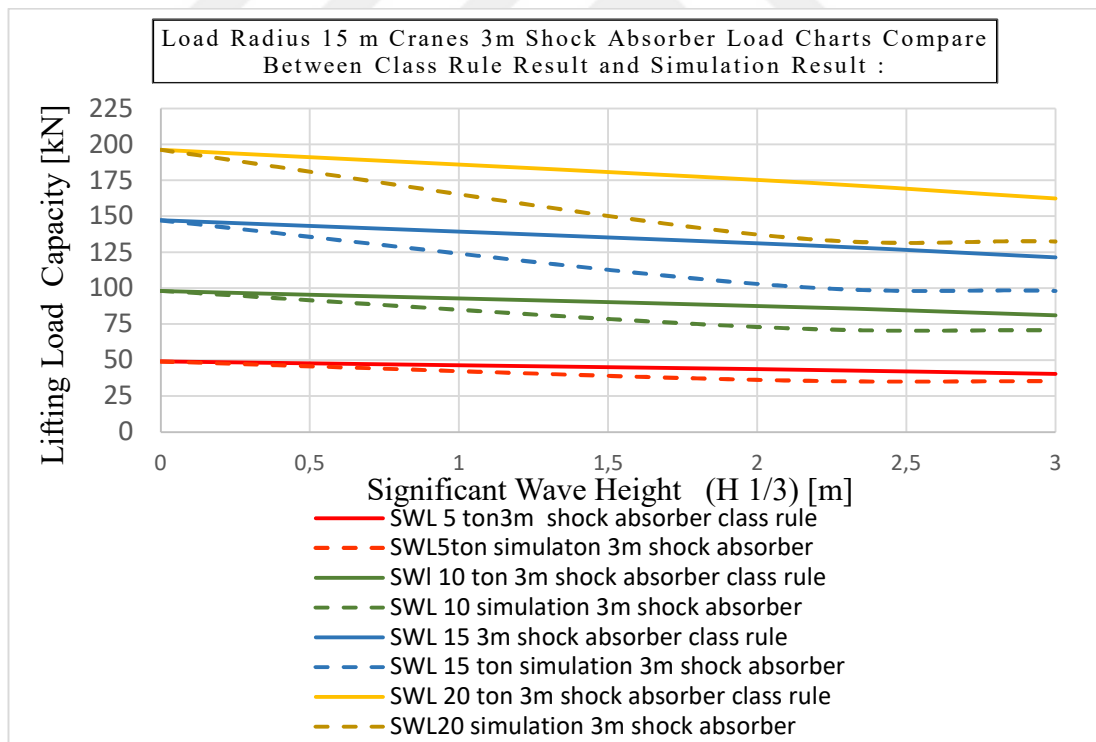


Figure 4.19: 15 m, Cranes 3m Shock Absorber Load Charts

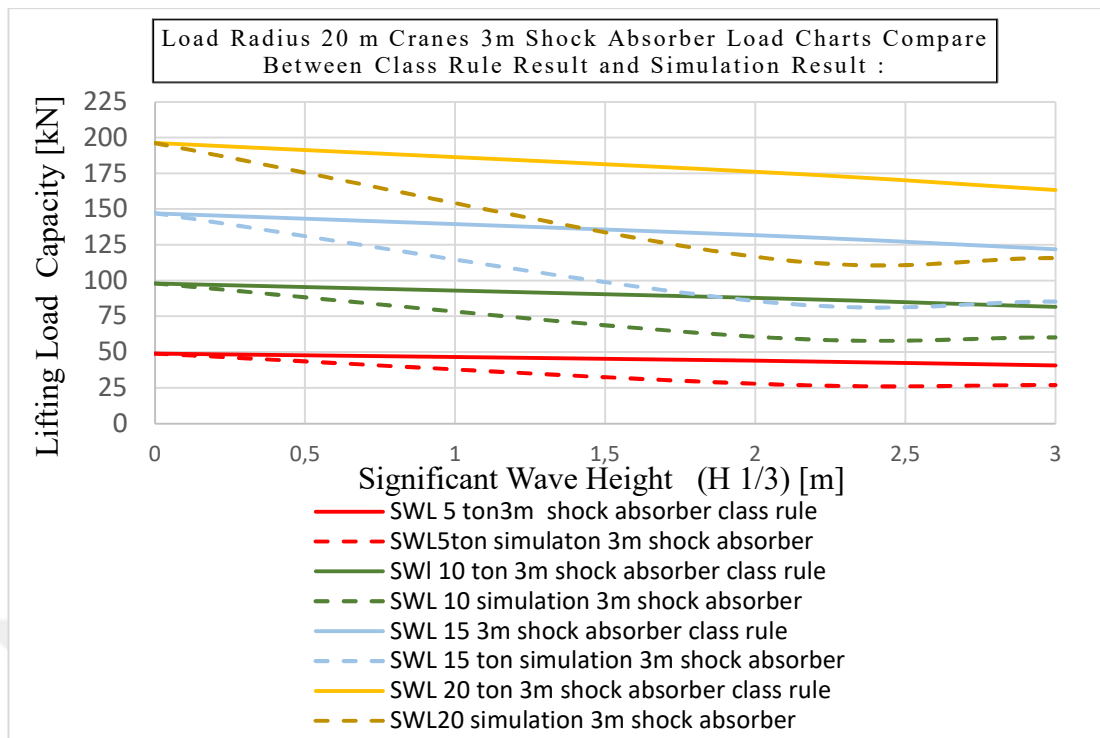


Figure 4.20: 20 m, Cranes 3m Shock Absorber Load Charts

5. RESULT AND REVIEWS

In our study, the design and dynamic system calculations of 16 different hydraulically driven cranes, frequently preferred on ships, with capacities ranging from 5 tons to 20 tons and reach distances from 7.5 meters to 20 meters, were conducted. Firstly, the load reduction coefficients of the crane under rough sea conditions were evaluated with two dynamic system models and three different approaches according to class requirements. Calm water, 2-meter wave, and 3-meter wave situations were assessed. All elements that affect the dynamic system modeling of the cranes to be used in the models were designed according to EN and international class rules references. For strength calculations, mathematical equations were obtained according to class rules, a calculation program was developed, and this calculation program was verified with FEA (Finite Element Analysis) calculations. Subsequently, based on the obtained system parameters, evaluations were made for a total of 320 different locations, including 3 sea states with 3 different calculation methods and calculations with and without shock absorbers.

The results of the calculation program used in the strength calculations in the study are very similar to those of the FEA program. This program can be used for the preliminary design of the crane.

The following evaluations can be made for the 3 different shock absorbers used in the calculations, ranging from 3 meters to 1 meter: If the shock absorbers are handling cargo, especially in the splash zone of the sea or during ship-to-ship transfers, it is more advantageous for system and personnel safety if the shock absorbers stroke is equal to the wavelength. This helps to prevent the rope from suddenly slackening. Although changes in the spring coefficient resulting from differences in the stock of the pistons are effective in increasing the SWL (Safe Working Load) of the crane, the change in length does not have a linear effect on the SWL increase. The springiness of the shock absorbers used in offshore transfers should be linear; that is, the preload on the piston should be very low when the stroke is fully extended. The load should be at its maximum when the piston is fully compressed. If there is a load on the piston when it is fully extended, the piston will compress after the system reaches a certain load, leading to sudden impacts. In pistons used in deep dives, it may be possible for the piston to be preloaded at its most extended position because the system already carries

the load while lifting by crane, and the main purpose is to absorb the hydrodynamic effects and stabilize the position.

The simulation results showed that when the load is lifted in wavy seas without a shock absorber, a continuous sinusoidal load is applied on the crane due to the lack of damping effect of the system or steel structure. This has a significantly worse impact. These effects are very negative, especially in terms of fatigue resistance.

Although the results of the models prepared in MATLAB or Working Model 2D vary as the boom length increases, the obtained values confirm each other.

In rough seas, shock-damping elements have a significant effect on the load reduction coefficients, but to find this coefficient, it is important to obtain results with simulation models and know the damping values. The reason that class rules allow the load reduction effect of shock absorbers to be adjusted if desired is that the results obtained from shock absorbers can vary; therefore, a detailed evaluation of the subject has not been conducted.

In both simulation calculations, it was determined that when the boom length was extended, the weight of the boom reached values close to the lifted load, and when the wave height was very high, the carrying capacities were lower than the load obtained from the load reduction formula. Especially as the length of the crane increases, springing increases, and resonance effects occur. In these cases, the damping value of the shock absorber has serious positive effects on the system. In critical designs and special operations, damping and spring coefficient calculations should be made with a simulation approach.

Due to the differences between the simulation results and the rules as the boom length increases and the weight of the crane arm is not included in the load reduction formula, more detailed 3-axis simulation setups should be performed in the future to determine whether it is necessary to include this effect in the formula. According to the results of this preliminary study, future studies should include changes in the winding of the rope on the reel drum, the effects of dip waves, and time-dependent FEA calculations of the crane. It would be more beneficial to prove the accuracy of the calculation methods with pool experiments.

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BIOGRAPHY

Onur Erdoğan is an experienced mechanical engineer specializing in in-ship equipment and mechanical design. He holds a bachelor's degree in mechanical engineering from Trakya University and is pursuing a master's in mechanical engineering at Gebze Technical University.

Since 2015, Onur has been with Türk Loydu Conformity Assessment Services Corporation as Principal Plan Approval Engineer and Chief of the Equipment Certification Department. He oversees conformity assessments, leads a team of engineers and technicians, and supports rule development for Türk Loydu Classification rules.

Previously, Onur worked at Gemak Group of Companies, Tuzla Shipbuilding Industry, designing special production lines and machines for shipyards, including Osman Gazi bridge deck, the 3rd Bosphorus Bridge in Istanbul, 1915 Çanakkale Bridge, worked on ship and offshore vessels equipment design also his expertise includes gantry cranes, welding lines, steel transportation equipment, and heavy lifting operations.

Onur has over 20 years of experience, is fluent in English and Turkish, and holds several certifications for marine conformity assessment.

PUBLICATIONS AND PRESENTATIONS FROM THE THESIS

Erdoğan O., Lotfan S., (2025), “Dynamic Modeling and Performance Analysis of Shipboard Crane with Shock Absorber”, GTÜ 9. Lisansüstü Araştırmalar Sempozyumu



APPENDICES

APPENDIX-A: MATLAB CODES

U INPUT SIGNAL MATLAB EXCEL TRANSFER CODE (3M WAVE HEIGHT)

% Parameters

A = 1.5; % Amplitude

omega = 1; % Angular frequency

% Time intervals

t1 = linspace(0, 20, 500); % 0 to 20 seconds

t2 = linspace(20, 45, 500); % 20 to 45 seconds

t3 = linspace(45, 60, 300); % 45 to 60 seconds

t4 = linspace(60, 75, 300); % 60 to 75 seconds

% Coefficients for the linear term

V_H1 = 0;

V_H2 = 0.69;

V_H3 = 0;

V_H4 = -0.1;

% Functions

y1 = A * sin(omega * t1) + V_H1 * t1;

y2 = A * sin(omega * t2) + (A * sin(omega * t1(end)) + V_H1 * t1(end)) + V_H2 * (t2 - t2(1)); % Adjust starting value for the 2nd interval

y2 = y2 - y2(1) + y1(end); % Align y2 with the end of y1

y3 = A * sin(omega * t3) + (A * sin(omega * t2(end)) + V_H2 * (t2(end) - t2(1))) + V_H3 * (t3 - t3(1)); % Adjust starting value for the 3rd interval

y3 = y3 - y3(1) + y2(end); % Align y3 with the end of y2

```

y4 = A * sin(omega * t4) + (A * sin(omega * t3(end)) + V_H3 * (t3(end) - t3(1))) +
V_H4 * (t4 - t4(1)); % Adjust starting value for the 4th interval
y4 = y4 - y4(1) + y3(end); % Align y4 with the end of y3

% Combine time and Y values
t = [t1 t2 t3 t4];
y = [y1 y2 y3 y4];

% Get the desktop path
desktopPath = fullfile(getenv('USERPROFILE'), 'Desktop');

% Save the coordinates to an Excel file on the desktop
coords = table(t, y, 'VariableNames', {'Time_s', 'y_m'});
writetable(coords, fullfile(desktopPath, 'graph_results.xlsx'));

% Plot the graph
figure;
plot(t, y, 'b', 'LineWidth', 2);
hold on;

% Highlight different sections in different colors
plot(t1, y1, 'r', 'LineWidth', 2);
plot(t2, y2, 'g', 'LineWidth', 2);
plot(t3, y3, 'm', 'LineWidth', 2);
plot(t4, y4, 'c', 'LineWidth', 2);

% Add annotations for parameter effects
text(5, A, 'A', 'FontSize', 12, 'Color', 'r');
text(30, y2(1), 'V_{H2}=0.69', 'FontSize', 12, 'Color', 'g');
text(50, y3(1), 'V_{H3}=0', 'FontSize', 12, 'Color', 'm');
text(70, y4(1), 'V_{H4}=-0.1', 'FontSize', 12, 'Color', 'c');

% Graph properties
title('Combined Graph of  $y = A \sin(\omega t) + V_H t$ ');

```

```

xlabel('Time t (seconds)');
ylabel('y (meters)');
grid on;
legend('Combined y = A \sin(\omega t) + V_H t', '0-20s, V_{H1}=0', '20-45s, V_{H2}=0.69', '45-60s, V_{H3}=0', '60-75s, V_{H4}=-0.1');
hold off;

% Print message for successful Excel creation
disp(['Graph coordinates have been saved to ' fullfile(desktopPath, 'graph_results.xlsx')]);

```

MATLAB 4 STEP SIMULATION CODE

```

% Parameters
A = 1.5;    % Amplitude
omega = 1;  % Angular frequency

% Time intervals
t1 = linspace(0, 20, 500); % 0 to 20 seconds
t2 = linspace(20, 45, 500); % 20 to 45 seconds
t3 = linspace(45, 60, 300); % 45 to 60 seconds
t4 = linspace(60, 75, 300); % 60 to 75 seconds

% Coefficients for the linear term
V_H1 = 0;
V_H2 = 0.69;
V_H3 = 0;
V_H4 = -0.1;

% Functions
y1 = A * sin(omega * t1 ) + V_H1 * t1;
y2 = A * sin(omega * t2) + (A * sin(omega * t1(end)) + V_H1 * t1(end)) + V_H2 * (t2 - t2(1)); % Adjust starting value for the 2nd interval
y2 = y2 - y2(1) + y1(end); % Align y2 with the end of y1

```

```

y3 = A * sin(omega * t3) + (A * sin(omega * t2(end)) + V_H2 * (t2(end) - t2(1))) +
V_H3 * (t3 - t3(1)); % Adjust starting value for the 3rd interval
y3 = y3 - y3(1) + y2(end); % Align y3 with the end of y2

```

```

y4 = A * sin(omega * t4) + (A * sin(omega * t3(end)) + V_H3 * (t3(end) - t3(1))) +
V_H4 * (t4 - t4(1)); % Adjust starting value for the 4th interval
y4 = y4 - y4(1) + y3(end); % Align y4 with the end of y3

```

```

% Combine time and Y values

```

```

t_combined = [t1 t2 t3 t4];

```

```

y_combined = [y1 y2 y3 y4];

```

```

% Remove duplicate time points

```

```

[t_unique, unique_indices] = unique(t_combined);

```

```

y_unique = y_combined(unique_indices);

```

```

% Re-sample time vector to be monotonically increasing and evenly spaced

```

```

t_resampled = linspace(min(t_unique), max(t_unique), length(t_unique));

```

```

y_resampled = interp1(t_unique, y_unique, t_resampled);

```

```

% System parameters

```

```

M_c = 917; % Mass of the crane boom (kg)

```

```

M_s = 87; % Mass of the shock absorber (kg)

```

```

M_WL = 4075; % Mass of the working load (kg)

```

```

k_c = 554376; % Spring constant of the crane + crane hook ropes (N/m)

```

```

k_s = 24162; % Spring constant of the shock absorber system (N/m)

```

```

k_R = 9090572; % Spring constant of the lifting beam and lower rope (N/m)

```

```

c_s = 5800; % Damping constant of the shock absorber system (Ns/m)

```

```

d_c = 0.015; % Diameter of spring kc (m)

```

```

d_s = 0.03; % Diameter of spring ks (m)

```

```

d_R = 0.0226; % Diameter of spring kR (m)

```

```

% Input signal (u) olarak yol verilerini kullanın

```

```

u = y_resampled;

% State-space matrix
A_matrix = [0 1 0 0 0 0;
            -(k_c+k_s)/M_c -c_s/M_c k_s/M_c c_s/M_c 0 0;
            0 0 0 1 0 0;
            k_s/M_s c_s/M_s -(k_s+k_R)/M_s -c_s/M_s k_R/M_s 0;
            0 0 0 0 0 1;
            0 0 k_R/M_WL 0 -k_R/M_WL 0];
B_matrix = [0;
            k_c/M_c;
            0;
            0;
            0;
            0];
C_matrix = [1 0 0 0 0 0;
            0 0 1 0 0 0;
            0 0 0 0 1 0];
D_matrix = [0;
            0;
            0];

% Create state-space model
sys = ss(A_matrix, B_matrix, C_matrix, D_matrix);

% Simulate system response
[y_out, t_out, x] = lsim(sys, u, t_resampled);

% Calculate spring extensions (m)
delta_kc = u' - y_out(:,1); % Extension of spring kc
delta_ks = y_out(:,1) - y_out(:,2); % Extension of spring ks
delta_kR = y_out(:,2) - y_out(:,3); % Extension of spring kR

```

```

% Convert spring extensions to meters (m)
delta_kc_m = delta_kc; % Assuming input signal u is in meters
delta_ks_m = delta_ks; % Assuming displacements are in meters
delta_kR_m = delta_kR; % Assuming displacements are in meters

% Calculate spring stresses (N/m²)
A_c = pi * (d_c/2)^2; % Cross-sectional area of spring kc (m²)
A_s = pi * (d_s/2)^2; % Cross-sectional area of spring ks (m²)
A_R = pi * (d_R/2)^2; % Cross-sectional area of spring kR (m²)

F_kc = k_c * delta_kc_m; % Force in spring kc (N)
F_ks = k_s * delta_ks_m; % Force in spring ks (N)
F_kR = k_R * delta_kR_m; % Force in spring kR (N)

sigma_kc = F_kc / A_c; % Stress in spring kc (N/m²)
sigma_ks = F_ks / A_s; % Stress in spring ks (N/m²)
sigma_kR = F_kR / A_R; % Stress in spring kR (N/m²)

% Convert stress to N/mm²
sigma_kc_mm2 = sigma_kc * 1e-6;
sigma_ks_mm2 = sigma_ks * 1e-6;
sigma_kR_mm2 = sigma_kR * 1e-6;

% Plot input signal and outputs
figure;
subplot(5,1,1);
plot(t_resampled, u);
title('Input Signal u');
xlabel('Time (s)');
ylabel('u (m)');

subplot(5,1,2);
plot(t_resampled, y_out(:,1));

```

```

title('Output y1');
xlabel('Time (s)');
ylabel('y1 (m)');

subplot(5,1,3);
plot(t_resampled, y_out(:,2));
title('Output y2');
xlabel('Time (s)');
ylabel('y2 (m)');

subplot(5,1,4);
plot(t_resampled, y_out(:,3));
title('Output y3');
xlabel('Time (s)');
ylabel('y3 (m)');

% Plot spring stresses in separate figures
subplot(5,1,5);
hold on;
plot(t_resampled, sigma_kc_mm2, 'r');
plot(t_resampled, sigma_ks_mm2, 'g');
plot(t_resampled, sigma_kR_mm2, 'b');
title('Stress in Springs');
xlabel('Time (s)');
ylabel('Stress (N/mm²)');
legend('Stress \sigma_{kc}', 'Stress \sigma_{ks}', 'Stress \sigma_{kR}');
grid on;

% Animation
figure;
% Define positions of masses and springs
y1_base = 2; % Base position of mass 1 (m)
y2_base = 1; % Base position of mass 2 (m)
y3_base = 0; % Base position of mass 3 (m)

```

```

% Plot settings
axis([-0.5 1.5 -1 30]);
hold on;

% Initial positions of masses
h_m1 = rectangle('Position',[0.4 y1_base + y_out(1,1) 0.2 0.2], 'Curvature', [0.1 0.1],
'FaceColor', 'blue');
h_m2 = rectangle('Position',[0.4 y2_base + y_out(1,2) 0.2 0.2], 'Curvature', [0.1 0.1],
'FaceColor', 'green');
h_m3 = rectangle('Position',[0.4 y3_base + y_out(1,3) 0.2 0.2], 'Curvature', [0.1 0.1],
'FaceColor', 'red');

% Labels for masses
text(0.65, y1_base + y_out(1,1) + 0.1, 'M_c', 'FontSize', 12, 'Color', 'blue');
text(0.65, y2_base + y_out(1,2) + 0.1, 'M_s', 'FontSize', 12, 'Color', 'green');
text(0.65, y3_base + y_out(1,3) + 0.1, 'M_{WL}', 'FontSize', 12, 'Color', 'red');

% Initial positions of springs
h_spring1 = plot([0.5 0.5], [3 y1_base + y_out(1,1) + 0.2], 'r', 'LineWidth', 2);
h_spring2 = plot([0.5 0.5], [y1_base + y_out(1,1) y2_base + y_out(1,2) + 0.2], 'g',
'LineWidth', 2);
h_spring3 = plot([0.5 0.5], [y2_base + y_out(1,2) y3_base + y_out(1,3) + 0.2], 'b',
'LineWidth', 2);

% Labels for springs and dampers
text(0.3, 2.5, 'k_c', 'FontSize', 12, 'Color', 'red');
text(0.3, 1.5, 'k_s', 'FontSize', 12, 'Color', 'green');
text(0.3, 0.5, 'k_R', 'FontSize', 12, 'Color', 'blue');
text(0.7, y1_base + (y2_base - y1_base) / 2 + y_out(1,1) / 2 + y_out(1,2) / 2, 'c_s',
'FontSize', 12, 'Color', 'black');

% Initial position of the damper

```

```
h_damper = plot([0.45 0.55], [y1_base + (y2_base - y1_base) / 2 + y_out(1,1) / 2 +
y_out(1,2) / 2 y1_base + (y2_base - y1_base) / 2 + y_out(1,1) / 2 + y_out(1,2) / 2],
'k', 'LineWidth', 6);
```

```
% Animation loop
```

```
for i = 1:length(t_resampled)
```

```
    % Update positions of masses
```

```
    if invalid(h_m1)
```

```
        set(h_m1, 'Position', [0.4 y1_base + y_out(i,1) 0.2 0.2]);
```

```
    end
```

```
    if invalid(h_m2)
```

```
        set(h_m2, 'Position', [0.4 y2_base + y_out(i,2) 0.2 0.2]);
```

```
    end
```

```
    if invalid(h_m3)
```

```
        set(h_m3, 'Position', [0.4 y3_base + y_out(i,3) 0.2 0.2]);
```

```
    end
```

```
% Update positions of springs
```

```
    if invalid(h_spring1)
```

```
        set(h_spring1, 'YData', [3 y1_base + y_out(i,1) + 0.2]);
```

```
    end
```

```
    if invalid(h_spring2)
```

```
        set(h_spring2, 'YData', [y1_base + y_out(i,1) y2_base + y_out(i,2) + 0.2]);
```

```
    end
```

```
    if invalid(h_spring3)
```

```
        set(h_spring3, 'YData', [y2_base + y_out(i,2) y3_base + y_out(i,3) + 0.2]);
```

```
    end
```

```
% Update position of the damper
```

```
    if invalid(h_damper)
```

```
        set(h_damper, 'YData', [y1_base + (y2_base - y1_base) / 2 + y_out(i,1) / 2 +
y_out(i,2) / 2 y1_base + (y2_base - y1_base) / 2 + y_out(i,1) / 2 + y_out(i,2) / 2]);
```

```
    end
```

```

    % Pause to create animation effect
    pause(0.01);
end

```

```

hold off;

```

COMBINED SPRING COEFFICIENT GRAPHIC CODE

```

% Verileri tanımlama
load_radius = [7.5, 7.5, 7.5, 7.5, 10, 10, 10, 10, 15, 15, 15, 15, 20, 20, 20, 20];
swl = [5, 10, 15, 20, 5, 10, 15, 20, 5, 10, 15, 20, 5, 10, 15, 20];
combined_spring_coefficient = [554, 984, 1558, 1647, 461, 805, 1022, 1267, 233,
388, 636, 730, 155, 307, 503, 570];

% Verileri grid oluşturacak şekilde yeniden düzenleme
[load_radiusGrid, swlGrid] = meshgrid(linspace(min(load_radius),
max(load_radius), 100), linspace(min(swl), max(swl), 100));
combined_spring_coefficientGrid = griddata(load_radius, swl,
combined_spring_coefficient, load_radiusGrid, swlGrid, 'cubic');

% 3D yüzey grafiği oluşturma
figure;
surf(load_radiusGrid, swlGrid, combined_spring_coefficientGrid);
xlabel('Load Radius (m)');
ylabel('SWL (ton)');
zlabel('Combined Spring Coefficient (Nmm)');
title('3D Surface Plot of Load Radius, SWL, and Combined Spring Coefficient');

% İzometrik görünüm ayarlama
view(3);
grid on;

% Yüzeyin geçişlerini iyileştirme
shading interp; % Pürüzsüz geçişler için
colorbar;

```

```
colormap jet;
```

CRANE VERTICAL SPRING COEFFICIENT GRAPHIC CODE

```
% Verileri tanımlama
```

```
load_radius = [7.5, 7.5, 7.5, 7.5, 10, 10, 10, 10, 15, 15, 15, 15, 20, 20, 20, 20];
```

```
swl = [5, 10, 15, 20, 5, 10, 15, 20, 5, 10, 15, 20, 5, 10, 15, 20];
```

```
crane_vertical_spring_coefficient = [732, 1502, 2560, 3329, 609, 1243, 1504, 2534,  
279, 509, 882, 1251, 179, 405, 702, 990];
```

```
% Verileri grid oluşturacak şekilde yeniden düzenleme
```

```
[load_radiusGrid, swlGrid] = meshgrid(linspace(min(load_radius),  
max(load_radius), 100), linspace(min(swl), max(swl), 100));
```

```
crane_vertical_spring_coefficientGrid = griddata(load_radius, swl,  
crane_vertical_spring_coefficient, load_radiusGrid, swlGrid, 'cubic');
```

```
% 3D yüzey grafiği oluşturma
```

```
figure;
```

```
surf(load_radiusGrid, swlGrid, crane_vertical_spring_coefficientGrid);
```

```
xlabel('Load Radius (m)');
```

```
ylabel('SWL (ton)');
```

```
zlabel('Crane Vertical Spring Coefficient (Nmm)');
```

```
title('3D Surface Plot of Load Radius, SWL, and Crane Vertical Spring Coefficient');
```

```
% Izometrik görünüm ayarlama
```

```
view(3);
```

```
grid on;
```

```
% Yüzeyin geçişlerini iyileştirme
```

```
shading interp; % Pürüzsüz geçişler için
```

```
colorbar;
```

```
colormap jet;
```