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RUHR-UNIVERSITÄT BOCHUM

**MechEng**  
Mechanical Engineering



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# DESIGN OF A SOFT-SEATING LOWER ANCHOR CONNECTOR FOR SNAP LOAD REDUCTION IN TENSION LEG PLATFORM TETHERS

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Vietnamese-German University

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# **Design of a soft-seating lower anchor connector for snap load reduction in Tension Leg Platform tethers**

A BSc Thesis presented

by

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Vietnamese-German University

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## **Declaration of authorship**

I, Dang Hoang Dat, declare that this thesis is the product of my own work. I have accredited all the aids provided to me in the progress of making this work. Any thoughts directly or indirectly taken from somebody else's sources are made discernible as such.

To date, the thesis has not been submitted to any other board of examiners in the same or a similar format and has not been published yet.



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## Abstract

Tension Leg Platform (TLP) is a type of offshore structure that is widely used in deep water oil and gas production as well as the foundation for offshore wind turbines. As a floating platform, the TLP has excellent stability in various wave conditions with very strict limitation in vertical movements while allowing horizontal movements to some degree. This is achieved by using taut mooring system with tendons (tethers) that hold the platform down from its excess buoyancy and create a balance in vertical direction. In rough sea conditions, the tension load on tendons can vary greatly from zero to almost double the static pre-tension force in a short time, such an event is called a snap condition. Studies showed that brief periods of slacking do not itself damage the tendons but the snap load during re-tensioning poses severe fatigue to this crucial component of the TLP.

This thesis presents the design process for an alternative lower anchor connector (LAC) that aims at relieving the high stress in TLP tendons due to snap load by the use of a “soft-seating” mechanism. The result of this study shows the feasibility and good potential for reducing snap load in tendons with the model of a new tendon connector with integrated suspension system.



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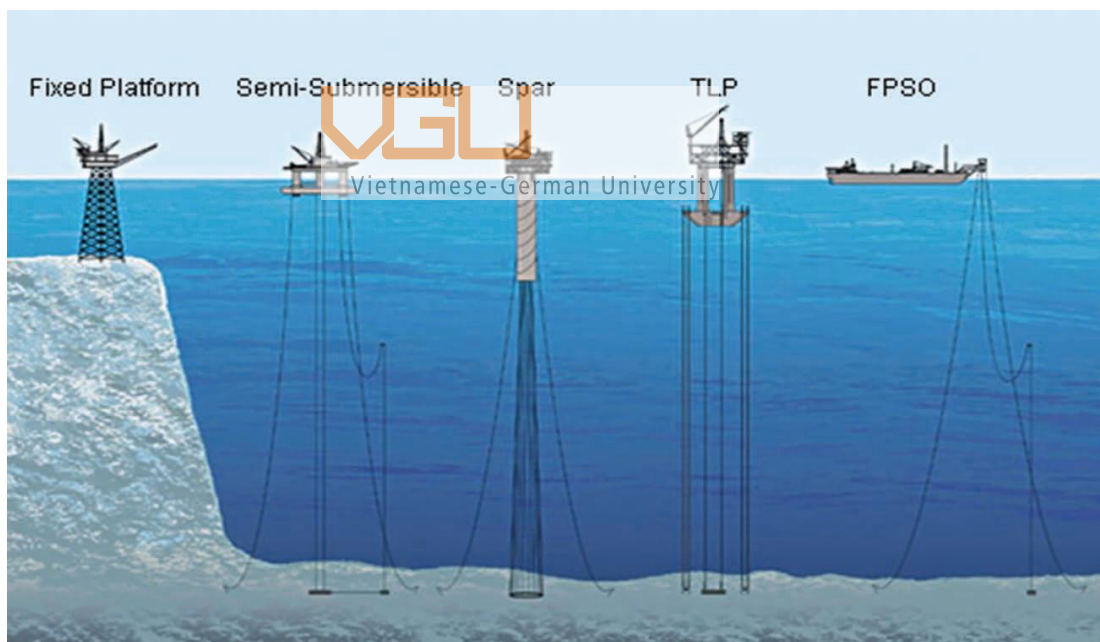
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# 1. Introduction

## 1.1. Overview of Tension Leg Platform

As offshore oil and gas exploration and production moved into deeper water over the years, floating platforms have become a safer and more economical choice compared to fixed platforms for oil field development. Semi-submersible is a floating platform with hollow members and uses slack mooring cables (catenary mooring) to stop it from floating away. The Spar has a large and hollow vertical cylinder to keep it afloat, with ropes or chains as mooring cables. Floating Production, Storage and Offloading systems (FPSOs) are vessels converted from oil tankers or specially built ships for storage and onboard production facilities, they are fitted with stations that are moored to the seabed and receive hydrocarbon through risers. These are called turrets because the vessels can rotate freely from them, allowing the FPSOs to point towards the direction of wind and currents, in order to alleviate the impacts of weather on the floating vessel.

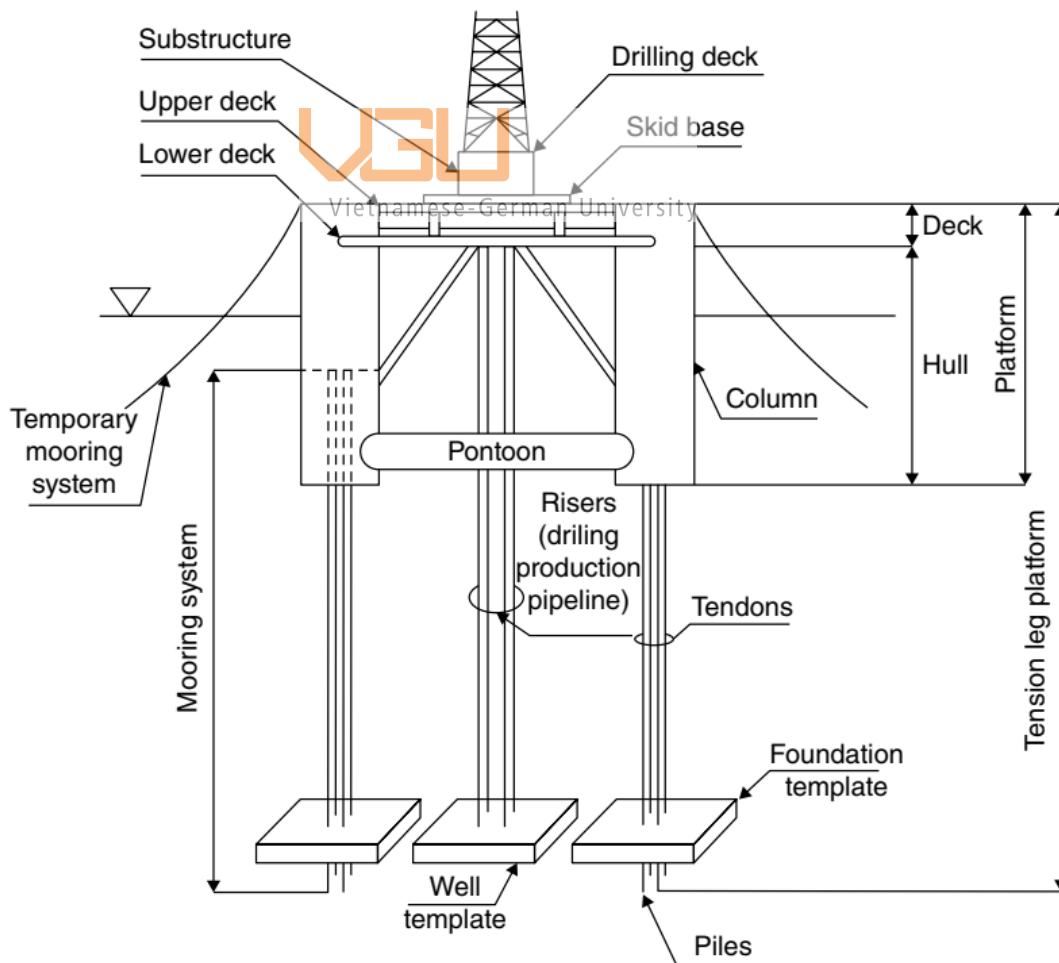


**Figure 1:** Types of offshore platforms [1]

Among the floating structures, Tension Leg Platform (TLP) is widely used thanks to its high vertical stability. A typical TLP consists of a top deck, below are hollow components: vertical columns and horizontal pontoons to provide redundant buoyancy. The platform is held down by tendons (or tethers) that are hollow pipes or stranded cables made of steel. A TLP is designed so that there is always tension in the tendons to keep

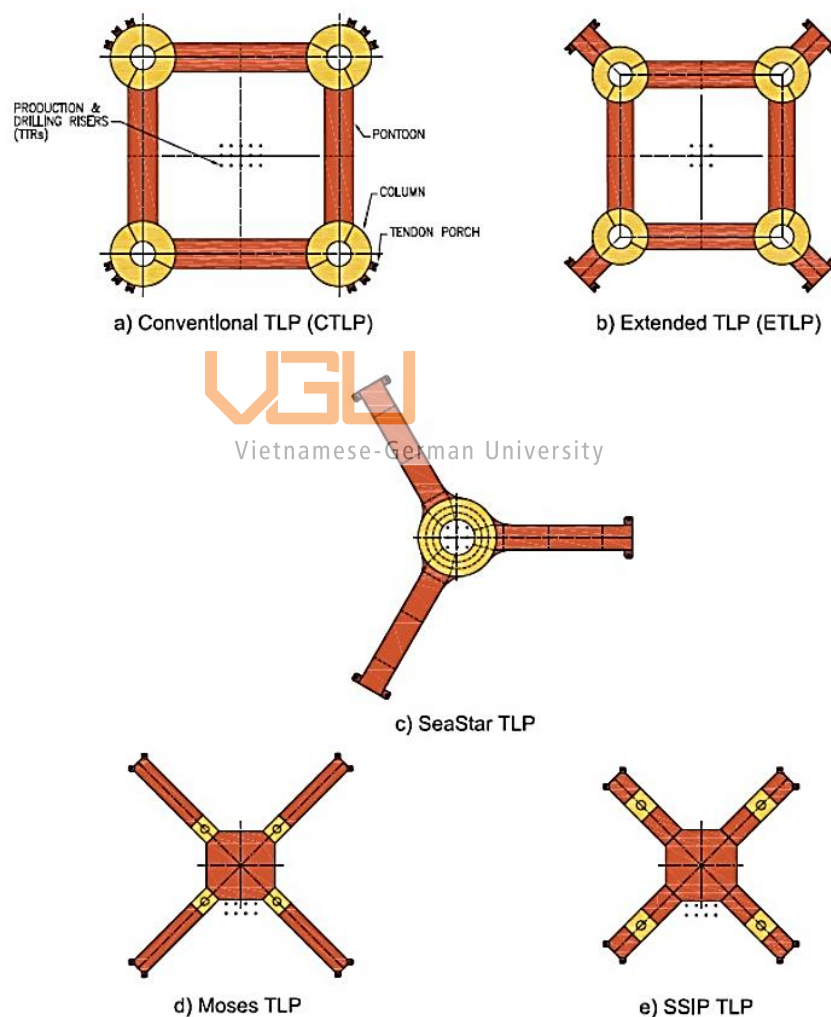
them from becoming slack for a significant period of time that can lead to platform instability and buckling failure of tendons in the case of steel pipes. Pre-tension in the tendons is the result of difference between buoyancy and dead weight of the hull, the weight is usually only around 2/3 of buoyancy [2].

Unlike catenary mooring on semi-submersibles, which uses slack and heavy steel chains to restore platform position by their weight, taut mooring on TLP uses the high tension and high stiffness of its tendons to restrain platform motions [3]. Vertical movements such as heave, roll and pitch are greatly diminished by the balance between excess buoyancy and high axial stiffness of steel tendons. As a result, the natural frequency of these vertical motions is much higher than that of sea waves. On the other hand, lateral movements such as surge, sway and yaw have lower frequency than sea waves due to low geometric stiffness of tendons [3], horizontal position of platform is restored by pre-tension force.



**Figure 2:** Typical configuration of a conventional TLP with 4 columns [3]

This response characteristic makes TLP a suitable choice for fields with various environmental conditions including rough seas because the distinct range of natural frequencies from sea waves help it avoid resonance with environmental influence. Besides wet trees (subsea trees), suppressed vertical motions of TLP also enables the use of dry tree systems that are otherwise only on fixed platforms, allowing access to oil wells directly from the platform, this significantly reduces the costs for well completion and intervention compared to subsea systems. If a subsea system is used, then the integrity of production risers, a critical yet vulnerable component of production system, can benefit from high stability of the platform in extreme sea states.

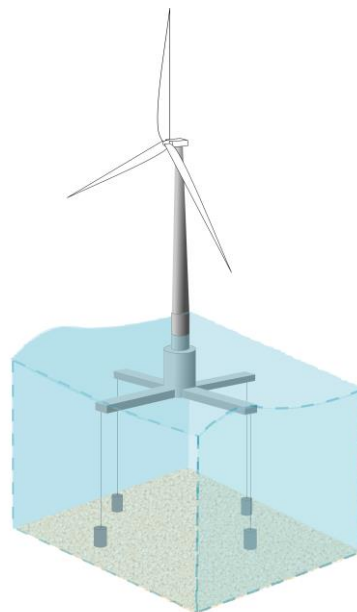


**Figure 3:** Different TLP configurations [4]

Since the first TLP installed for the Hutton field in 1984 (the field is situated in the East Shetland Basin in the UK North Sea), at the depth of about 160m, this type of platform has taken many forms over the years [4]. The “classic” TLP has 4 columns and a ring of

pontoons with tendons attached directly to the hull columns. The “extended” TLP has similar shape but it has extra horizontal extensions from the columns as top termination ends for tendons. More lightweight configurations such as SeaStar® and Moses® TLPs have a single central column as the main structure with 3 or 4 cantilevered pontoons extending from the hull to attach tendons, respectively. The single-column configurations have the advantage of lower costs for fabrication and installation thanks to more simple designs of them.

With the rise of renewable energy, numerous studies have been conducted in the last two decades focusing on the implementation of TLP as the foundation for offshore wind turbines. New designs have been proposed, and extensive analysis were done for this type of structure [5], [6]. Unlike conventional TLPs used in oil and gas with 4 columns and pontoons between them, the foundations for these wind turbines mostly employ the “SeaStar® mini TLP” design, which has a cylindrical hull as one single column and pontoon “legs” attached to tethers. This implementation for wind turbines has great potential because offshore wind is more abundant and more powerful, allowing the use of turbines with high power output up to 5MW. Their locations far from the coasts also avoid noise pollution to nearby residents, unlike ground wind turbines. Although water depth at the locations for these turbines is usually below 100m, the floating foundation allows them to reach further offshore while cut down the costs compared to fixed foundations on the sea bed.

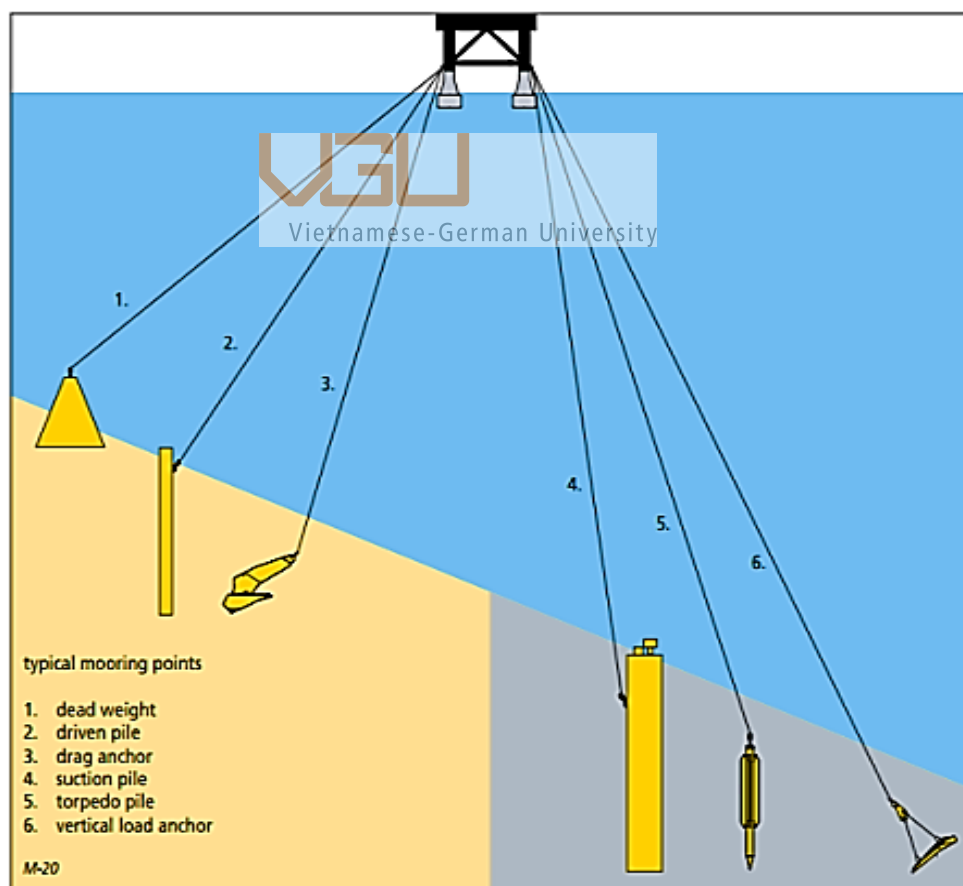


**Figure 4:** A SeaStar® mini TLP for offshore wind turbine [7]

## 1.2. TLP mooring system

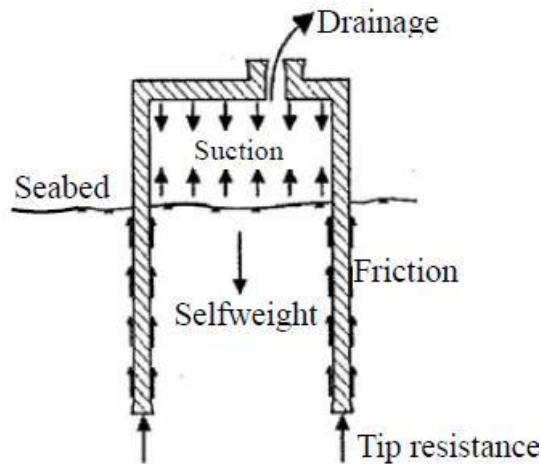
Taut mooring system of TLP consists of 2 main components: anchors and mooring cables. There are various solutions for anchoring of offshore floating platforms [8]:

- Deadweight anchor: a heavy object that uses its weight to resist movements in any direction. This is the simplest anchor.
- Drag Embedment Anchor (DEA): uses its unique shape to ‘grab’ the soil, commonly used for ships or catenary mooring where cables are diagonal because it only resists horizontal displacements.
- Anchor piles (driven pile, suction pile or torpedo pile): utilizes a long cylindrical tube with open bottom and a valve at the top. This type can resist loads in all directions.



**Figure 5:** Anchor types [8]

The most commonly used type of anchor for TLP is suction pile because it can hold the load well in vertical direction. The piles are lowered to the seabed and initially penetrate the soil with its own weight, then the water inside is pumped out to create a suction and force it down to designated depth.



**Figure 6:** Suction pile anchor diagram [9]

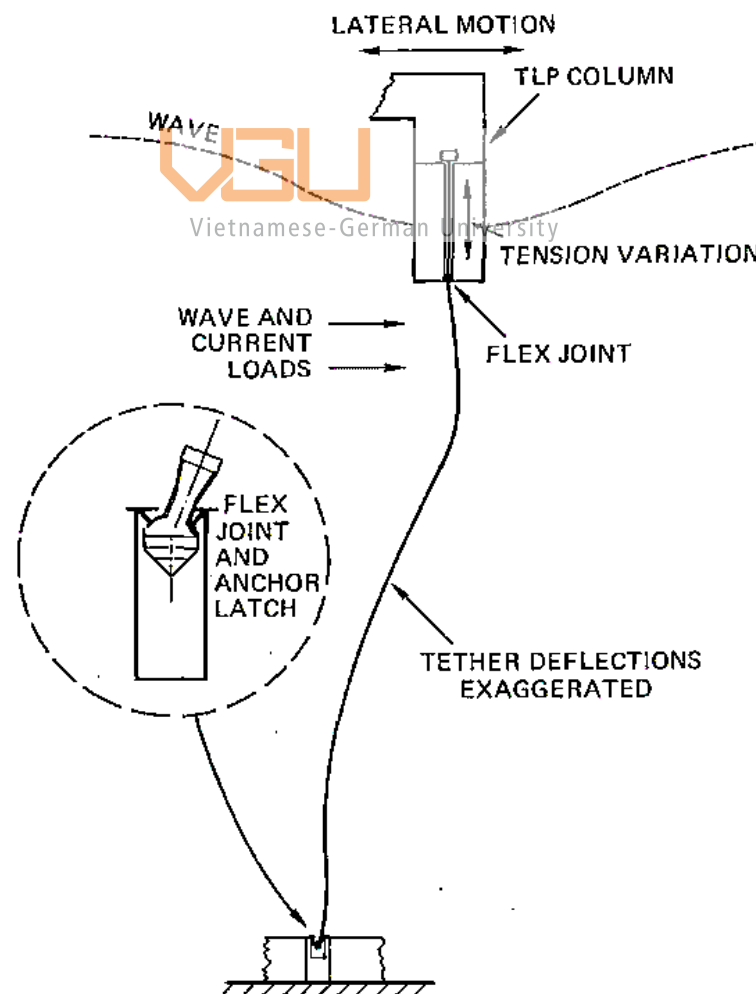
Mooring cables or tendons can be steel chains, stranded wire ropes, solid rods or steel tubulars. Hollow steel tubulars are the most common choice for TLP tendons, they are thin-walled pipes filled with air inside, their high buoyancy helps reduce tendons weight in water while the bore can be utilized for routing umbilical between seafloor and surface [10]. The tubular tendons are designed to only take axial load, that is why they always have to be kept taut by excess tension, the loss of tension can cause them to fail due to bending or buckling. The application of synthetic composites such as aramid and HMPE has been studied and may be considered as the alternative material for tendons.

### 1.3. Introduction to problem

One of the primary concerns for TLP is the integrity of its tendons because they are constantly subjected to high stress, even in calm weather. In more severe conditions, the cyclic loadings due to sea waves create even higher risk for the safety of the platform and its operations. Failure of one tendon can cause a sudden increase of stress in remaining ones, without immediate actions, this could lead to breakage of other tendons. Without this crucial component holding the platform down, it becomes unstable due to very high natural buoyancy [2].



The conventional anchor connector consists of anchor latch to secure the bottom of tethers or tendons, and flex joint to allow pivoting when the platform moves sideways. The current designs of the tendon bottom connector are illustrated in the figures in Appendix II. While this design tolerates rotation at the lower connector, it is still unforgiving when it comes to axial displacement. When the TLP goes from wave trough to wave crest during rough seas, platform's buoyant members (columns) go into a cycle of displacing less water to more water respectively. This results to fluctuation in structure's buoyancy and consequently tension force in tendons. Snap load, a sharp increase in tension within a short time, occurs when the tendons go from slack to taut and it costs serious fatigue to the tendons. With conventional rigid anchor connection, the snap load is translated to a spike in axial stress of tendons. Therefore, a "soft" connector that can absorb the impact and relieve the snap load on tendons is favored.



**Figure 7:** Conventional anchor connector for taut mooring [11]

## 1.4. Objectives

The aim of this thesis research is to develop a design for the soft-seating mechanism to reduce snap load on tendons and analyze the design. Design requirements and environmental parameters are identified to generate concepts, the most suitable solution is then selected from these concepts by using criteria that focus on reducing costs and having high reliability. This research project should produce a concept solution that is able to relieve the sudden increase of load in tendons but not eliminate the axial stress entirely. The mechanism will have minimal impact on the overall design of TLP as well as its function in normal sea conditions. The design is to be applied to tension leg platforms for oil and gas production in the Vietnamese coastal regions of the South China Sea.



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## 2. Literature review

### 2.1. Studies on the effects of snap loading

Tendons in taut mooring system of TLP are subjected to two kinds of axial loads: static load from pre-tension and dynamic loads due to the environment such as waves, currents, or seismic movements of the seabed. Among these factors, sea wave is the prominent and more frequent impact on tension force in tendons because it constantly changes platform's water displacement, thus has direct effect on the balance between its buoyancy and weight. Sea currents also has its presence under water surface but because of the relatively small cross section of tendons and horizontal direction of currents, its contribution to snap load could be negligible. Movements of the Earth's crust combined with high sea waves do contribute to the significant dynamic load in tendons [12]. However, this scenario rarely happens in the South China Sea [13].

A mooring line is said to experience snap load when the dynamic force is greater than 90% of the static load [14]. A snap condition usually occurs when the tendons go through a slack-taut process, in which they have insufficient tension (slack) and then bear large tension force (taut) as the result of sea waves. A study on the stress in mooring lines in the slack-taut condition found that the dynamic tension is even several times of pre-tension [15]. Another research that investigated on the segmented cable consisting of stranded steel cable on top and lower nylon rope concluded that the sudden impact has maximum magnitude at the top of cable and it can be large enough to cause fatigue failure even for designs with high safety factor [16]. It is also confirmed that the occurrence of snap load has a noticeable correlation with wave motions, it can happen not only in extreme sea states but also in other conditions [17]. According to a study that simulates the effect of one tendon failure on the safety of the TLP in hurricane conditions, the loss of a tendon during big storms will immediately cause an increase of tension in neighboring tendons, this can cause them to progressively fail [18]. Hence, the breakage of one tendon could lead to failure of the entire structure, causing loss of life, energy resources and environmental disaster.

## 2.2. Existing solutions for reducing snap load

Because of the damaging impact that snap loading has on the service life of TLP tendons, several solutions have been proposed over the years. Due to the hydrodynamic force on the submerged structure that opposes the effect of waves on buoyancy, optimization of the ratio between horizontal members (pontoons) and vertical members (columns) can minimize the changes in cable tension [19].

The use of multiple tuned mass dampers (TMD) deployed in series is shown to be effective in controlling surge response of the platform. However, this passive control device only proves its efficiency in hostile sea states and when being used in groups [20]. Another research studying the implementation of TMD inside hull columns also showed its effectiveness in suppressing heave motion [21]. Vibration absorber technology uses caissons that house a column of water as variable mass, an air chamber providing the stiffness and an orifice plate for damping [22]. Platform's heave motion is reduced by the out-of-phase pressure variation in the air chamber against environmental forces. This system can be tuned to change the response frequency characteristics of the TLP to avoid resonance in different sea conditions that can cause extreme stress on the structure. Positive test results of the model yield promising applications of this technology for improving tendon fatigue life.

A different approach to the problem is improving hydrodynamic performance of the hull by automated parametric design [23]. The automated process utilized algorithms to generate concepts and then optimized these designs based on analysis and important parameters. With the use of computer power, this method can create more alternative designs and study them quicker, thus optimal hull shapes can be found faster than conventional design process.

Highly deformable elements as shock absorber were found to significantly mitigate the impact load in cables [16]. A common measure against snap load is increasing pre-tension in tendons to maintain positive tension in the cables at all time, especially in aggressive sea states, this helps to avoid the slackness that can lead to snap load when the tendons are taut again [15]. So far, this has been the most commonly applied solution in the oil and gas industry although it comes with some disadvantages. High pre-tension requires more water displacement of the hull, leading to more hydrodynamic drag of the platform, and requires bigger tendons to bear the axial load [11]. More static tension force also

means less loading capacity of the platform, since it has to be lightweight enough to maintain the excess buoyancy [24], this makes the TLP fall short compared to other types of floating platforms when it comes to storage capacity per given size.

From the literature review, it is found that snap loading is primarily the result of sea wave motions and it has significant impacts on the safety of the TLP and its operations. Increasing pre-tension in tendons is the main solution to avoid snap loading as a part of the platform design but engineers have always been looking for better alternatives. Many scientific journals have proposed different ways to restrict and dampen platform's motions to keep it stable during dangerous sea states. Although these findings do not directly focus on relieving the snap load in the tendons, they provide promising designs that can inspire a soft-seating mechanism for the anchor connector.

### **2.3. Project motivations**

Although the effect of snap load for TLP tendons has been acknowledged and many solutions have been put forward, they mostly focus on the hull design or mechanisms placed inside the hull to oppose the vibration of TLP due to sea waves. The implementation of a shock absorbing device placed directly at the end of tendons has not been fully studied yet. This concept focuses on relieving the high and sudden stress in this critical component of the TLP and it also has the potential to lower pre-tension load while avoiding the slack-taut condition. Therefore, several concepts for the soft-seating mechanism are postulated in this research project. A concept is then selected according to a range of suitability with the requirements and criteria. A detail design is then conducted to develop it to completion. New materials such as flexible synthetic cables are also considered for tendons but the proven and commonly used hollow steel tubulars are still preferred.

### 3. Theoretical background

#### 3.1. Wave theories

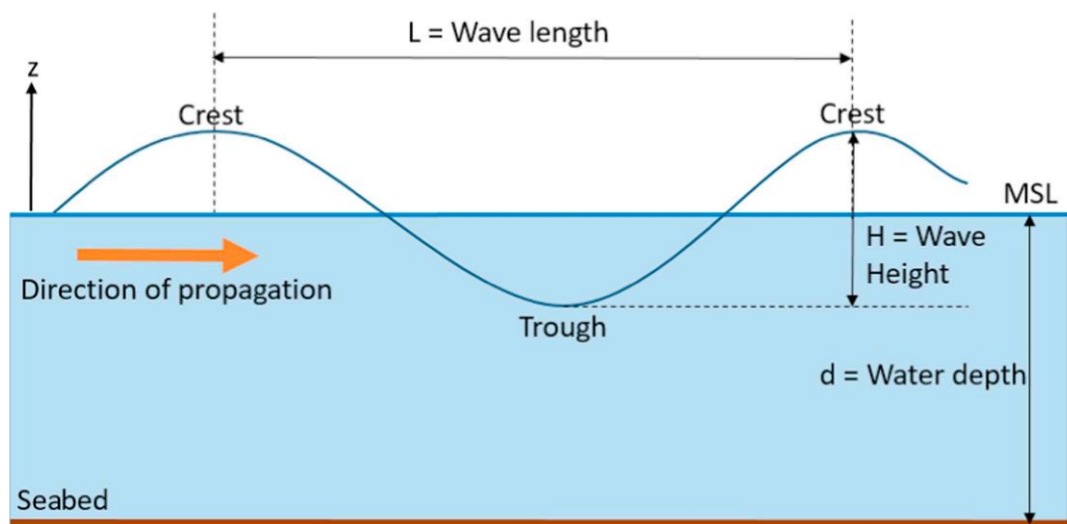
Sea waves is a complicated phenomenon. Although it looks like a periodic motion, the waves come with variations in height and frequency, their shape also change from time to time. To study the impact of waves on ships and offshore structures, mathematical models are often developed. There are several ways to model sea waves but the most widely used method is Airy's theory, also known as the theory of linear waves [25]. This theory assumes harmonic plane waves with linearity between wave height and kinematic quantities [26]. The modelled sea wave using Airy's theory basically has a sinusoidal profile with the following parameters: wave height (H), length (L) and period (T). However, this simplified wave model is best applied for deep water conditions:  $\frac{d}{gT^2} > 0.8$ , where  $d$  is water depth and  $T$  is wave period. For shallower waters where the wave motions can be affected by the sea bed, Stokes theory is more suitable.

In Airy's theory, the wave profile  $\eta$  is defined as:

$$\eta(x,t) = \frac{H}{2} \cos(kx - \omega t)$$

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with  $k, \omega, H$  are wave number, angular frequency, and wave height, respectively.



**Figure 8:** Wave model and parameters in Airy's theory [26]

Although Airy's linear wave theory is not the best to represent the complex behavior of sea waves with its simple sinusoidal harmonic form, it is the most straightforward and easy to use method. To simplify the calculations of wave-induced loadings on TLP in this research, Airy's theory of harmonic wave will be used.

### 3.2. Dynamic response of TLP

As a three-dimensional structure, a TLP has 6 degrees of freedom. An important characteristic of TLP is the discrete range of response frequency between vertical and lateral motions. Due to high axial stiffness of steel tendons, natural periods of heave, pitch and yaw are between 2s and 4s, which is lower than wave period. On the other hand, natural periods of surge, sway and yaw are 80s to 120s, which is well above wave period, thanks to the low horizontal stiffness [27]. This helps the TLP to avoid resonance with wave motion but the low damping in high frequencies leaves it vulnerable to excitation by non-linear forces.

For each degree of freedom, this general equation of motion can be applied [28]:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f[x(t), \dot{x}(t), \ddot{x}(t), t]$$

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This is the equation for forced vibrations, where  $x$  is displacement in any kind of motion,  $m$  is mass of the platform,  $c$  is structural damping coefficient,  $k$  is structural stiffness, and  $f[x(t), \dot{x}(t), \ddot{x}(t), t]$  is the force induced by sea waves. The complete analysis of TLP response would be a set of six equations with one equation for each degree of freedom, such a mathematical model is very difficult to solve analytically. In addition, the function of forces by sea waves are usually obtained by experimental data [28], so the solutions for this kind of mathematical problem are mostly found by numerical methods.

Hydrodynamic forces resulted from the interaction between the TLP structure and sea currents under the waves are often complicated to simulate and formularize, they also depend on the specific geometry of the hull. Hence, the main factor to be considered here will be the force resulted from the change of platform buoyancy, caused by the fluctuation in submerged volumes due to wave motions. Because the force of buoyancy and TLP weight both act in the vertical orientation, the analysis in this thesis will focus on TLP response in this direction.

## 4. Input parameters and product requirements

### 4.1. Environmental parameters for TLP

The inherent job of offshore engineering is to create manmade structures that deal with challenges from nature. Therefore, we need to collect relevant environmental data from the area of interest, that are characteristics of sea waves in the selected regions.

TLP comes in a variety of designs, water depth and operating functions. The TLP shows its advantage in water depth beginning from 150m and can operate in the depth up to 1000m, the “Big Foot” is currently the deepest TLP with over 1500m between its water line and the sea bed [29]. The deeper the water is, the more challenges for TLP designing teams to face, especially with its mooring system because the tendons will be subjected to more extreme loads. For very deep to ultra-deep waters, the feasibility of using TLP has to be carefully considered with weather conditions in the area since rough seas will be risky for this type of platform. Therefore, the FPSOs (Floating Production, Storage and Offloading system) are more commonly used in the fields with depth exceeding 1000m and has hostile weather conditions.

It is reasonable to choose TLPs installed in oil and gas fields of moderate depths between 150m and 500m as the subject for the new soft-seating anchor connector design. The potential site of choice is Ca Voi Xanh (Blue Whale) field, situated in block 118 of the South China Sea. Located around 100km offshore central Vietnam at the depth of 242m [30], this field is a suitable location to install TLP as production platform because it has considerable depth and seasonal storms usually sweep through this region. These conditions make snap load failure of tendons a perceptible risk.

The detailed data about sea conditions in South China Sea, particularly the regions near the coast of Vietnam, is rare in published journals. The mean significant wave height is 1.75m in the collected data in year 2000 [31]. Extreme wave height in 10-year return period was estimated to be 8m in average, this was a forecast based on previous wave simulation [32]. The wave data provided in a marine journal shows that the mean wave period in the area around this field is 5.50s to 5.75s, the peak period is up to 7.50s in the winter months [33]. These data also reveal the proportional relationship between wave period and wave height: bigger waves take more time to complete a cycle.



Two separate studies on seafloor sediments in the northern slope and south east region of the South China Sea both find that the soil type is primarily fine-grained sand and silty clay [34], [35]. The type of soil determines which types of anchors to be used and thus has an influence on the design as well.

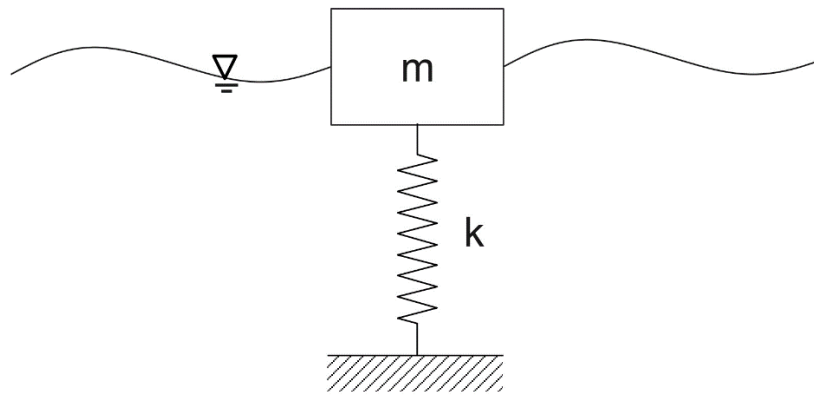
Platform mass can vary greatly depending on production capacity but it is about 40,000 metric tons in average and its water displacement is around 1.5 times its gross weight. Because tendon pre-tension is the difference between platform displacement and its dead weight, the larger a TLP is, the more static axial load is in the cables and anchor. This leads to the need for a bigger snap load absorbing system if it is also designed to withstand the pre-tension load together with the rest of the mooring system.

## 4.2. Requirements for new design

The new soft-seating connector must be able to alleviate the impact force from snap load for tendons by absorbing or dissipating the energy of the impact. The design also aims to reduce the pre-tension in order to lower static axial loading from the current values of rigid anchor connections. This does not only relieve the high constant axial stress on tendons, which is one of the factors in tendon failure, but also helps accommodate for more load capacity of the TLP over a given size. These requirements should be achieved while keeping the natural frequencies of both vertical and horizontal responses away from the range of common wave frequency because resonance with environmental forces should be avoided as much as possible. This means the new design needs to be able to absorb the sudden increase of loading on tendons while maintaining sufficient overall stiffness of the mooring system to keep the platform from excessive motions caused by resonance.

A TLP using rigid tendon connection can be modeled as a vibration system shown in Figure 9, in which the platform with mass  $m$ , floating on the water and held down by tendons with stiffness constant  $k$ . The system oscillates with angular frequency  $\omega$ , given by:

$$\omega = \sqrt{\frac{k}{m}}$$



**Figure 9:** Model of a TLP with conventional rigid tendon connector

This equation shows that the frequency of vibration in a system is proportional to its stiffness constant. The TLP has high frequency of vertical oscillation thanks to great axial stiffness of solid steel tubular tendons. The soft-seating connector is intended to be inserted into the forementioned system so that it can absorb the snap load by the use of spring and damper. If the stiffness is too high, the spring will not be able to absorb much impact but rather passes that force onto the tendons. On the other hand, a suspension system with high absorption will allow for a lot of vertical movements, which may cause discomfort for the crew onboard a TLP, and has lower response frequency in heave direction. Being ‘soft’ enough to absorb the impact while being ‘hard’ enough to avoid resonance and excessive motions is hard to achieve simultaneously. Therefore, the ability to adjust critical system parameters such as stiffness and damping coefficient is desired, as different sea states require different values of these parameters for optimal structural response.

Due to the soft clay formation on the sea bed in the chosen area of South China Sea, suction pile is a suitable type of anchor for taut and vertical mooring configuration, it is also widely used for existing TLPs. A study on failure of suction pile anchor in soft clay using model testing concluded that cyclic loading leads to failure faster than static loading [36]. Hence, the placement or mechanism of the new anchor connector design needs to avoid transferring vibrations to the anchor piles.

Compatibility with the platform designs of existing TLPs is also a desired aspect although it is not mandatory. The popular steel tubular tendons should be able to work with it but alternative materials such as synthetic wire ropes for cables should also be considered due to their high strength and light weight.

## **5. Design process**

### **5.1. Design methodology**

The aim of this thesis project is to create a solution for the problem of reducing snap load in TLP tendons and improve it to develop a useful product that is relevant to current demands in the industry. To achieve this, the product will be engineered to meet the design specifications while being as practical as possible. Several concepts are proposed, the design that satisfies the criteria the best is then put forward to narrow down the list. The selected solution is optimized by inheriting the assets of others if possible. Further comprehensive calculations are done to find necessary parameters for manufacturing.

For some limitations, the physical model of the final design cannot be made and thus lab testing is not a part of this research project. The engineering design process in the project follows three major stages:

#### **Stage I:**

In this stage, environmental influences, basic platform and mooring properties are considered. With the identified requirements for the product, the design specifications are defined to make sure the new design can perform as intended. Concepts are then developed through brainstorming and demonstrated in sketched illustrations to show the main mechanisms and their basic working principles.

#### **Stage II:**

The conceptual designs in the previous stage are graded using Pugh concept selection method. Selection criteria are chosen from important design requirements. The Pugh matrix is generated with criteria of different weights and the concepts are evaluated for each criterion based on how well it will perform in that aspect. A concept with the highest overall score is selected for the detail design. The strength of other designs could be used to influence the optimization for the chosen one in the areas where it still lacks.

#### **Stage III:**

At the detail design stage, the selected concept in Stage II is studied in detail to produce a practical design, a mathematical model of the system is analyzed to find the key design parameters for the product. The dimensions of components are then calculated based on

these parameters and the chosen materials. The CAD model is built in order to show more refined mechanical features and dimensions of the product assembly and components.

## **5.2. Stage I**

Environmental factors, required functions and potential issues of the product are studied to identify the boundaries for the design. From these data, we define the product design specifications, which will act as guidelines for the design to follow in order to develop a successful product. Initial concepts that aim to solve the problem are created, they are generated on the idea of a mechanism that allows displacements within itself to absorb the high impact load and also dampen it. Some of them are inspired by equivalent systems in industrial applications or new ideas in reviewed literatures that are also set out to solve similar problems.

### **5.2.1. Product Design Specifications**

The Product Design Specification (PDS) is a list of precise and measurable requirements on the desired outcome of the product development, it defines the boundaries within which the solutions to the problem must lie. The product design aspects follow the PDS to meet its requirements and achieve a successful product. The PDS takes into account the customer needs and define measurable specifications without forcing any specific design approach to meet those requirements. Therefore, the PDS is established before the concept generation stage to steer the ideas to the right direction while avoiding the restriction of creativity, in order to come up with as many relevant designs as possible.

The aspects of the PDS are identified as follows:

#### **❖ Performance**

- Eliminate all snap loading in tendons for mean significant wave height of 1.75m
- Maintain response frequency of TLP in vertical and horizontal directions as regular rigid anchor connector system
- Reduce static pre-tension and fluctuations in dynamic tension in tendons

#### ❖ **Product life cycle**

- Expected service life of 20 years
- Yearly maintenance for inspection and corrosion protection
- Product end-of-life recycling requires no special process

#### ❖ **Working environment**

- Fully submerged in sea water (depth ranging from 0m to 242m)
- Ambient pressure: 1 bar to 25 bar (pressure at depth from 0m to 242m)
- Ambient temperature: 10°C (at 242m deep) to 25°C (at sea surface) [37]
- Subjected to marine growth on the outside

#### ❖ **Market identification**

- Potential customers are petroleum exploration and production companies such as Shell, Chevron, etc. or companies that build offshore platforms
- Competing products are existing tendon top and bottom connector assemblies from oilfield equipment companies such as Baker Hughes, Oil States Industries

#### ❖ **Manufacturing**

- Impact absorbing components such as spring and damper are off-the-shelf parts
- Support structures and connecting components are machined in house
- Anti-corrosion coating and cathodic protection are required for exterior surfaces

#### ❖ **Testing**

- Test product functions and reliability by applying simulated loads that exceed its designed capacity
- Perform pressure tests with applied external pressure to be higher than water pressure at designed depth

#### ❖ **Physical properties**

- Size will be able to fit existing designs of foundation templates and anchor piles or tendon porches
- Weight is limited to cargo capacity of installing cranes and transport ships

#### ❖ **Materials**

- High-strength steel alloys for main components, corrosion-resistant steel for the housing
- Rubber and elastomers for sealing or shock absorbers
- Cathodic protection anodes will be made of zinc, aluminium or magnesium [38]

❖ **Health, Safety and Environment (HSE)**

- Comply with mechanical, oil and gas industry standards such as ASME, API, ISO
- Has fail-safe mechanisms in case of system failure
- Avoid leakage of lubrication and working fluids into the sea
- Use non-toxic coating paints

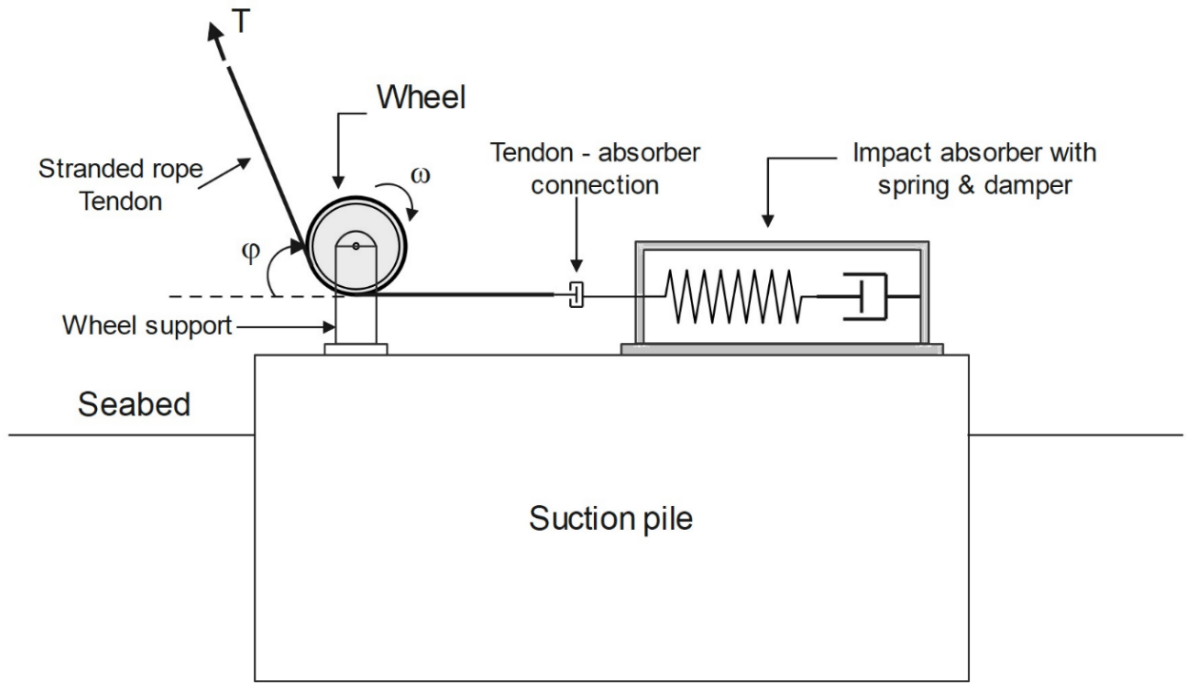
❖ **Shipping and Installation**

- Transported to site by standard vessels that are used for installing subsea systems
- Impact absorbing elements come preinstalled to anchor piles or TLP hull; final connections to tendons are made on site
- Remotely operated vehicles (ROV) may be required to aid the installation

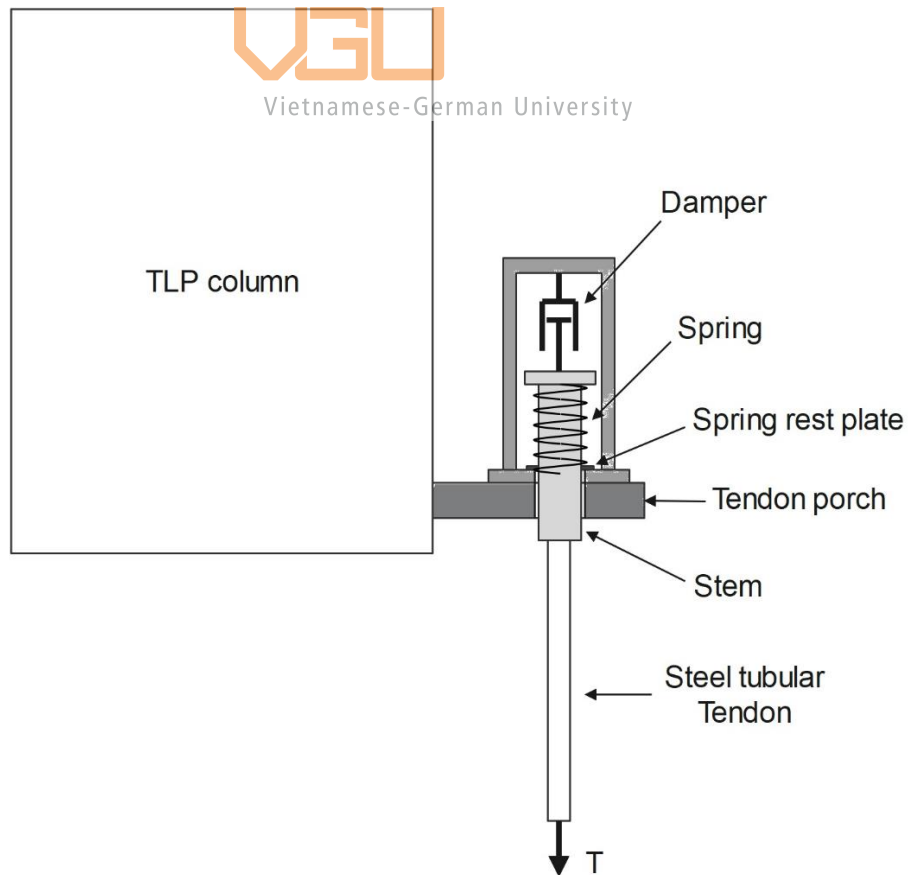
### 5.2.2. Design concepts

The conceptual designs provide a layout of the components, the way they interact with each other and handle the loading from the tendon. They display a variety of mechanisms and implemented locations. Eight concepts were generated, they are described briefly and shown in sketches (Figure I to Figure VIII) for visual illustration:

- **Concept 1:** The impact absorber sits inside a housing that is welded to the top of the anchor pile but in parallel with the sea bed. The tendons are converted from vertical to horizontal direction by a wheel and fed into the device. This design can avoid direct transmission of vibration to the anchors but it requires tendon that can bend quite easily such as stranded rope. The angle  $\phi$  between the exiting cable and horizontal plane should be smaller than  $90^\circ$  to avoid excessive bending.
- **Concept 2:** The device is placed on modified tendon porches that are welded to the side of the TLP hull columns while the anchor connection still remains a rigid one. The tendon top is connected to a stem, which transfers the load to a spring and a damper in vertical arrangement. This concept allows for easier access to the device for adjustment and maintenance.

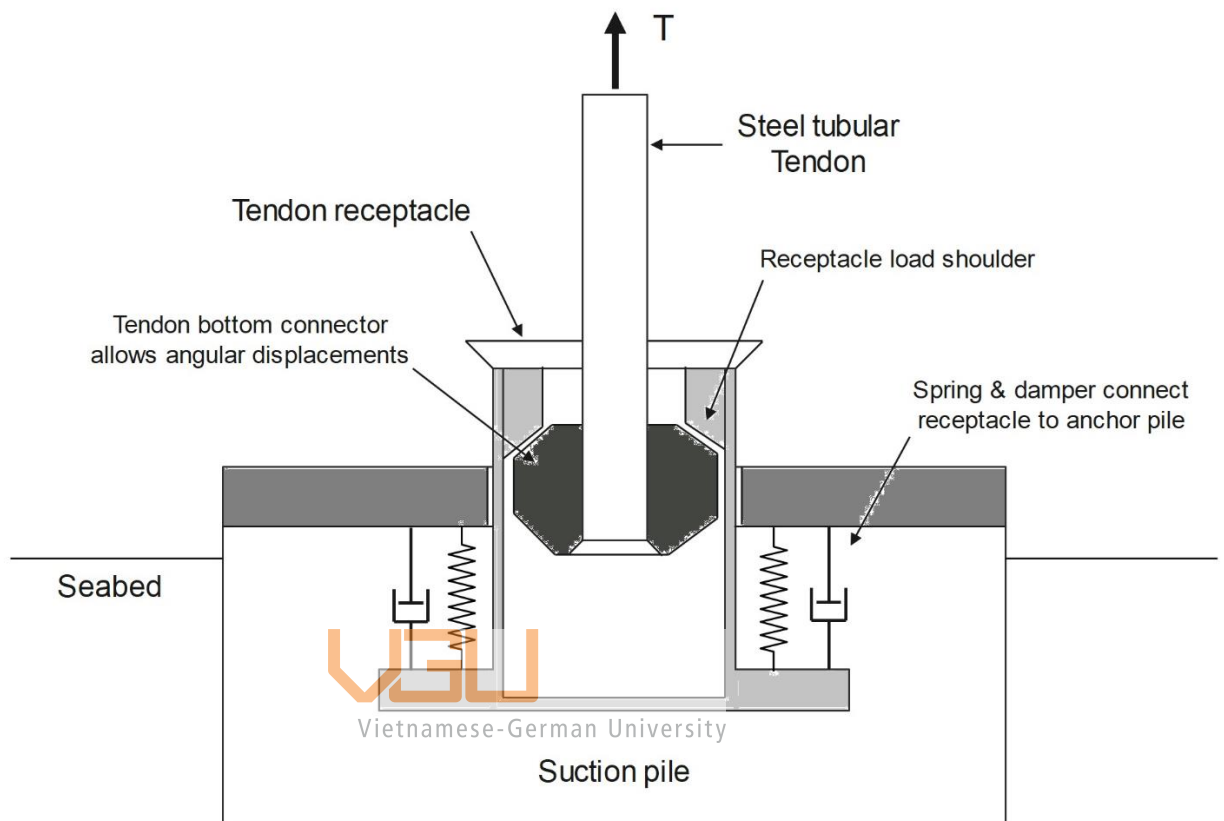


**Figure I: Concept 1**



**Figure II: Concept 2**

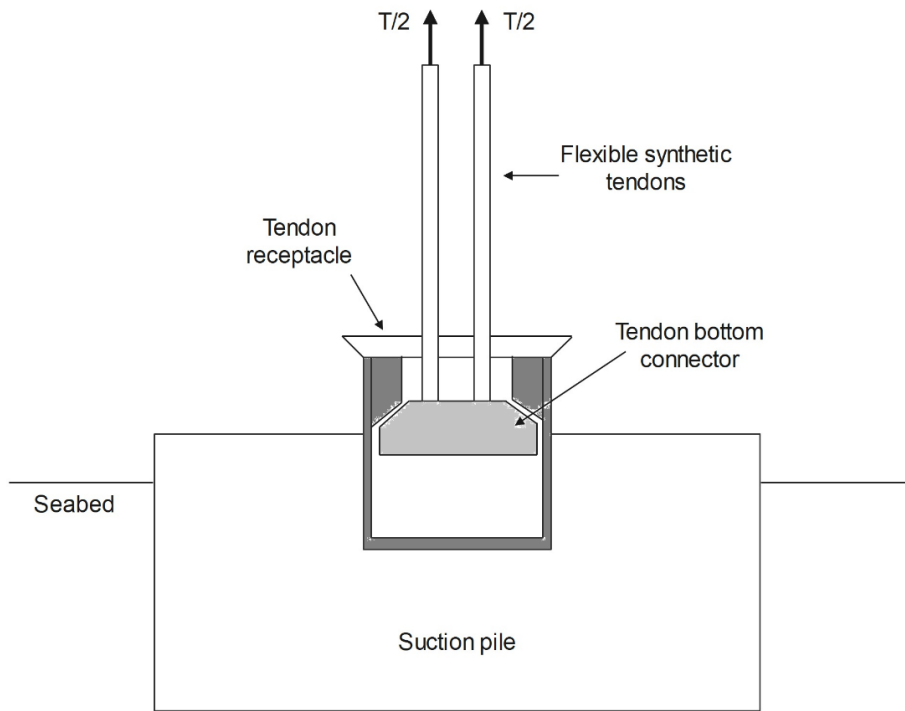
- **Concept 3:** The tendon-anchor connection is still rigid while springs and dampers are placed between the receptacle and the top of suction pile. Impact from snap loading is transferred from tendon to the load shoulder and absorbed at the suspension system attached to the base of the receptacle unit.



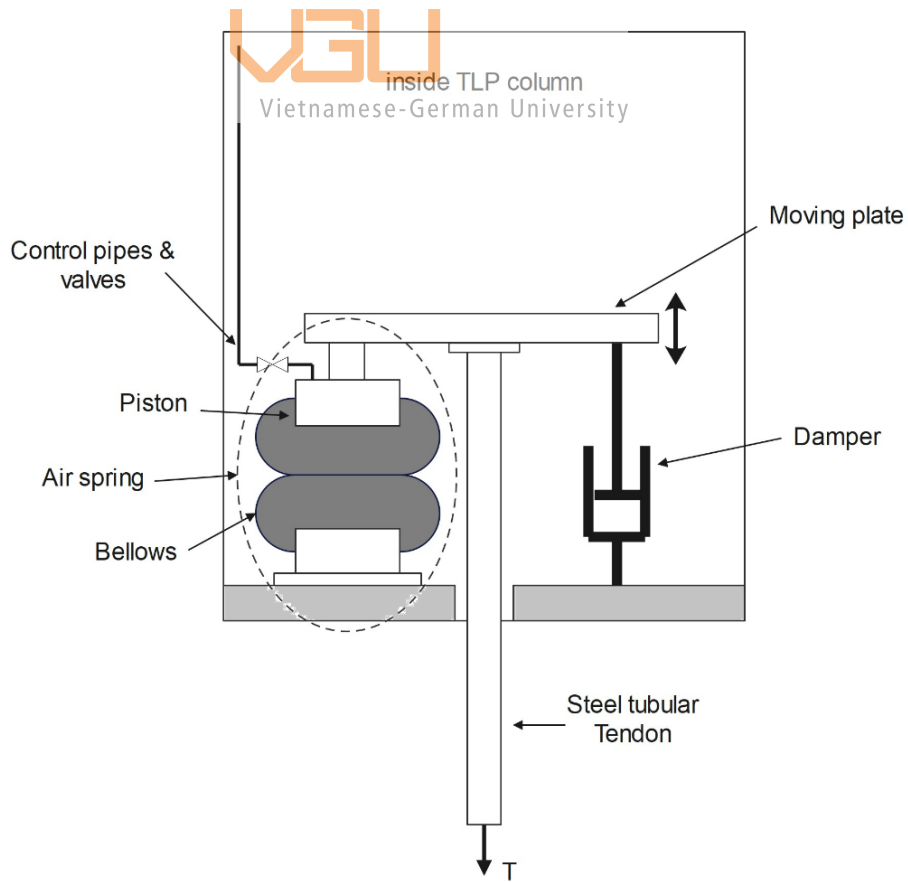
**Figure III:** Concept 3

- **Concept 4:** Flexible synthetic tendons are utilized in this concept. Shock absorption and tension restoration solely rely on the flexibility of the cables. This is the simplest design because it requires no additional mechanisms but the tendons themselves do the job.
- **Concept 5:** This mechanism uses air spring for restoring tension and hydraulic damper as shock absorber. Air spring uses high-pressure gas to create stiffness, it has great ability to adjust the spring constant and can also have its own damping effect by the use of auxiliary chamber and pipe flow restriction. But the high capacity for a big TLP requires a separate shock absorber, so a hydraulic damper is used. Since the air spring and damper are placed on either side of the tendon, there is a risk of imbalance, resulting to bending of tendons, if they have different responding speed.



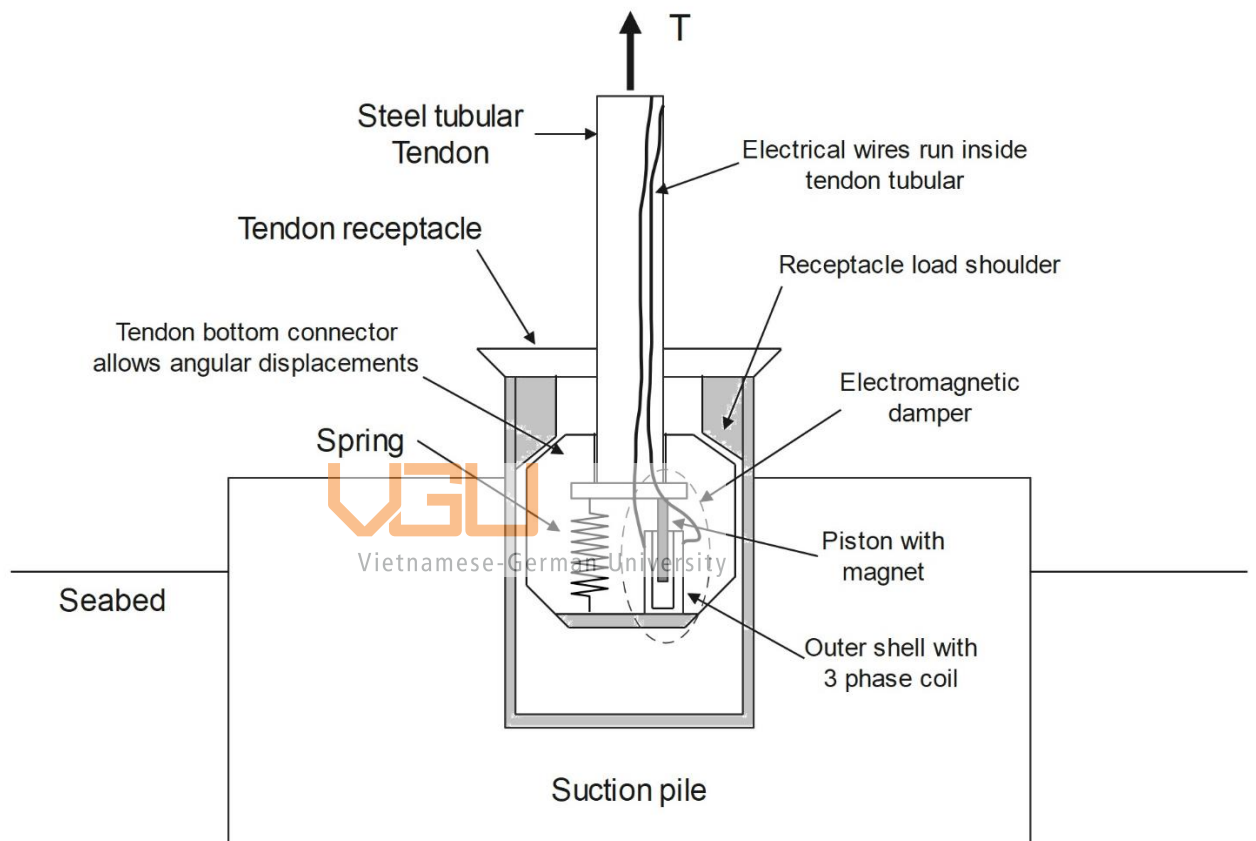


**Figure IV: Concept 4**



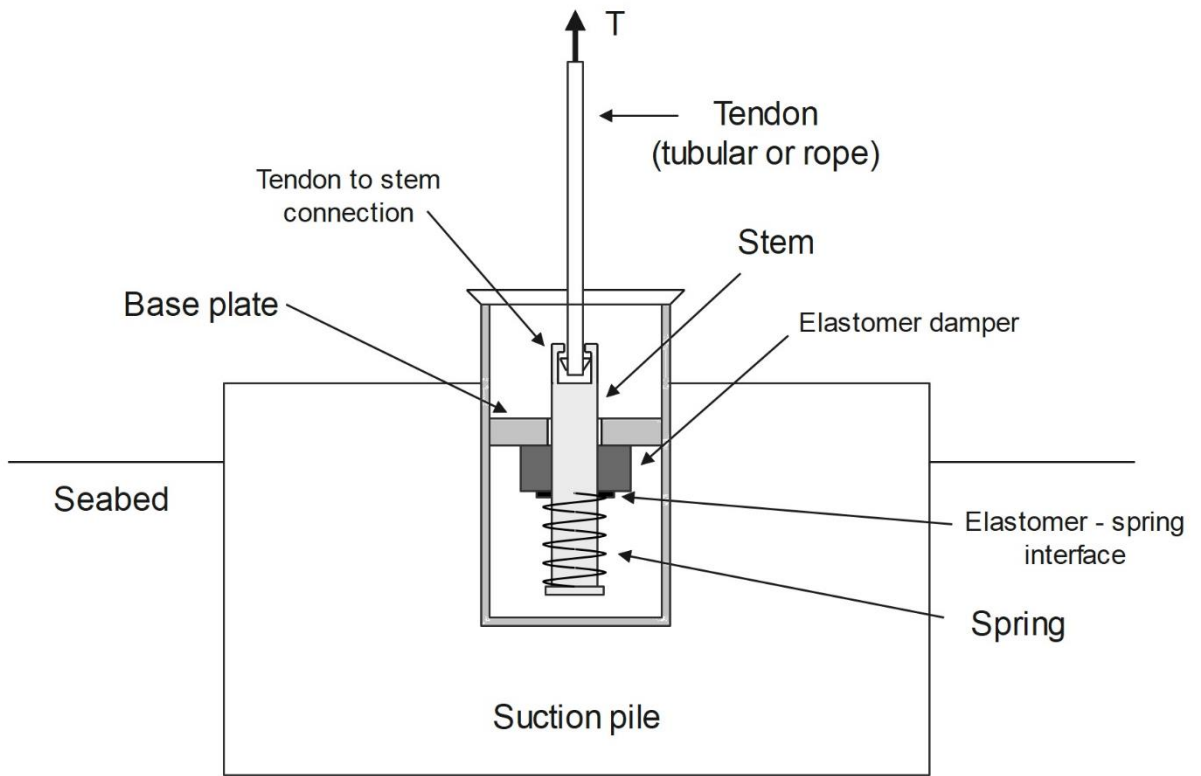
**Figure V: Concept 5**

- Concept 6:** Regular mechanical spring is used with electromagnetic damper. The damper uses electrical currents in copper coils to create magnetic field that interacts with a permanent magnet on the piston to provide resisting force for dampening vibrations. This system has faster response and overcomes fluid related issues of viscous dampers so it has already been used in car suspension. The spring and damper are preinstalled inside tendon bottom connector, which has rigid contact with the receptacle.



**Figure VI:** Concept 6

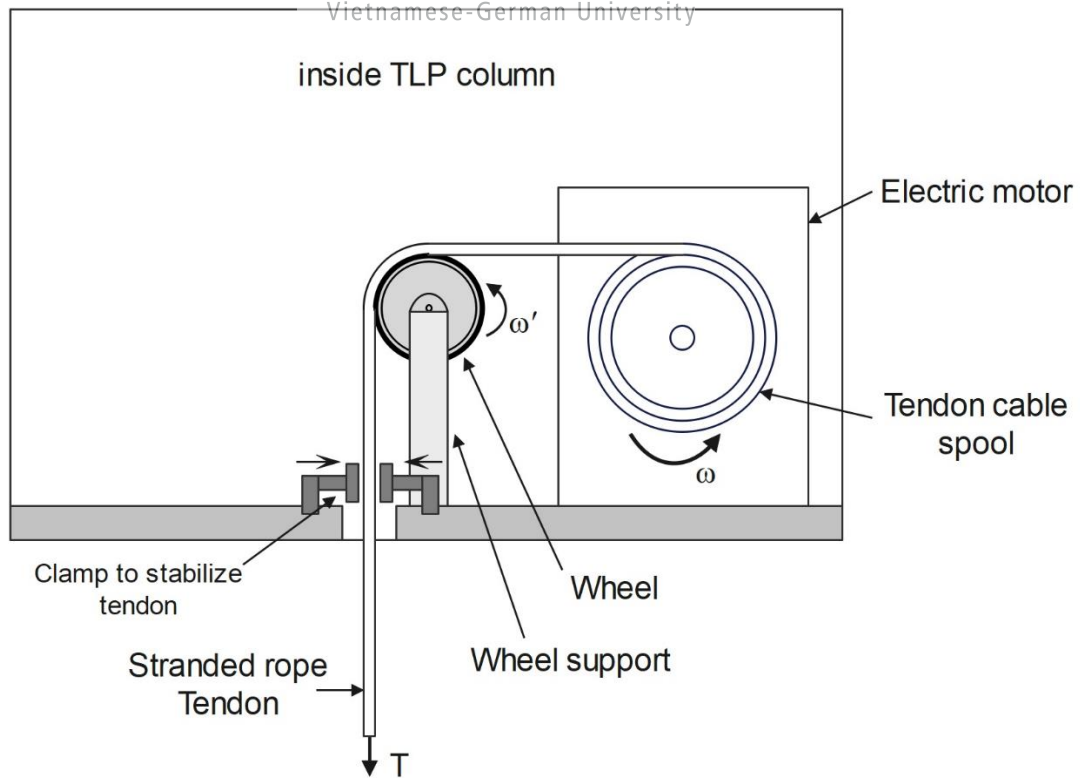
- Concept 7:** In this concept, helical spring is combined with the damper made of elastomer block to absorb impact. This has the advantages of being simple and reliable with very few moving parts, the elastomer provides quicker response than hydraulic dampers. The limited damping ability is the main drawback of this design.



**Figure VII: Concept 7**



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**Figure VIII: Concept 8**

- **Concept 8:** This device does not employ the regular spring-damper configuration but uses an electric motor that actively changes the length of tendons in order to maintain tension. The tension  $T$  is translated to torque loading on the motor shaft. A sensor will detect variations in torque loading on the shaft and prompt the motor to release or tighten the cable spool to maintain the tension. When the sea is calm, a clamp holds the tendon to relieve the motor from constant loading.

### 5.3. Stage II

Pugh concept selection method is a commonly used technique, invented by Stuart Pugh (a British design engineer, who is well known for being the inventor of the Total Design methodology [39]). This technique enables engineers to narrow down the choice of design concepts and highlight the most promising one by the use of a credit weighting structure. Evaluation criteria are influenced by product design specifications and requirements, criteria that are considered more important have higher weight but the weights of all criteria add up to 100%. The concepts are rated, using a scale with the following numerical values that represent how well they will perform in each criterion:

- 0 = Unsatisfactory
- 1 = Tolerable
- 2 = Adequate
- 3 = Good
- 4 = Very good

The rated scores are then multiplied with their respective weight and summed up for the overall rating of each concept. The concept that arrives with the highest score is chosen for the embodiment design and detail design.

#### 5.3.1. Design selection criteria

Selection criteria are qualities for finding the best overall designs, they reflect the important requirements for a good product. The level of significance of each criterion is depicted by its weight as a fraction of 100, the weights of all criteria have a total of 100%. The number of criteria is controlled so that we take all of the crucial requirements into

account while being able to distribute their weights with substantial difference to highlight the decisive factors from the rest. There are 8 criteria, defined as follows:

- **Stability:**

A design has good stability when it can maintain the balance in the mechanism and control the vibrations well when subjected to loading and movements. Mechanical systems that have fewer moving parts tend to be more robust and stable. Stability is a priority for this mechanical design so it has the second-highest weight of 18%.

- **Ease of installation and maintenance:**

The choice of mechanism and its location with respect to the TLP and mooring system should promote simple installation and easy maintenance. This is an important characteristic in the design of this product because it is the key for cutting down service life costs. Oil and gas platforms are usually in service for several decades, thus systems that have cheap maintenance procedures tend to be more competitive in their overall costs. A weight of 9% is given to this criterion because it contributes to a cheaper product, although price is not the highest concern.

- **Reliability:**



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The most important aspect of any offshore platform design is the safety of the crew, the expensive equipment onboard the TLP and the integrity of riser system (a conduit of hydrocarbon flow from oil wells to the surface). Failure of the mooring system can lead to catastrophic consequences for human lives and the environment. To ensure the safety, this product needs to have consistent performance and easily predictable behavior. A reliable design requires less maintenance and repair during its service life. Fail-safe mechanisms should be included to avoid critical failures in the event of malfunction. Reliability is a critical aspect with 31% of weight because it ensures safety, the main focus of every offshore or oil and gas project.

- **Compatibility with existing TLP/anchor designs:**

The introduction of an alternative connector between tendons and anchor piles will likely require modifications to the anchors or the TLP hull itself. A system that can be fitted to existing platform or anchor foundation designs will be more preferable than the ones that need a redesign of these structures to accommodate it because the engineering time and costs for such tasks are high. Compatible shape and size of the

connector will allow it to fit into the anchor or TLP hull. This is a desired quality but otherwise uncritical so it is 7% in weight.

- **Ability to absorb impact:**

The main purpose of this product is to avoid snap loading by absorbing and dissipating the energy from the abrupt increase of tension inside TLP tendons. Hence, the design should be efficient in taking the snap load from the tendons to its impact absorber. The importance of this requirement is reflected by the weight of 12%.

- **Speed of restoring tension:**

Another function of this product besides impact absorption is to keep the tendons taut when their dynamic tension drops as a result of the decrease in hull buoyancy at wave trough. The faster it can compensate for the lost tension, the lower risk of tendon becoming slack there will be. This quality contributes to avoiding the snap loading so it shares 10% of weight.

- **Ability to adjust system parameters:**

Sea conditions change from time to time, so flexibility of the system is desired. The platform operators should be able to adjust parameters such as stiffness or damping coefficient to optimize TLP response in each sea state. For example, lowering the stiffness allows more impact absorption but this also reduce response frequency, which can lead to resonance with sea waves. On the other hand, increased stiffness helps to avoid resonance but more stress will be transferred to the tendons. However, flexible parameters raise the complexity of the design while they are optional, so this one has the lowest weight of merely 5%.

- **Wear on tendons:**

The mechanism should interact with the tendons in a way that it does not deteriorate them over time by creating more physical deformations such as bending the tendons or additional friction with its components. Tendons are expensive to replace and they are critical parts of the TLP so wearing them out should be avoided. Nonetheless, steel and synthetic fiber ropes are durable enough to withstand some bending so the weight of 8% is assigned to this one.

### 5.3.2. Concept evaluation

Each concept is given scores that depict how well they perform in an aspect, relatively to other ones. The following observation is made for the pros and cons of the concepts when being evaluated in each design selection criteria:

- Most designs have high stability, except the third and fifth ones because they have spring and damper on two sides that may create imbalance when one side is compressed more than the other.
- With more complex components that require additional pipes or wiring, concepts 5 and 6 are harder to install and perform maintenance.
- Concept 4 is highly compatible to existing TLP systems because it does not require any extra component rather than the tendons themselves, while concept 3, 5 and 8 will require substantial modifications to the TLP hull and anchor.
- The elastomer damper in concept 7 will allow quicker rebound than other designs and restore tension faster.
- The first and last concepts have the tendon cable wrapped around the wheels so they can wear out the cable at the contact points and cause early fatigue due to bending.
- Since the tendon in concept 4 is flexible synthetic cable, it has lower strength than steel, hence lower loading capacity; the elastomer block in concept 7 cannot absorb as much impact as piston-type dampers.
- Concept 5 and 6 utilize the alternative air spring and electromagnetic damper for the purpose of customizing suspension parameters so they stand out from the rest in the ability to alter system parameters.
- All the concepts are created to deliver reliable performance, but concept 2 is the most highly regarded one in this aspect because it is simple and straightforward while still being effective.

### 5.3.3. Final design selection

After the evaluation of the concepts using Pugh method, the design that has the highest total score is concept number 2, it is then selected to proceed in the later design stage. The chosen concept does not outperform others in every single aspect but it rather has the

best overall performance reflected by its top score in total. The detailed Pugh matrix scoring for concept selection is given in Appendix I.

## 5.4. Stage III

### 5.4.1. Mathematical modeling of system

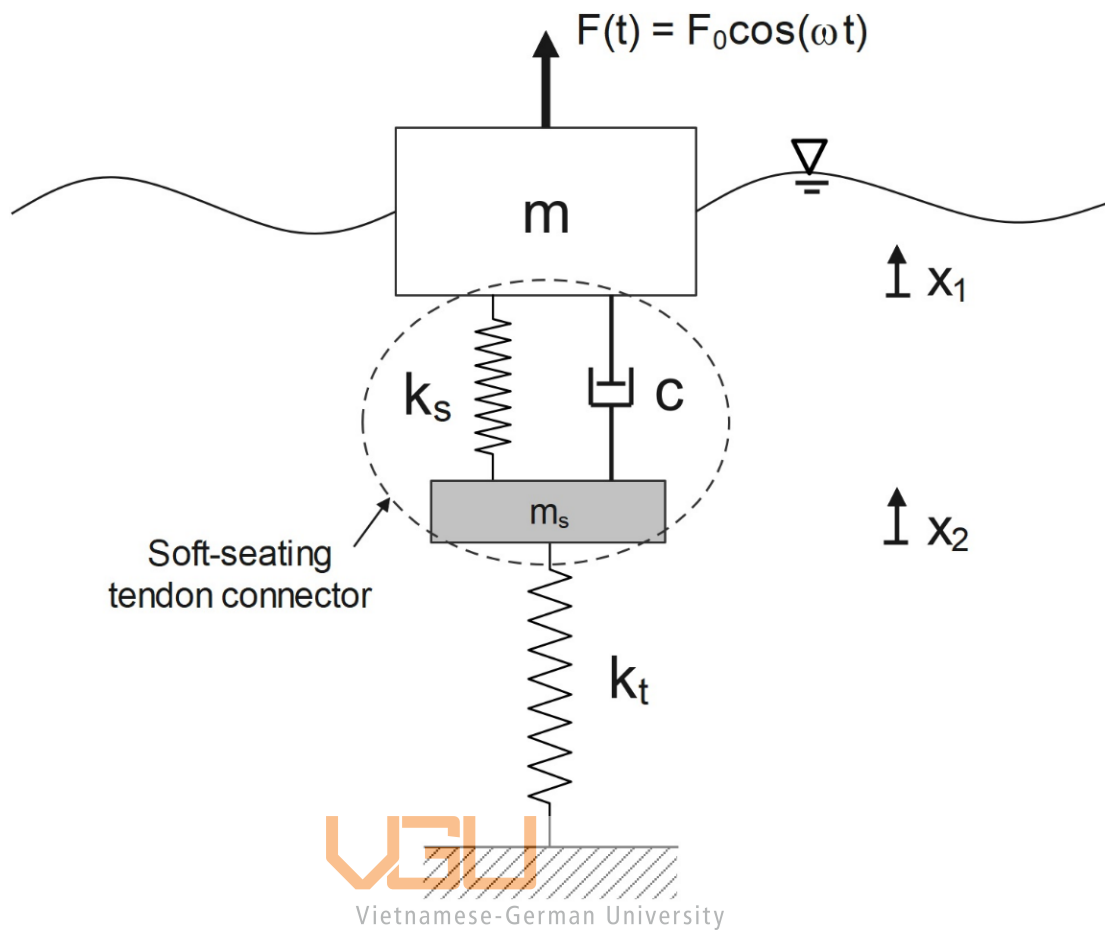
Because the soft-seating connector will be installed on each tendon, the analysis will focus on the dynamics of a single tendon instead of the entire platform. Thus, the main degree of freedom to be studied in this research is heave response, which are vertical motions. Although the surge response in horizontal direction could cause additional stress in the long tendons, it has little effects on the dynamics of this soft-seating connector so only vertical movements are examined.

In the chosen design (concept 2 – Figure II), the suspension system configuration may look as if they consisted of spring and damper in series, because of their vertical arrangement. However, one end of the spring has to be directly connected to the damper in such configuration, while that is not the case in this design: both the spring and damper rest against the stem. The stem will have a rigid connection with the tendon so in fact, the system can be modeled as in Figure 10.

The two coordinates  $x_1$  and  $x_2$  describe the position of the TLP and the amount of tendon elongation, respectively. The coordinates are measured from the equilibrium positions of the platform and the tendon. The soft-seating tendon connector is depicted with spring and damper in parallel, with stiffness  $k_s$  and damping coefficient  $c$ , these are the two design parameters that we need to find; and the suspension system has mass  $m_s$ . Each tendon has axial stiffness  $k_t$ . The mass  $m$  is platform mass divided by the number of tendons. The TLP is excited by a harmonic force  $F(t)$  induced by sea waves, it is a function of time  $t$ , where  $F_0$  is the force amplitude divided by the number of tendons and  $\omega$  is frequency of sea waves.

The tendons are made of thin-walled steel tubulars with the interior filled with air, this helps them to be neutrally buoyant underwater [40], in other words, the weight of each tendon in water is relatively small so it is negligible in this calculation. The mass of each tendon is uniformly distributed along its length so it is irrelevant in the dynamics analysis of a system whose motions concentrate at the top.





**Figure 10:** Model of TLP with soft-seating tendon connector

This model has complex response with 2 degrees of freedom (DOF). If it can be proved that the tendon has large enough axial stiffness that its elongation  $x_2$  is minimal compared to platform movement  $x_1$ , then the whole system can be simplified to have only 1 DOF, which is  $x_1$ . Additionally, the mass of the tendon connector to be designed is likely to be very insignificant compared to the load of TLP mass on each tendon because the platform is many times larger than the tendon connectors and the porches holding them,  $m_s \ll m$ . This means the suspension mass  $m_s$  will not have much effect on the dynamics of the overall system and could be omitted. The detailed parameters of the TLP and its tendons to be investigated in this detailed design are chosen from a journal article that also studies on a TLP model in the South China Sea [2]. This TLP has 4 columns and 4 pontoons, each column is held by 2 tendons on the side porch to make a total of 8 tendons. The relevant parameters of the platform and tendons are presented in Table 1 and Table 2, respectively. The unit “MT” stands for metric ton and “m” is meter. The TLP layouts with additional structural dimensions are shown in Appendix III.

Parameters	Value
Platform mass	32,127 MT
Platform displacement	49,515 MT
Draft	30.5 m
Column diameter	19.5 m
Pontoon height	8.5 m
Pontoon width	8.5 m

**Table 1:** TLP parameters [2]

Parameters	Value
Tendon outside diameter	1.106 m
Tendon wall thickness	0.038 m
Tendon length	375.3 m
Axial stiffness ( $k_t = \frac{EA}{L}$ )	6494.7 MT/m
Top pre-tension	2008 MT

**Table 2:** Tendon parameters [2]

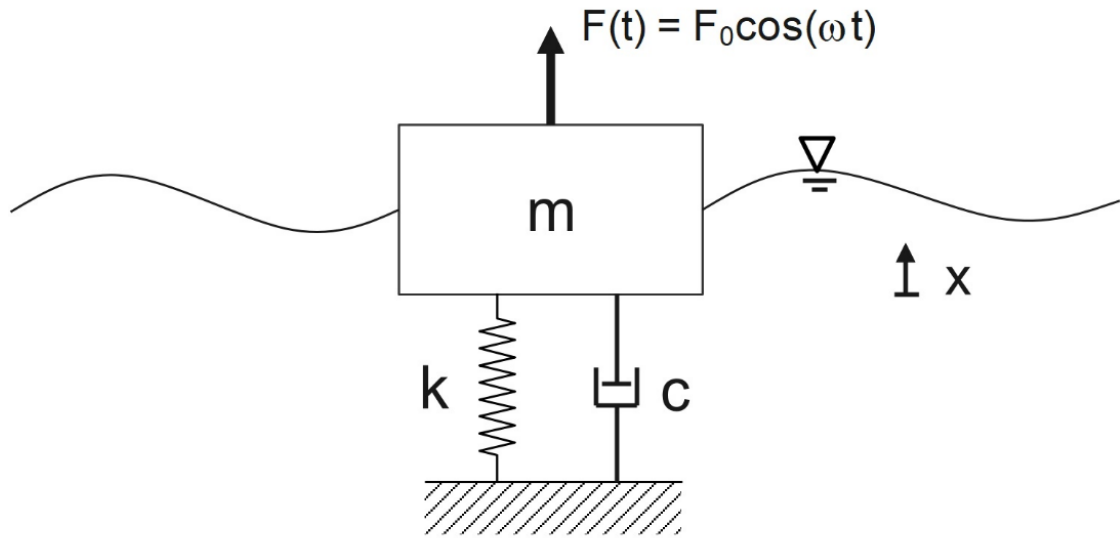
The force  $F(t)$  acting on the TLP is created by the variation in platform buoyancy when different volumes of water are displaced by the column as the waves interact with it. Thus, the force amplitude  $F_0$  is the additional water displacement from equilibrium when the platform is at wave crest, divided by 2 tendons for each column:

$$F_0 = \frac{Ah\rho_w g}{2} \quad (1)$$

where  $A$  is cross-sectional area of a column,  $h$  is wave amplitude, a half of wave height,  $\rho_w$  is density of sea water at the surface, and  $g$  is the gravitational acceleration. The axial stiffness of tendons is given in ton/m so it must be multiplied with the gravitational acceleration  $g$ , accordingly  $k_t = 6.369 \times 10^7 N/m$ . Using the mean significant wave height of 1.75m [31], and surface water density  $\rho_w = 1022 kg/m^3$  in the South China Sea [41], the calculation yields  $F(t)_{max} = F_0 = 1.31 \times 10^6 N$ , at wave crest. If the tendon is directly connected to the platform, without suspension in between, the maximum elongation can be found by  $\Delta l_{max} = \frac{F(t)_{max}}{k_t} = 0.02 m$ . The tendon is stretched by only 2cm over its entire length with the rigid connector, so when it is put

together with the suspension unit which has  $k_s \ll k_t$ , the amount of tendon elongation will be even smaller and hence  $x_2 \ll x_1$ .

With suspension mass  $m_s$  and tendon elongation  $x_2$  confirmed to be minimal, the TLP system can be reduced to the 1 DOF model in Figure 11.



**Figure 11:** Simplified system model  
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In this simplified model, platform position is described by the coordinate  $x$ , which is corresponding to  $x_1$  in the previous model. Because  $m_s$  is now neglected, the tendon and suspension spring can be modelled as 2 springs in series, so the equivalent stiffness  $k$  of the system is:

$$\frac{1}{k} = \frac{1}{k_t} + \frac{1}{k_s} \quad (2)$$

Using Newton's Second Law, the system equation of motion is obtained as:

$$m\ddot{x} = -c\dot{x} - kx + F_0 \cos(\omega t) \rightarrow m\ddot{x} + c\dot{x} + kx = F_0 \cos(\omega t) \quad (3)$$

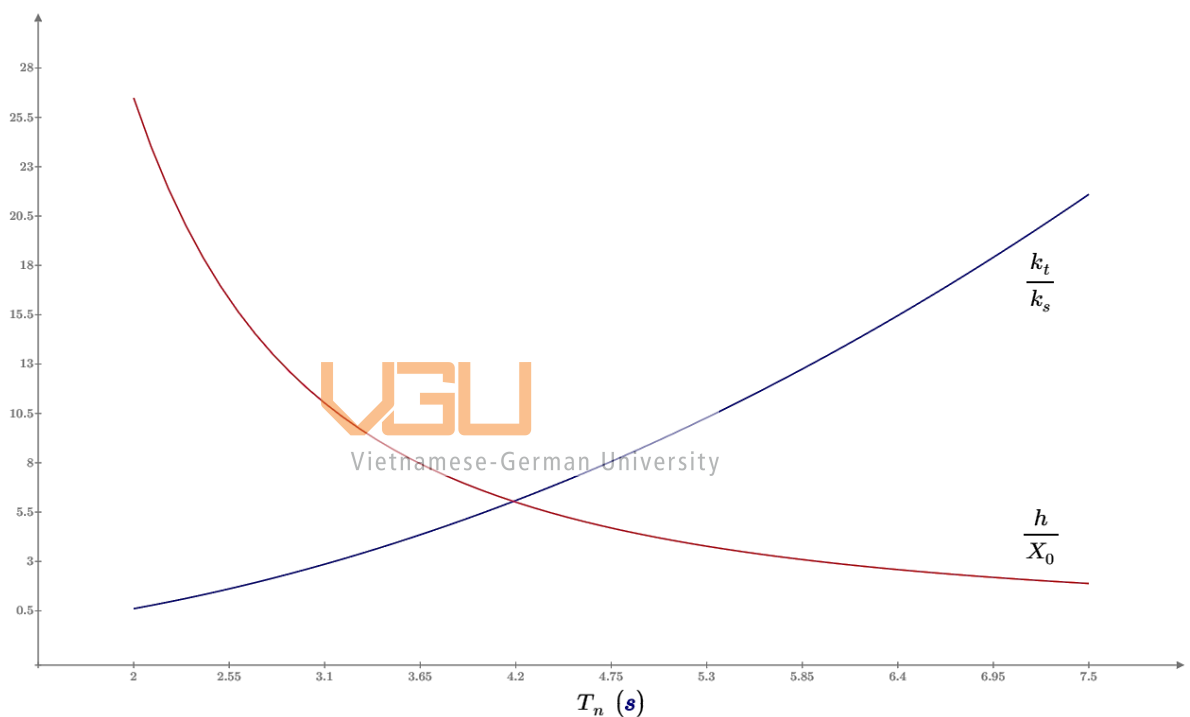
The above equation becomes:

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2x = \omega_n^2X_0 \cos(\omega t) \quad (4)$$

if the following parameters are substituted into the equation:

$$\omega_n = \sqrt{\frac{k}{m}} = \frac{2\pi}{T_n}, \quad \zeta = \frac{c}{2\sqrt{mk}}, \quad X_0 = \frac{F_0}{k}$$

where  $\omega_n$  is the natural frequency of the vibration system,  $\zeta$  is the damping ratio, and  $X_0$  is the amplitude of the vibration. The equivalent stiffness  $k$  of the system can be calculated when a proper value of the natural vibration period  $T_n$  is chosen. Wave data in the area around Ca Voi Xanh field gives the mean period of 5.75s [33], so the natural system period should be out of range of this value in order to avoid intense excitation by resonance with sea waves. The spring rate  $k_s$  can be calculated from the formula (2) after system stiffness has been found. The ratios of tendon stiffness to suspension spring rate ( $k_t/k_s$ ) and wave amplitude to platform movement amplitude ( $h/X_0$ ) are also considered to determine the most suitable value for  $T_n$ , they are presented in Figure 12.



**Figure 12:** Plotted values of  $k_t/k_s$  and  $h/X_0$

The ratio  $k_t/k_s$  must be reasonably large, which is close to 10, because the spring needs to be “soft” enough to absorb the snap load when it occurs, the helical spring rate is also practically much lower than axial stiffness of steel tubular tendon. The amplitude of platform movement should be several times smaller than wave amplitude for the comfort of its crew and to avoid breaking the production risers,  $h/X_0$  can be around 4 because the wave amplitude is not too big in the first place. In view of the favorable values for these two ratios and the wave period to be avoided, the system vibration period is decided to be:  $T_n = 4.5s$ , at which  $k_t/k_s = 7.13$  and  $h/X_0 = 5.23$ . The equivalent system stiffness, suspension spring rate, and amplitude of TLP movement can be obtained as:

$$k = 7.829 \times 10^6 \text{ N/m}, \quad k_s = 8.926 \times 10^6 \text{ N/m}, \quad X_0 = 0.167 \text{ m}.$$

When the waves travel through the TLP columns at the height of 1.75m, the platform only moves up and down by a total distance of  $2X_0 = 0.334 \text{ m}$ , this is an acceptable amount of vertical displacement for a big floating structure like the TLP.

To find the damping coefficient  $c$ , a value for the damping ratio  $\zeta$  is chosen. A critically damped system, where  $\zeta = 1$ , can suppress all the vibrations created by wave excitation but such high damping is difficult to achieve in a heavy system as this one. Therefore, the damping ratio is decided to be 0.7, this is also a common value for suspension systems that provide high comfort and stability. Then, the value for damping coefficient  $c$  can be acquired as:

$$c = 2\zeta\sqrt{mk} = 7.85 \times 10^6 \text{ Ns/m}.$$

#### 5.4.2. Calculation of spring dimensions

After obtaining the spring rate, we will now select the materials and calculate the dimensions of the spring in order to achieve that stiffness. Such a high stiffness would be impractical to get with one helical spring, since the wire diameter (diameter of the circular rod that is wound to make the spring) would have to be so great that such a spring is impossible to manufacture. The equivalent stiffness of a system of springs in parallel is given as:

$$k_s = k_1 + k_2 + \dots + k_n \quad (5)$$

where  $k_1, k_2, \dots, k_n$  are stiffness of the constituent springs. If several identical springs are used in parallel for the suspension of each soft-seating tendon connector, then the spring rate of each individual spring is:

$$k_{si} = k_s/n \quad (6)$$

with  $n$  as the number of springs. The helical springs will be arranged radially around the central stem, so there will be 5 springs for a nicely balanced distribution of load, the spring rate of each one will be  $k_{si} = 1.785 \times 10^6 \text{ N/m}$ .

The material for the spring will be Stainless Steel Type 301, this is a steel alloy with chromium and nickel. This type of steel is chosen for the spring because of its high strength and ductility, making it suitable for forming and drawing, this alloy also has

great corrosion resistance. The spring rate is computed by the dimensions of each spring by the following formula:

$$k_{si} = \frac{Gd^4}{8D^3N_a} \quad (7)$$

where  $D$  and  $d$  are the mean diameter of spring and wire diameter respectively,  $N_a$  is the number of active coils, and  $G$  is the shear modulus of elasticity. For this high strength steel alloy,  $G = 80 \text{ GPa} = 80 \times 10^9 \text{ Pa}$ . The mean diameter and wire diameter of helical spring are related to each other by the spring index, defined as:

$$C_{spring} = \frac{D}{d}$$

For a heavy-duty spring used in this application, the spring index will have a small value but it has a bottom limit of 3 to ensure manufacturability, so a reasonable value for spring index is 5, this means  $D = 5d$ . If the spring has 4 active coils ( $N_a = 4$ ), then the wire diameter and mean diameter can be calculated using formula (6) to be:

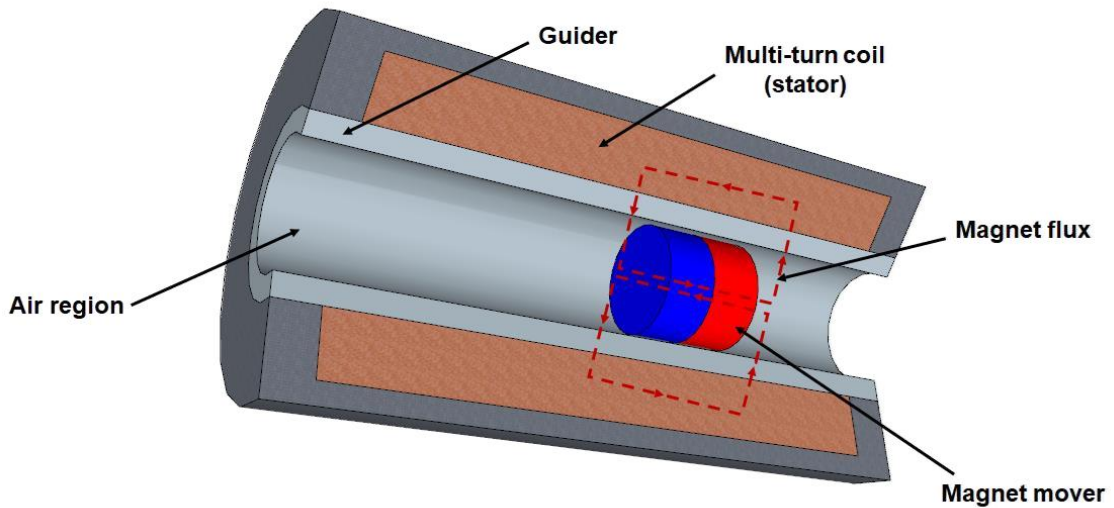
$$d = 0.09 \text{ m}, \quad D = 0.45 \text{ m}.$$



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### 5.4.3. Damper selection

The damper or shock absorber is a device that suppresses the unwanted vibrations by converting the mechanical energy into thermal energy, this makes the vibrations die down quickly. A suitable type of shock absorber for the soft-seating tendon connector is electromagnetic damper (EM damper). The main components of an EM damper are permanent magnet and copper coils. When the magnet moves relatively to the coils, the changing magnetic field through the coils create an electrical current in them, this is known as the Eddy current. The Eddy current in turn, produces a repulsive force that resists the movements, this force is proportional to the relative velocity between the magnet and the coils [42]. This behavior is consistent with the mathematical model of a typical damper. Such a device is regarded as a passive EM damper because it actually does not require electrical power to operate. An active EM damper utilizes electrical currents in the coils to create variable damping force.



**Figure 13:** Configuration of an electromagnetic damper [43]

Compared to regular hydraulic dampers, which use the flow restriction of viscous fluid for dampening effect, EM damper has the following advantages:

- It has variable damping characteristics that can be adjusted by controlling the electrical current going through the coils.
- It does not rely on fluid so there is no risk of leakage that leads to loss of fluid pressure and polluting the environment.
- It can be utilized for power regeneration by converting energy from mechanical vibrations back to electrical energy.
- It can still provide damping effect in case of power loss, by acting as a passive EM damper.

The damping coefficient of an EM damper is given as:

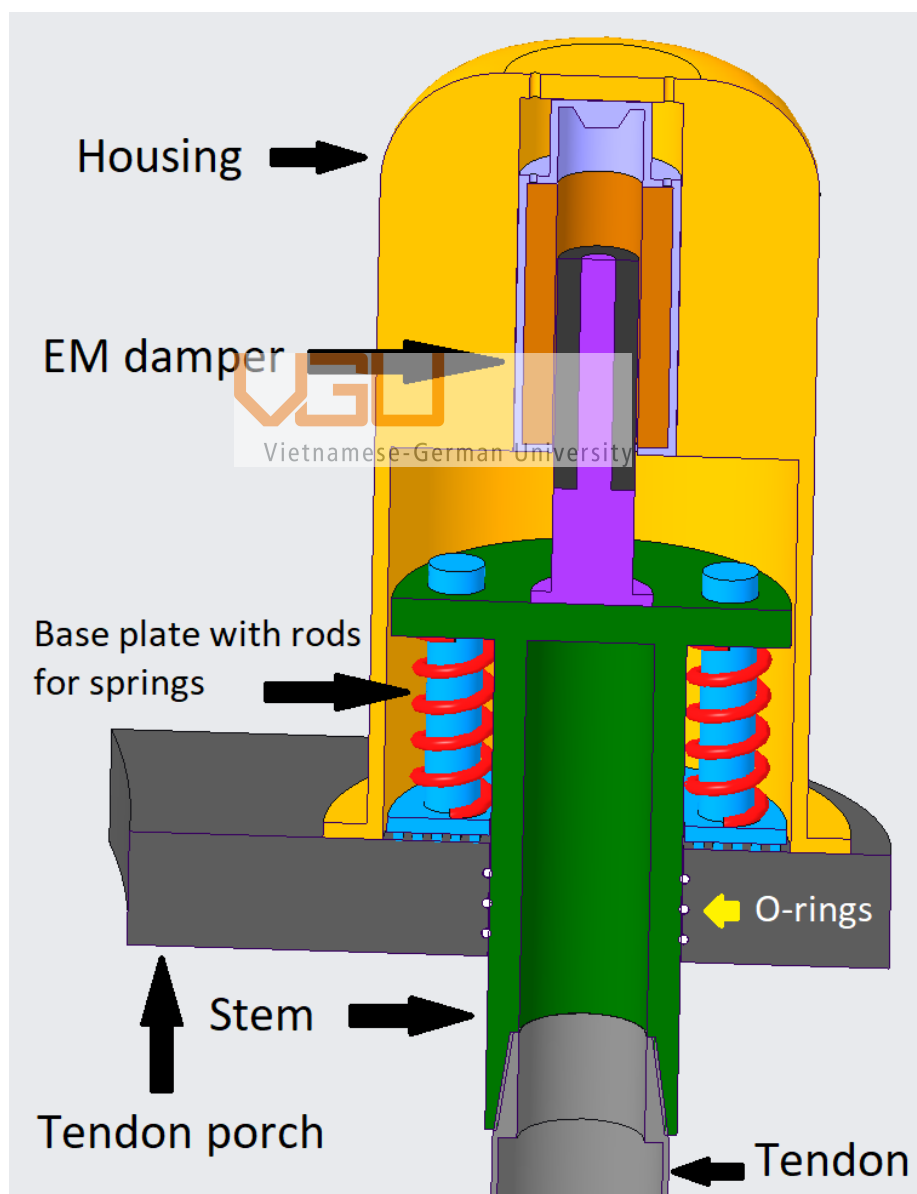
$$c = \frac{K_t^2}{\sqrt{(R_{coil})^2 + (\omega L_{coil})^2}} \quad (8)$$

where  $R_{coil}$  is the coil resistance,  $L_{coil}$  is the inductance of the coil,  $\omega$  is the frequency of the input excitation, and  $K_t$  is the motor constant of the damper [43]. The parameter  $K_t$  is defined by a very complicated formula containing many other physical dimensions and magnetic properties of the damper. Finding those parameters will be the scope of future works that concentrate on the design optimization of EM dampers.

## 6. CAD model

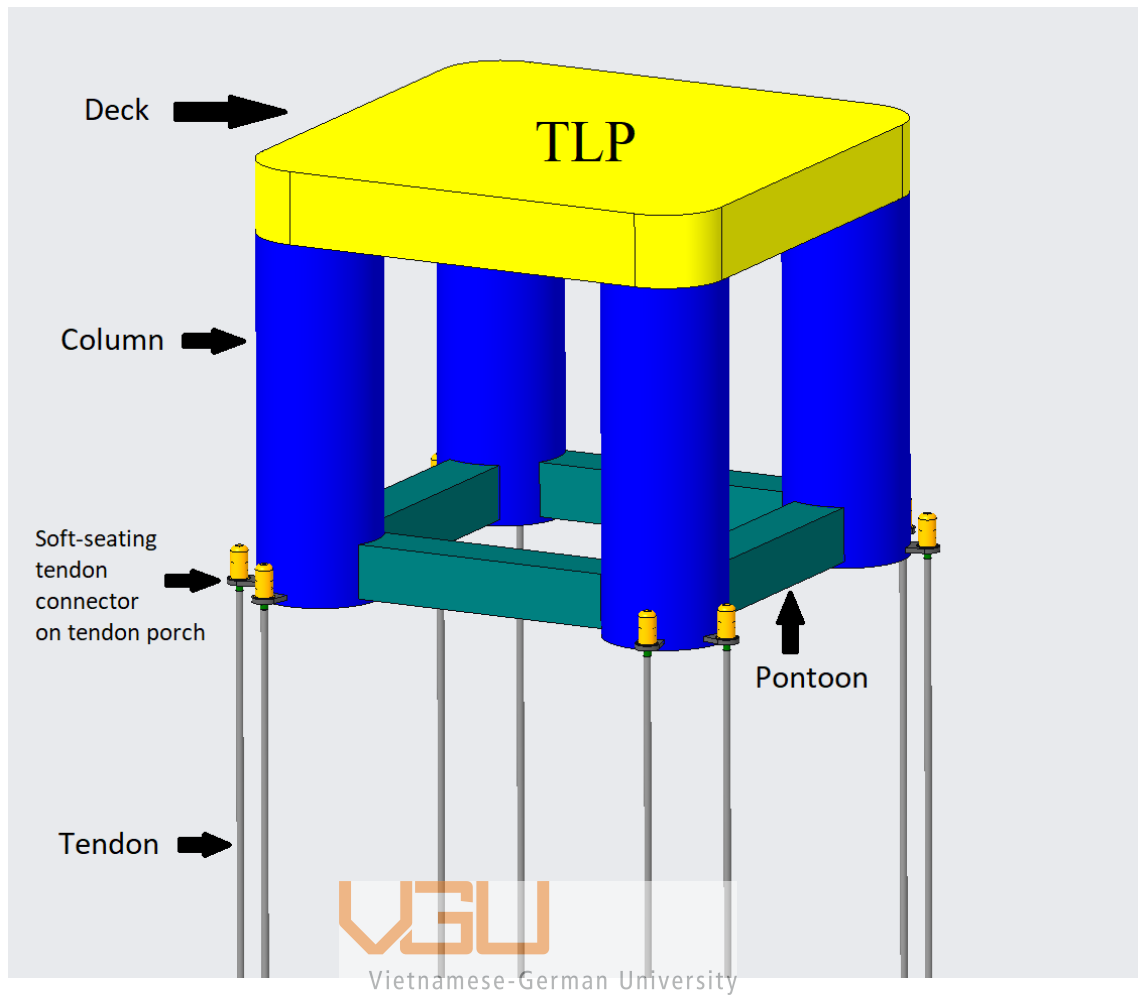
### 6.1. Assembly overview

With the dimensions from the TLP and tendons model and the calculated spring dimensions, a 3D computer-aided design (CAD) model is created. This model is based on the design of concept 2 but with a modification from a single spring in the center to five smaller springs around the stem. A cross-sectional view of the soft-seating tendon connector assembly is shown in Figure 14.



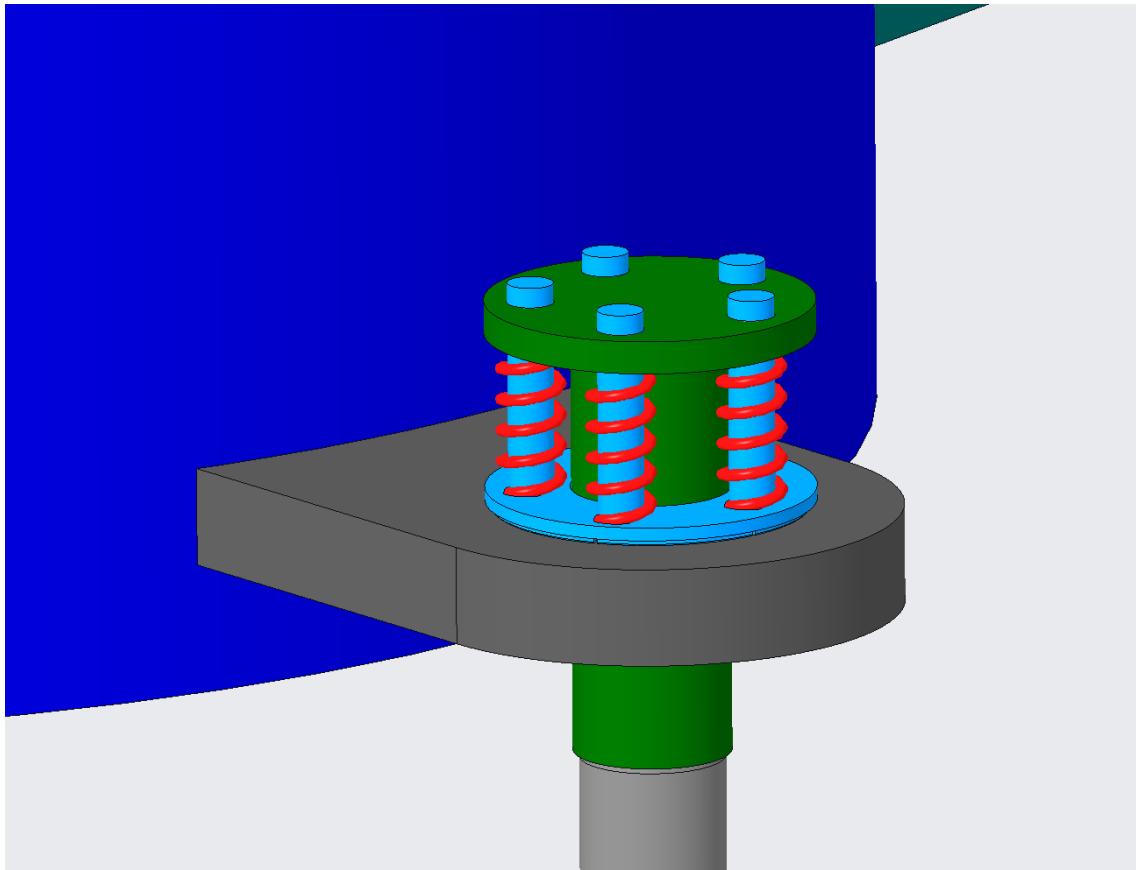
**Figure 14:** Overview of the soft-seating tendon connector assembly





**Figure 15:** Rendering of TLP model with the new tendon connector

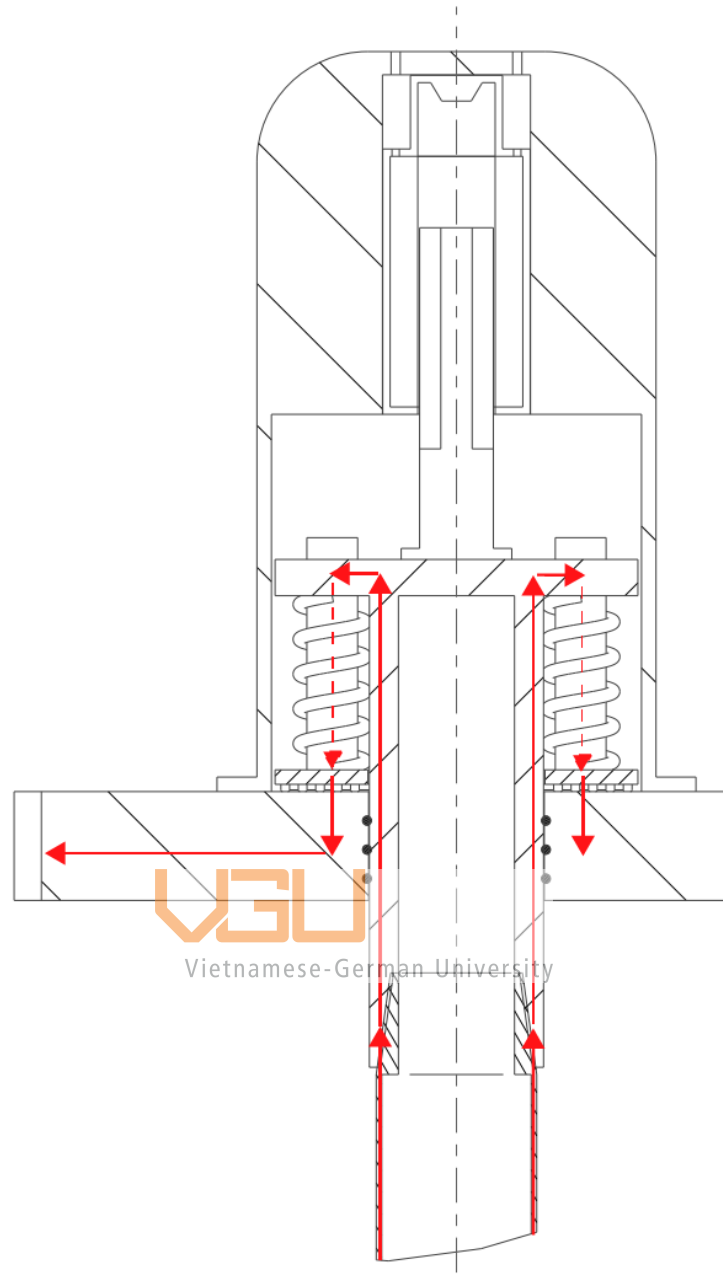
This design includes the modified tendon porch, which serves as a foundation for the assembly, the porch is a thick plate of steel to bear all the loads from the tendons, it is welded to the outside of TLP hull column, near the keel. Figure 16 shows the stem and springs configuration (damper and housing are hidden for clarity), the base plate has 5 vertical rods welded to it, these rods help to guide the springs and stop the springs from buckling when they are compressed. The rods fit into the holes at the top of the stem so the base plate and the stem can freely move up and down relative to each other but they will rotate together. This mechanism eliminates any radial force on the springs as the tendon rotates during its operation. The force flow inside the system is demonstrated in Figure 17, the load goes from the tendon to the stem through a tapered thread connection, it then travels through the springs to the base plate and finally to the tendon porch. The springs shown in this model are in their uncompressed state.



**Figure 16:** Springs arrangement around the stem

Since the base plate may rotate on the porch while taking a great downwards force onto it, there should be a way to reduce the friction at the contact surface between them. A thrust bearing that is designed to take axial load is a potential solution, but it is more suitable for high speed and not so heavy applications. Our system on the other hands, takes very heavy load and the rotations are infrequent, most of the time the base plate will be stationary as the tendon rarely rotates. Therefore, the underside of the base plate has concentric grooves with a purpose of reducing the surface area of contact between it and the porch. The two surfaces are further lubricated by grease. Three o-rings made of elastomer are installed in the grooves in the porch, they provide a tight seal around the stem to keep out the corrosive brine and avoid seepage of grease into the environment.

The choice of connection between the tendon top and the stem is a tapered thread because it is simple yet effective. This type of thread is commonly used as joints for drill pipes and also tendon sections themselves because it has good metal-to-metal seal and high axial loading capability. The thread gets tighter as it further engages so it can resist torque loading very well.

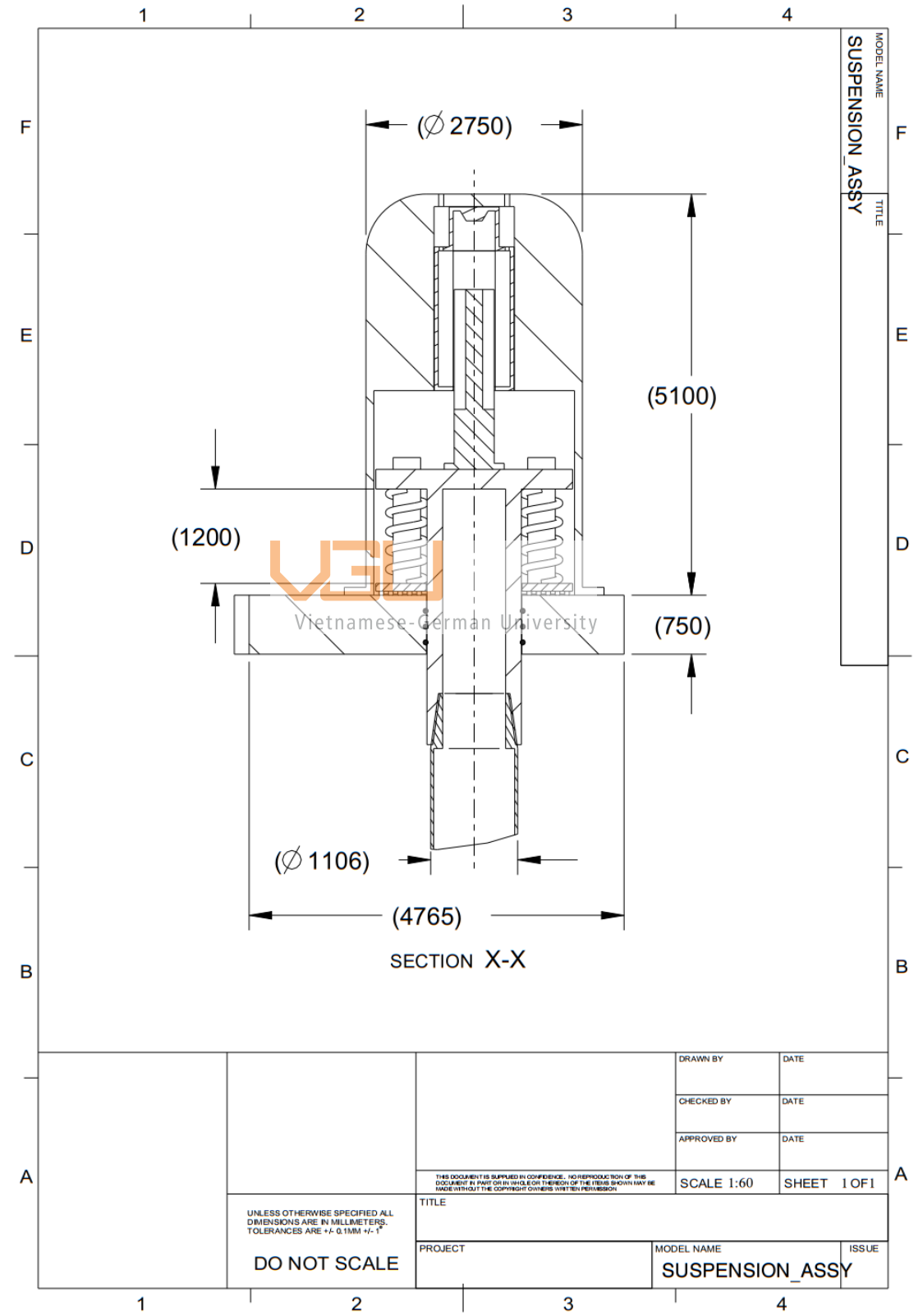


**Figure 17:** Force flow inside the soft-seating tendon connector

The dimensions of the electromagnetic damper have not been finalized so only a general 3D model depicting its shape and approximate size is made. The housing has a function of containing the assembly from the environment and also keeping the EM damper fixed to it. Thus, only the diameter of the housing is determined but its height is to be confirmed in the future. There are 2 holes on the top of the housing for electrical cables that power the EM damper. The housing will be attached to the tendon porch by screws or studs, as well as the connection between the stem and the EM damper piston. These are minor details on the parts that have not been finalized so they are not included in this model.

## 6.2. Drawings and component details

The assembly drawing is created to show the arrangement of the soft-seating tendon connector assembly and its general dimensions.



The selected material for the assembly components is Inconel Alloy 625. This is an alloy of nickel and chromium, known for its high strength and great resistance against corrosion. For these characteristics, Inconel 625 is one of the most widely used metal alloy for oil and gas equipment, so it makes a good choice of material for the assembly parts, which are subjected to a lot of stress and a highly corrosive environment of the sea.

The main components of the tendon connector assembly are described as follows:

❖ **Stem**

- Function: to join tendon to soft-seating connector and transfer load from tendon to springs and damper.
- Material: Inconel Alloy 625.

❖ **Springs**

- Function: to allow vertical displacements inside the connector and restore positive tension in tendon.
- Material: Stainless Steel Type 301.

❖ **Base plate**

- Function: to support springs and permit rotation with the stem.
- Material: Inconel Alloy 625.

❖ **Tendon porch**

- Function: to support and attach the tendon connector assembly to the TLP column.
- Material: Inconel Alloy 625.

❖ **O-rings**

- Function: to provide a seal between the inside and outside of the tendon connector assembly module.
- Material: Nitrile (elastomer).

❖ **Electromagnetic damper**

- Function: to absorb the impact and dampen the vibrations from snap loading.
- Material: to be decided.

❖ **Housing**

- Function: to shield the interior components of the tendon connector assembly from the outside environment (sea water) and house the EM damper.
- Material: Inconel Alloy 625.

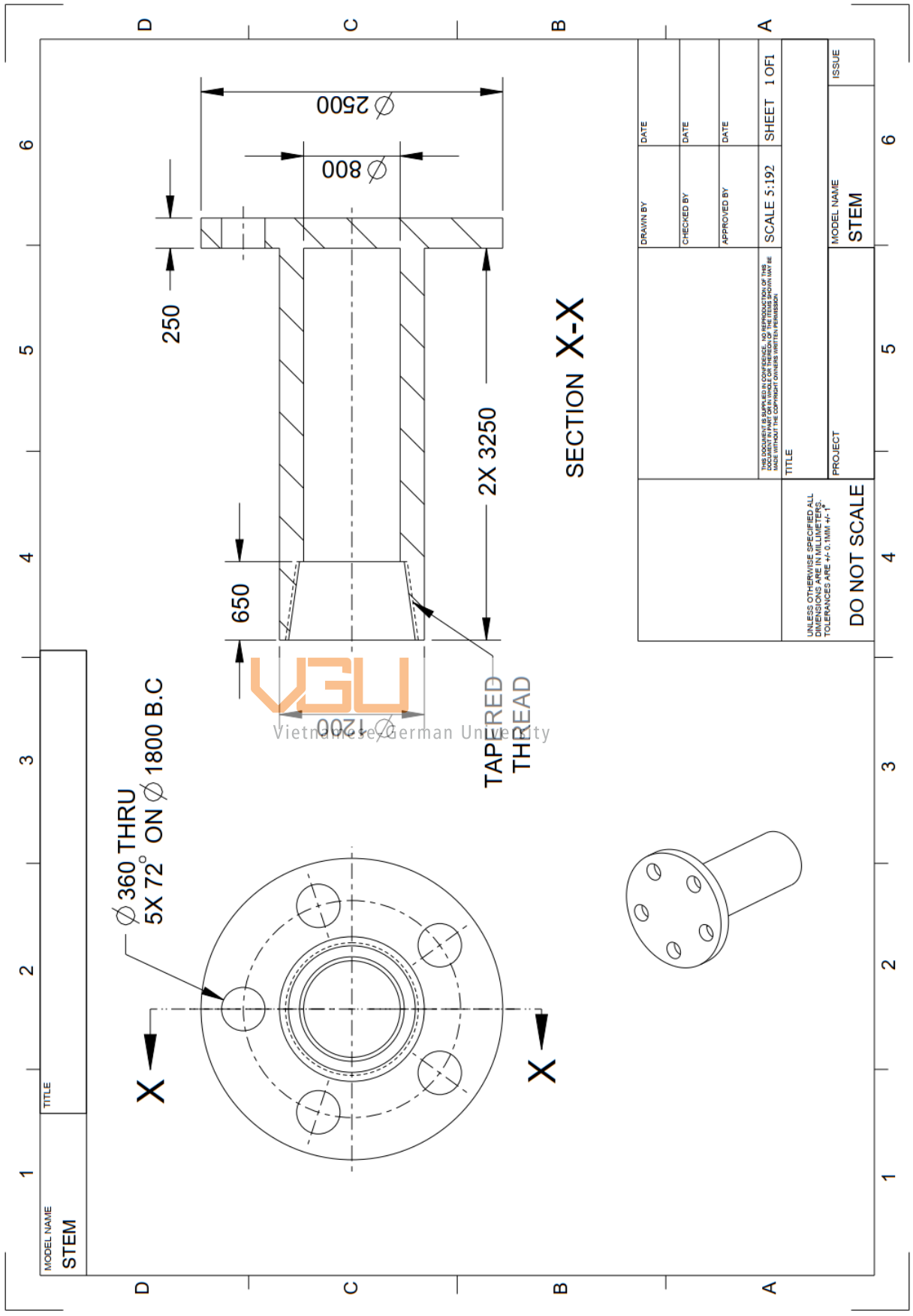
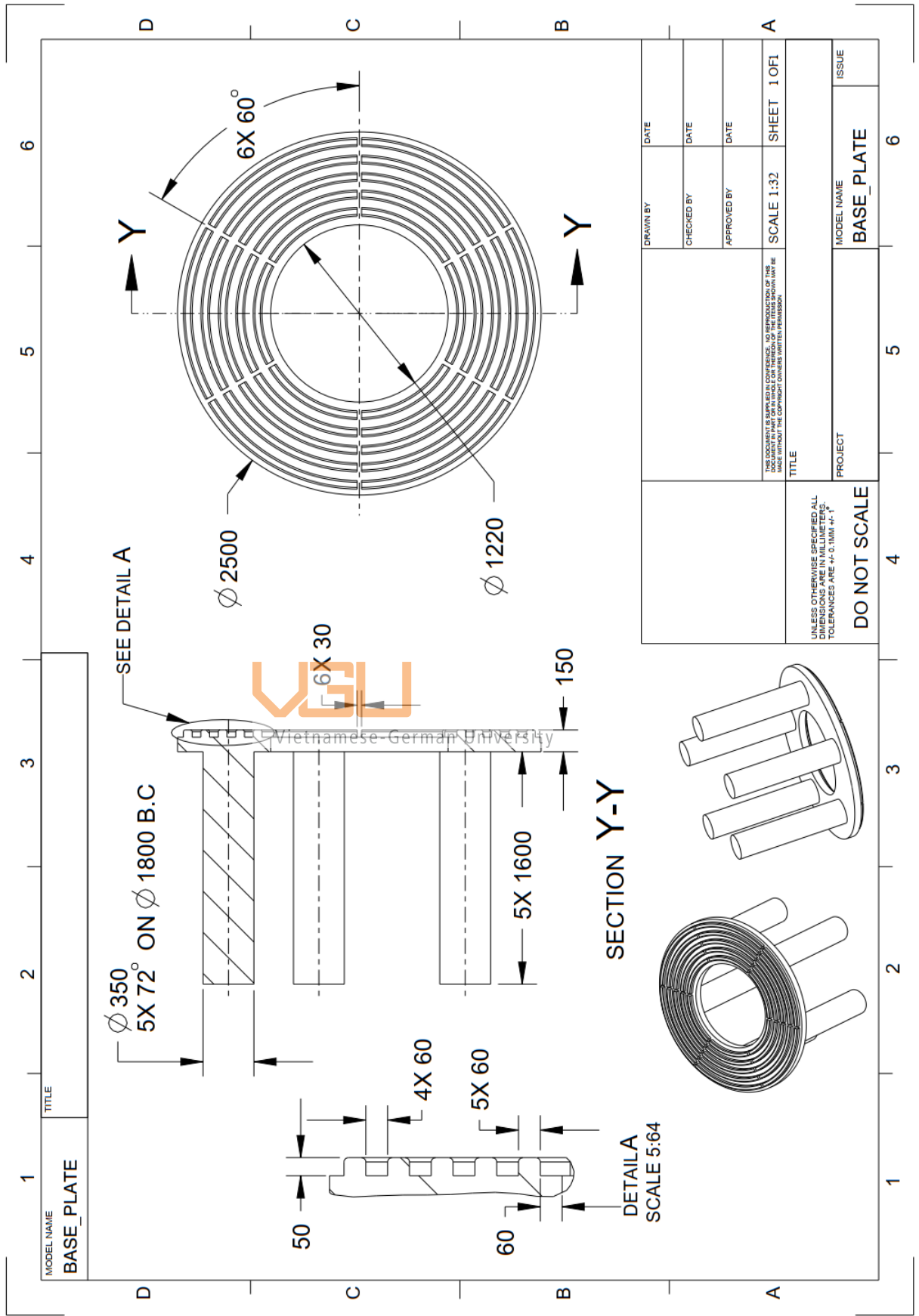
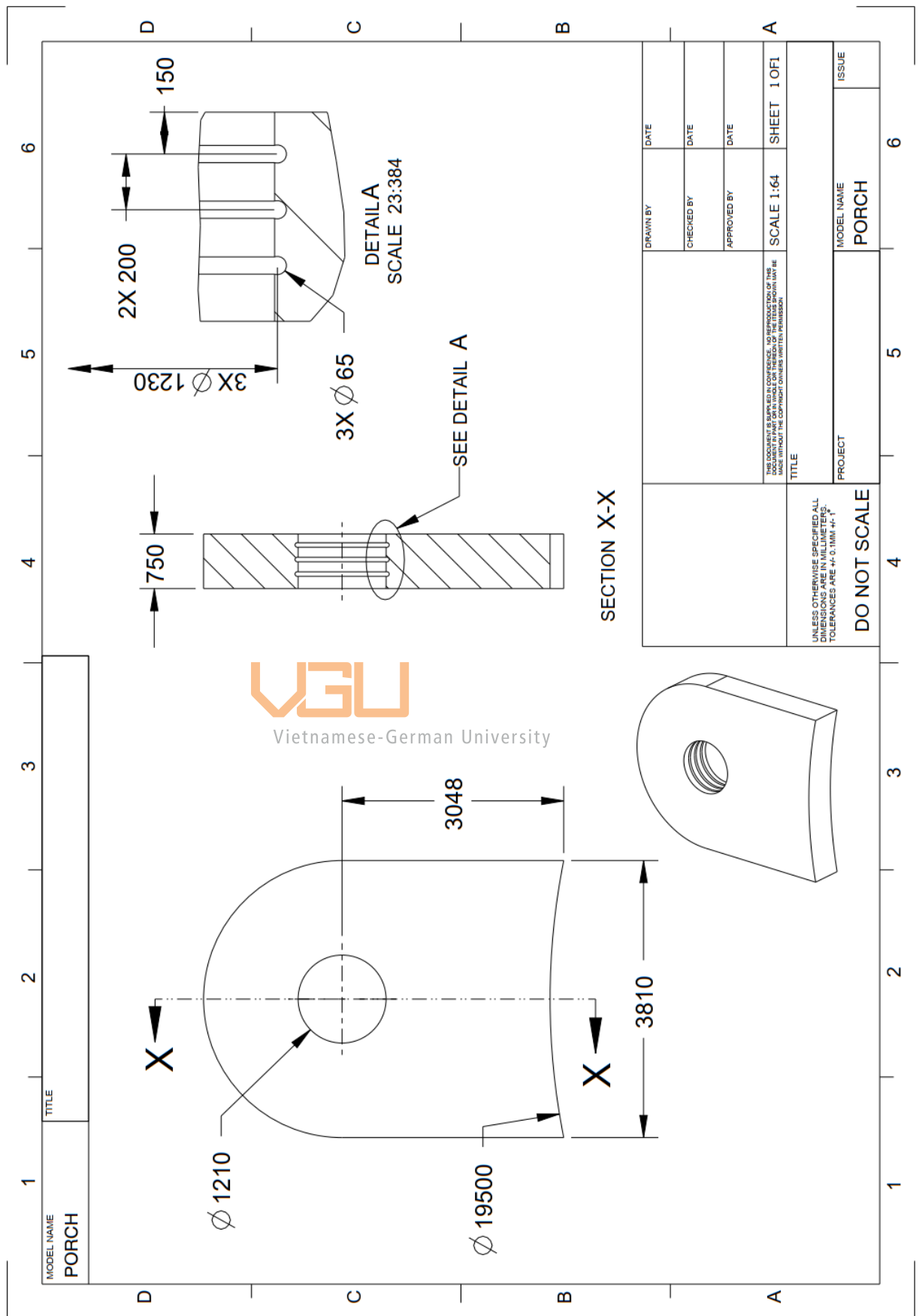


Figure 18: Drawing of stem



**Figure 19: Drawing of base plate**



**Figure 20:** Drawing of tendon porch



## **7. Conclusion and Future Work**

### **7.1. Conclusion**

This thesis documented the design process of the alternative lower anchor connector for the Tension Leg Platform, from gathering requirements, generating ideas to concept selection and detail design. The final design employs a spring-damper suspension system and is positioned at the tendon porch, on the outside of the TLP hull column instead of the anchor, hence the name “soft-seating tendon connector”. It is able to reduce snap load in tendons by functioning as a suspension system that vibrates at a different natural frequency from sea waves, in order to avoid resonance. The 3D CAD model of system components and detailed drawings for some of the parts are created to show the working principle and dimensions of the device. There has been a change in the spring arrangement from one central spring in the conceptual design to five smaller springs around the stem in the final detailed design.

A comprehensive calculation for the parameters of the damper will be the subject of future work because the design of such an EM damper requires an extensive study in the field of electrical engineering, which is not the main focus of this thesis. This thesis has produced a 3D model of the full-scale product but lab testing with a physical model should be done and stress simulations are needed to validate and optimize this design.

### **7.2. Recommended future work**

The work in this thesis has shown the viability of an alternative tendon connector that reduces snap load in TLP tendons. The following studies can be carried out to turn this design into a commercial product:

- The EM damper should be further studied to confirm its practical application in this heavy-duty suspension system.
- More stress analysis to be performed to optimize the component geometry and dimensions for reducing costs of material and manufacturing.
- The other design concepts in this thesis have good potential for future development.
- Coupled analysis of static and dynamic load in tendons can be done to reduce pre-tension to relieve stress in tendons and the connector.

## 8. Appendices

### 8.1. Appendix I

No.	Selection Criteria	Weight	Concepts																							
			1		2		3		4		5		6		7		8									
			Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score								
1	Stability	18%	3	0.54	3	0.54	2	0.36	3	0.54	2	0.36	3	0.54	3	0.54	3	0.54								
2	Ease of installation and maintenance	9%	2	0.18	3	0.27	3	0.27	3	0.27	2	0.18	2	0.18	3	0.27	2	0.18								
3	Compatibility with existing TLP/anchor	7%	3	0.21	3	0.21	2	0.14	4	0.28	2	0.14	3	0.21	3	0.21	2	0.14								
4	Speed of restoring tension	10%	3	0.3	2	0.2	3	0.3	3	0.3	3	0.3	3	0.3	4	0.4	3	0.3								
5	Wear on tendons	8%	1	0.08	3	0.24	3	0.24	2	0.16	3	0.24	3	0.24	3	0.24	1	0.08								
6	Ability to absorb impact	12%	3	0.36	3	0.36	3	0.36	1	0.12	4	0.48	3	0.36	2	0.24	3	0.36								
7	Ability to adjust system parameters	5%	2	0.1	2	0.1	2	0.1	1	0.05	4	0.2	4	0.2	1	0.05	3	0.15								
8	Reliability	31%	3	0.93	4	1.24	3	0.93	3	0.93	3	0.93	3	0.93	3	0.93	3	0.93								
	<b>Total</b>	<b>100%</b>		2.7		3.16		2.7		2.65		2.83		2.96		2.88		2.68								

Table 3: Pugh matrix for concept selection

## 8.2. Appendix II

Designs of conventional tendon bottom connector.

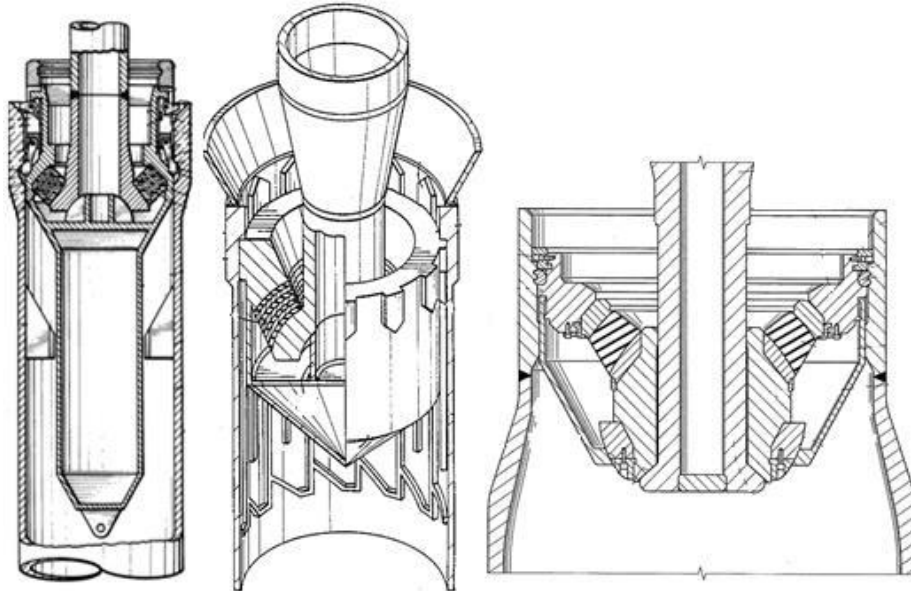


Figure 21: Snap Ring type Bottom Tendon Connector [44]

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Figure 22: Rotolatch type Bottom Tendon Connector [44]

### 8.3. Appendix III

Layouts of the TLP model in South China Sea.

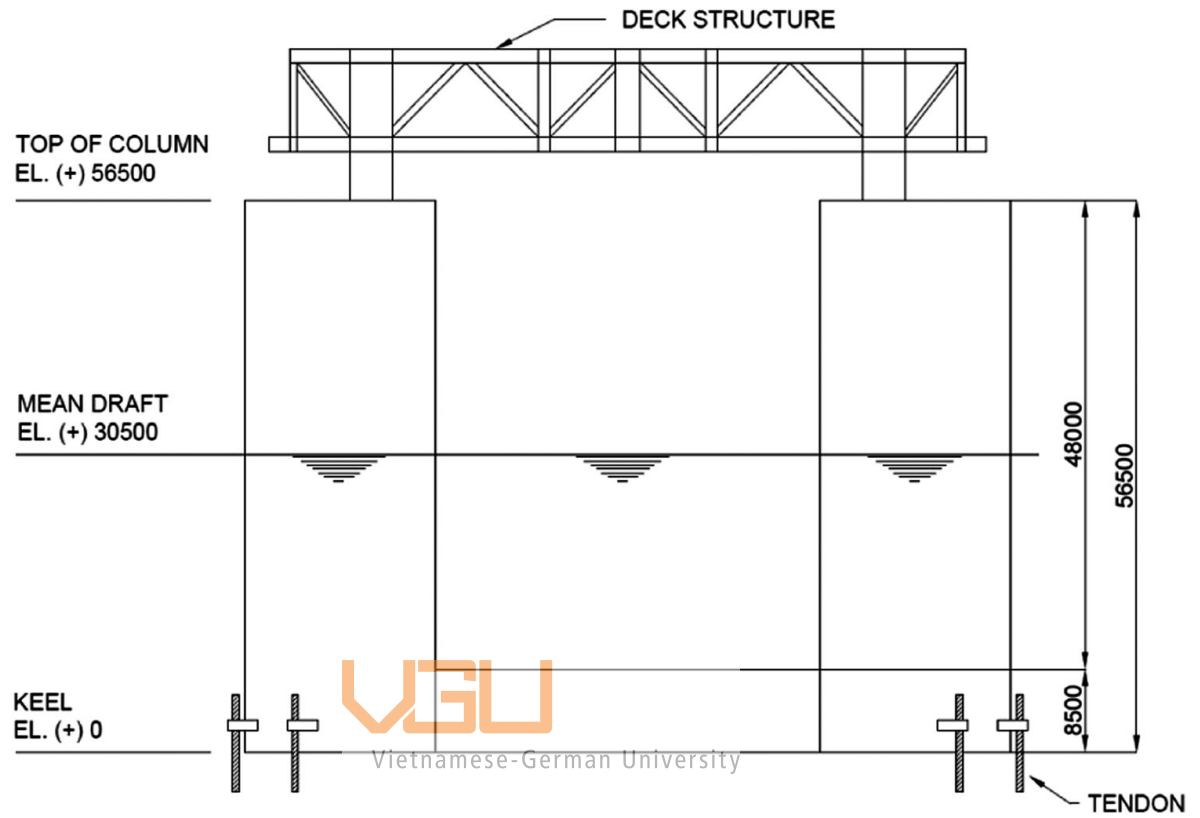
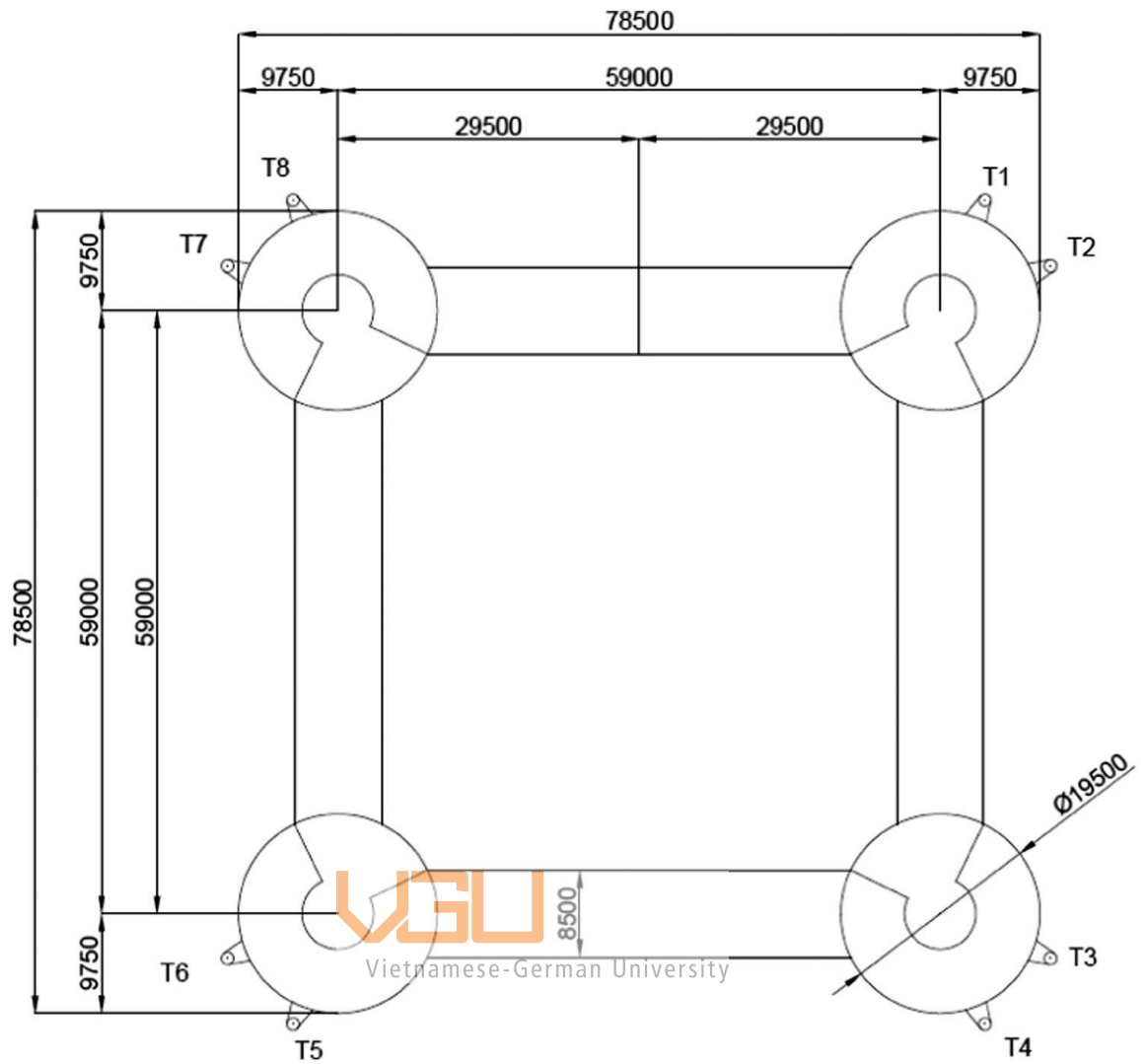


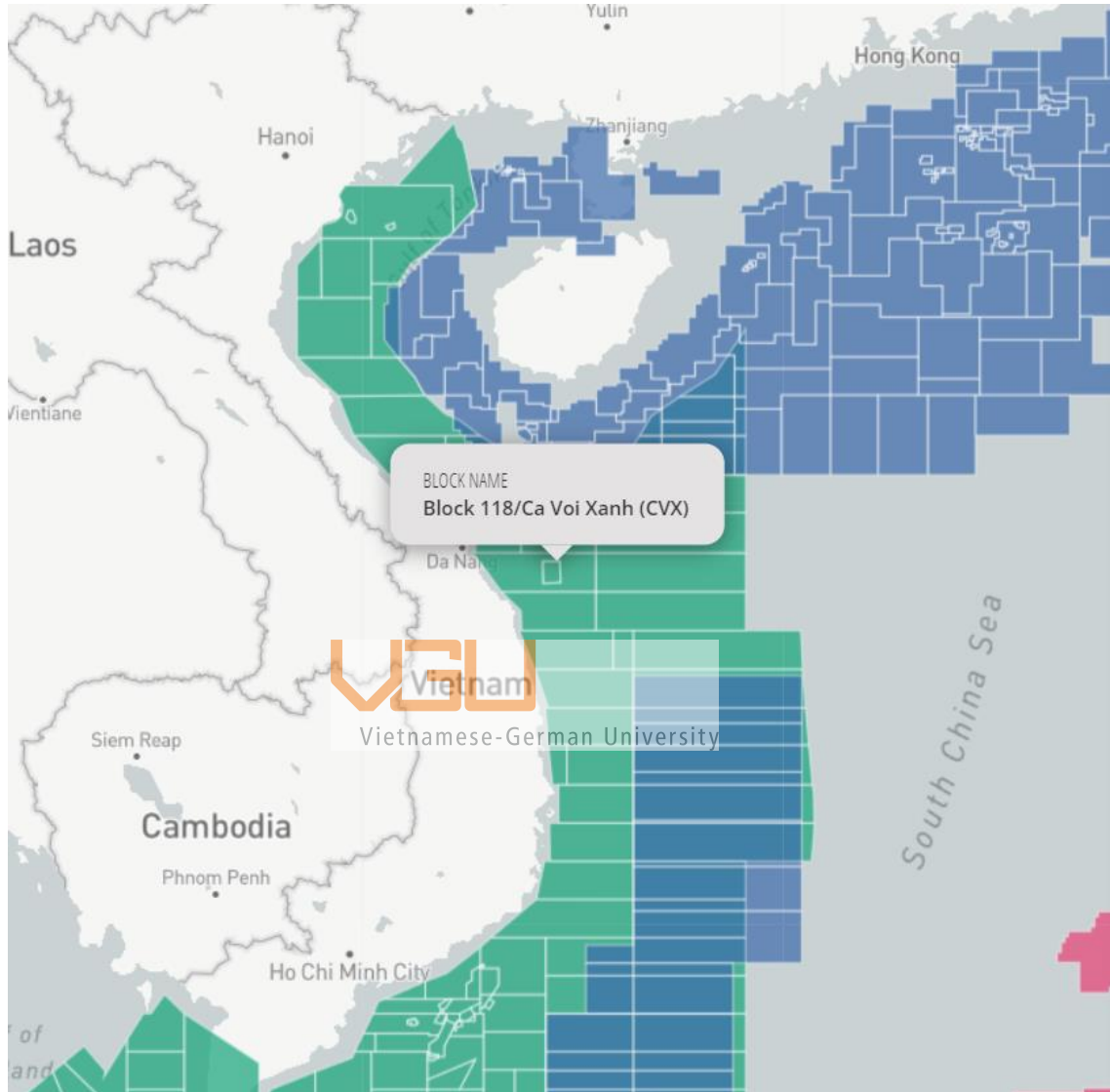
Figure 23: TLP layout elevation view [2]



**Figure 24:** TLP layout plan view [2]

## 8.4. Appendix IV

Position of the Ca Voi Xanh oil and gas field in the map.



**Figure 25:** Ca Voi Xanh field, Block 118 [45]

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