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A SYSTEMATIC DESIGN APPROACH OF GRIPPER'S HYDRAULIC SYSTEM UTILIZED IN OFFSHORE WIND TURBINE MONOPILE INSTALLATION

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ABSTRACT

This paper presents a systematic approach for designing the hydraulic mechanism as a part of the gripper system employed in offshore wind turbine monopile installation. Traditionally, such equipments used in marine operation are designed based on deterministic approach, selecting actuators and power pack by applying a safety margin which is not explicitly derived from a systematic load/ load effect analysis, or a reliability based method. The method in this article offers a systematic way of designing the hydraulic power system, actuators and supporting structure to overcome extreme and fatigue loadings during operation. The design starts with a global analysis and modelling of monopile and installation vessel. The forces and motions from global analysis are then employed for designing hydraulic actuators and power system. A dynamic model of hydraulic system is built to analysis dynamic response in hydraulic system. The results from this local dynamic model can be used in power management and system optimization. The proposed method is a step forward to apply reliability-based design on mechanical components in marine applications through a systematic long-term load and load effect analysis.

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INTRODUCTION

Today, wind energy accounts for 18% of the European Union's total installed power generation capacity comprising 153 GW onshore and 15.8 GW offshore [1]. In offshore wind industry, monopile (MP) foundations are the most dominant technology in shallow waters. MP foundations are cylindrical piles used as substructures in bottom-fixed offshore wind turbines. As of 2017, 81% of 3589 offshore wind turbines in the Europe were installed on monopiles [2].

Monopiles are transported by barge to the installation site where they are hammered to the seabed. The installation process starts with lifting the monopiles by a crane on the installation vessel. Then the monopile is held by means of a gripper device, fitted to the installation vessel, and then it is hammered to the seabed. The gripper is also used to correct the position of the monopile. Self-propelled jack-up vessels are commonly used for offshore wind turbine installations - like INNOVATION vessel for instance [3]. Apart from jack-up vessels, floating installation vessels are also introduced in the market. Fig. 1 presents a floating vessel specialised for offshore wind turbine installations, the gripper device is shown in Fig. 2.

The gripper device is often designed separately and fitted

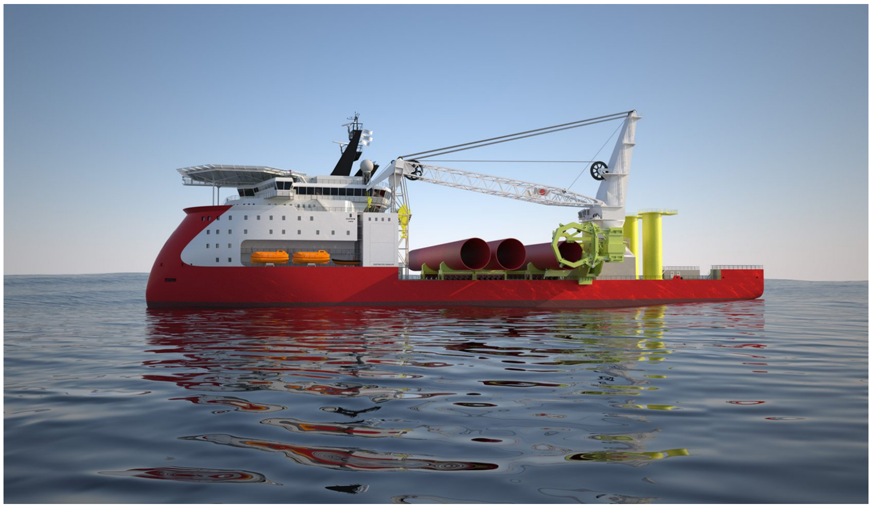

FIGURE 1: Floating installation vessel, ULSTEIN-HX102, with Ulstein wind installation equipment (Illustration courtesy of Ulstein) [4].

FIGURE 2: Pile gripper frame monopile lift (Illustration courtesy of Ulstein) [4].

on the installation vessel. It consists of a hydraulic mechanism to hold the monopile and to rectify the inclination angle. The acceptable inclination angle is often about 0.25 degree. While the monopile itself is studied extensively - see for example [5,6], limited works are devoted to investigate the dynamic analysis of installation operation [7–9]. The gripper device, however, is often selected based on experience without passing through a site-specific systematic design approach. Deterministic safety factors are often used to design the hydraulic actuators and power pack system. Since these safety factors are not derived from a systematic reliability approach, the level of reliability which can be achieved will not be clear. Therefore, this method results in a system with unknown reliability level, it may be over or under estimated design. Offshore wind turbine installation is a costly operation which is carried out in a limited weather window. Thus, tools and equipments used in this operation must be designed to meet a certain level of reliability to avoid interruption during operation. Hydraulic systems used in marine operations need to be designed to fulfil their operational requirements and expected life. One of the most important operational requirements during the execution of a marine operation is to ensure the structural integrity of the components.

This paper presents a systematic design procedure of the hydraulic system and its supporting structure, as a part of the gripper device. At first a global model including the vessel, monopile and supporting structure is considered. The forces and motions on the gripper device at different environmental conditions are obtained from the global analysis. Second the results from the global analysis is used to design the hydraulic actuators, pump and power pack, and to set up the control system. The actuators should be designed based on the limit state design approach where ultimate limit state (ULS) and fatigue limit state (FLS) are evaluated. In the third step the dynamic behaviour of the hydraulic system is studied through a local model. This approach can be employed for designing mechanical systems used in offshore installations in a systematic manner through load and load effect analysis to ensure a certain level of the system reliability. The proposed method is illustrated by a case study model of a monopile installation by a floating vessel.

PROPOSED DESIGN APPROACH

The design of the gripper device is governed by vessel and monopile motions and forces. The motions and associated induced forces on the gripper can be estimated by the global dynamic response analysis.

A systematic design procedure can be in general summarized in following three steps:

- **Step 1: Global analysis** to estimate load effects and motions on gripper device at different environmental conditions. A dynamic model of the vessel, monopile and seabed is needed for this step. Moreover the environmental loads must also be modelled.
- **Step 2: Design of hydraulic actuators**, hydraulic power pack, control system, and supporting structure based on the limit state design approach. First the actuators, and their supporting structure, are designed based on the extreme load (ultimate limit state or ULS design) and then their fatigue life is evaluated (fatigue limit state or FLS design).
- **Step 3: Dynamic analysis** of the hydraulic system, to evaluate the system behaviour under dynamic loadings during operation.

In this paper, the systematic design method is illustrated by an example of a monopile installation by means of a floating installation vessel. Table 1 provides the specification of the heavy lift vessel (HLV) and the monopile - more details about this model can be found in Li et. al [7]. The HLV is considered to be moored by eight catenary mooring lines to the seabed during the operation.

TABLE 1: Heavy lift vessel (HLV) and monopile (MP) specification.

Parameter	Value
HLV displacement (ton)	5.12×10^4
HLV length (m)	183
HLV breadth (m)	47
HLV draught (m)	10.2
MP mass (ton)	500
MP diameter (m)	5.7
MP length (m)	60

Step 1: Global analysis

For the global analysis of the case study model, the numerical model of the coupled HLV-MP (heavy lift vessel - monopile) system was established using MARINTEK SIMO program [10] and time-domain simulations were carried out. The model includes coupled two-body HLV-MP system with mooring line positioning system on the HLV and soil interaction forces on the MP at different penetration depths. The methodology for the modelling and the time-domain simulation of HLV-MP are explained in detail in Li et. al [7].

The global HLV-MP coupled dynamic system has 12 degrees of freedom (DOFs), and the coupled equations of motion are solved in the time-domain. The HLV and MP are coupled through the gripper device. The external forces include the first-order wave excitation forces, the second-order wave excitation forces, the mooring line forces on the HLV, and the soil reaction forces on the MP. Eight catenary mooring lines for the HLV were modelled.

The hydrodynamic interaction between the HLV and MP mainly affects the forces on the MP due to the vessel presence, which is considered as “shielding effects” [9]. In the current numerical model, the hydrodynamic interaction problems between the HLV and the MP were solved using the panel method program WAMIT [11] in the frequency domain.

Step-by-step integration methods were applied to calculate the responses of the coupled HLV-MP system with a time step of $0.01s$. JONSWAP spectrum with index $n = 3$ for the short-crested wave spreading function cos^n is applied for all sea states [12].

The gripper device is a ring-shaped structure with four hydraulic cylinders in a radial array which provides pressure and thus compression forces on the MP during the initial hammering process. In the global model, the gripper device was simplified by four fender components with chosen stiffness and damping coefficients. When the MP tends to move away from the gripper, the fenders provide compression forces and limit its horizontal motions. The elastic model for the gripper contact elements are illustrated in Fig. 3. Sensitivity studies on the gripper stiffness during lowering of a MP were performed in Li et. al [13]. The study showed that the contact force and the relative motion between the MP and the gripper device were very sensitive to the gripper stiffness.

The parameters for the gripper are chosen based on specifications of typical hydraulic cylinders which are applied

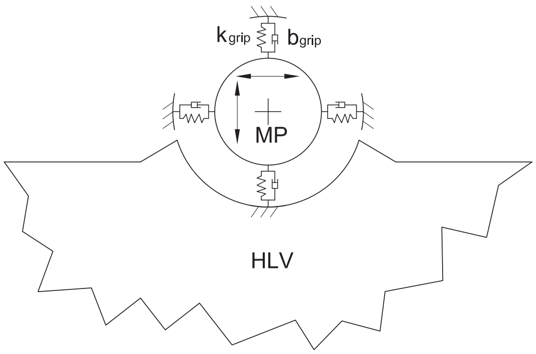

FIGURE 3: Elastic model of the gripper contact elements.

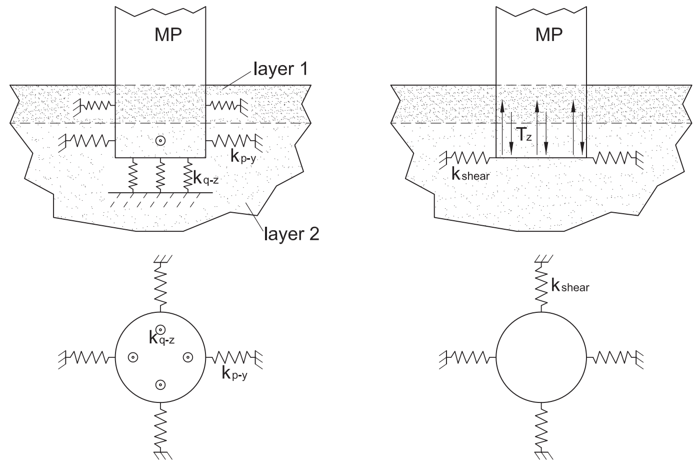

FIGURE 4: Numerical models for the soil-MP interactions.

in practice for MP installation. During the hammering process, the valves of the hydraulic cylinders are normally closed. The stiffness of the cylinder with closed valves was chosen to be $3 \cdot 10^7 N/m$ in the global model. The damping in the numerical model is taken to be 20% of critical damping [14]. A pre-compression force of 150 kN for each hydraulic cylinder is applied.

In the global model, the soil-MP interaction is modelled using Winkler model by means of distributed springs and the hysteresic material damping [15, 16]. The penetration of the MP during the initial hammering process ranges from around 2 m (self-penetration) to around 6 to 8 m (depending on soil properties). The soil-MP interaction forces in the shallow penetration phases are three-dimensional, therefore the 2D Winkler model is extended to 3D by using non-linear springs distributed in both axial and circumferential directions along the MP. The distributed springs include the traditional lateral load-deflection $p-y$ curve, the friction $T-z$ curve which was found to be significant for large diameter piles with shallow penetrations [17], the base shear curve and the tip load-displacement $Q-z$ curve.

The configuration of the springs as shown in Fig. 4 is summarized as follows: 4 vertical springs K_{q-z} to model $Q-z$ curves at the bottom of the MP; 4 springs T_z on the side of the MP to model the $T-z$ curve for the friction force from both inside and outside wall of the MP, and the vertical position of the T_z springs are calculated by considering the distribution of the friction along the whole MP penetration length. For $p-y$ curves, the whole penetration is divided into several 2m-layers, and 4 circumferential springs K_{p-y} are applied for each layer. On the bottom of the MP, 4 springs K_{shear} are used to model the shear resistance force.

An estimate of the stiffness for all the non-linear distributed springs shown in Fig. 4 are taken from the API guideline [18]. The sensitivity study on the soil properties, it is concluded that the dynamic system behaviour during the operation does not change with the soil properties [7]. Therefore, representative values for the non-linear springs for the soil-MP interactions are considered to be sufficient. The soil properties used to calculate the spring stiffness can refer to Table 3 in Ref. [7]. The soil damping is included in this model in terms of dynamic friction force.

By performing time domain simulations at various environmental conditions, the forces and motions on the gripper device are obtained - see items 1-3 in Figure 5.

Step 2: Hydraulic system & supporting structure design

The hydraulic system includes actuators (cylinders), pump and electrical motor, control system and supporting structure. First, the actuators and supporting structure should be designed based on the global data. Second, pump and power pack and control system are selected and a hydraulic system model is developed to study the dynamic behaviour of the system. These tasks are detailed in following subsections.

Actuator & supporting structure design

Fig. 5 shows an overview of the methodology for systematic ULS and FLS design of the hydraulic actuators and their structural support. Items 1-3 cover the global dynamic response analysis as described earlier. All possible significant wave height (Hs) and spectral wave period (Tp) combinations should be considered as wave parameters for analysis of the monopile installation activity. The Hs and Tp parameters are used to model the sea wave spectra that is applied on the coupled dynamic system. The time history of the dynamic contact force between the MP and the hydraulic system roller is the output from the global analysis. This force is important for assessment of a critical structural component of the hydraulic system.

Normally, a structural system has one or few components

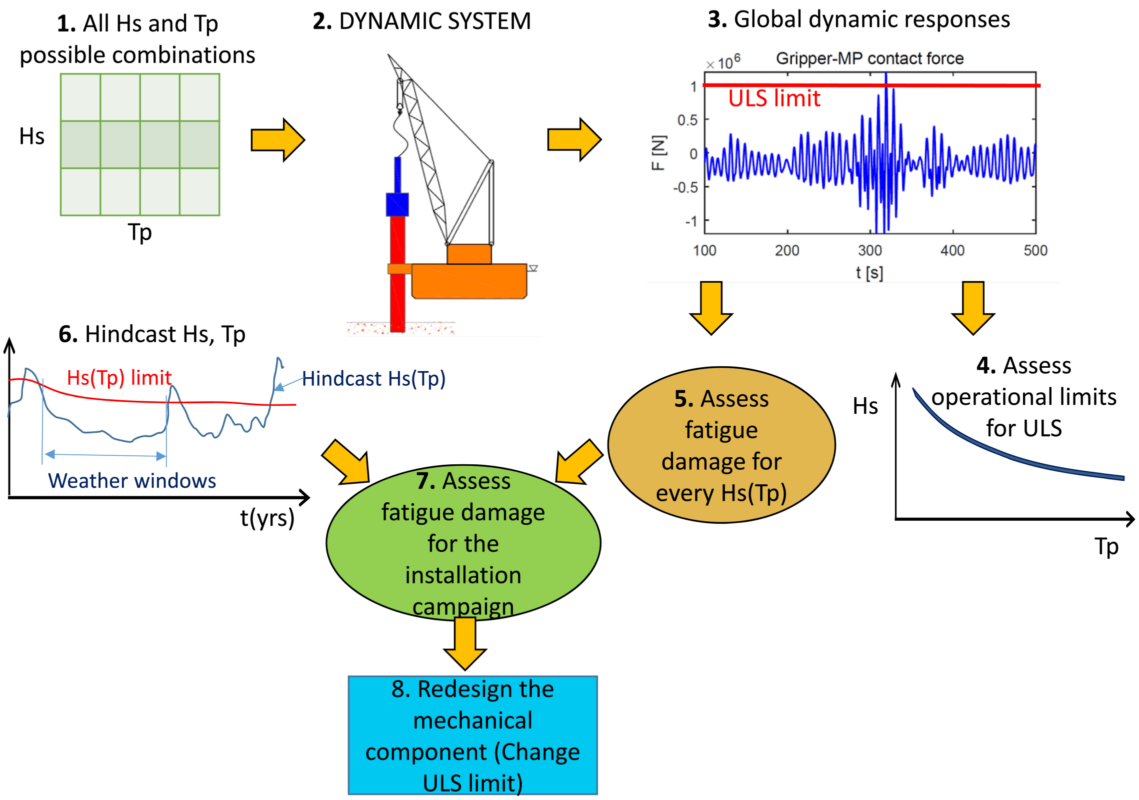

FIGURE 5: Methodology for ULS and FLS assessment of a hydraulic actuator and its supporting structure.

which limit the execution of a marine operation. In case of MP installation, a critical component can be the radial beams holding the MP in vertical position or the cylinder rod itself. The resistance of that specific component can be translated into an allowable ULS force. A characteristic value of the force obtained for each Hs(Tp) using several seeds can be compared with the allowable limit and the operational limits in terms of Hs(Tp) can be obtained, as shown in item 4. More details of a general methodology for assessment of the operational limits can be found in Guachamin Acero et al. [8].

The dynamic responses obtained from the global analysis can be used to assess the fatigue damage in the structural component and the actuators for every sea state. The fatigue damage will depend on the dynamic responses and duration of the operation. For assessment of the stochastic force or stress ranges and corresponding number of cycles, the Rainflow Counting Method (RFC) can be applied. On the other hand, historical wave data in the format of seasonal hidcast time histories and the operational Hs(Tp) can be used to assess the workable weather windows - item 6 in Figure 5. Total cumulative fatigue damage can be then calculated for the range of simulated Hs(Tp). This assessment can be carried out for several years of hindcast wave data to reduce uncertainty in the results. The fatigue damage need to be assessed for the complete duration of the installation campaign or the number of foundations to be installed. If the fatigue damage is more than 1, a redesign of the structural component will be needed.

A redesign of a structural component means that the allowable force and the operational limits should increase in order to meet the desired design life, provided the operational limits Hs(Tp) remain unchanged.

For the case study system, the actuators are first designed based on the extreme force during operation (ULS design). The extreme contact force, obtained from the global analysis, covers two responses: response from the steady-state conditions and the force applied to correct inclination angle. As discussed earlier the global analysis should cover all possible environmental conditions and durations. In this example the extreme dynamic forces are calculated as the maximum value in 3 hrs obtained from an earlier study by Li et. al [7].

The earlier study [7] shows that the cylinder force increases as the penetration depth increases. For a given Hs, the hydraulic force increases with Tp. Obviously contact force is higher in higher Hs. This also indicates that the design of the actuators is in fact an important parameter in selecting the operational weather window.

It should be noted that the extreme value obtained from the global analysis is associated with many uncertainties. Uncertainty in global model, wave and wind model, vessel and structural model, statistical uncertainty due to limited number of simulations, and uncertainty in modelling the hydraulic cylinders. These uncertainties must be quantified and considered in the design.

In designing the hydraulic cylinder, uncertainties are accounted in the safety factor. The main design driver in hydraulic cylinders is the piston rod buckling, which is calculated from Euler's formula [19]:

$$K = \frac{\pi^2 EJ}{L^2} \tag{1}$$

where K is the buckling load in kg, E modulus of elasticity in kg/cm^2, J is the second moment of area of the rod in cm^4 and L is the equivalent buckling length (cm) of the cylinder which depends on the mounting configuration (for example for double eye mounting L is equal to the distance between two eyes' centres). The allowable, safe, load F on the cylinder is then obtained from:

$$F = \frac{K}{SF} \tag{2}$$

in which SF is the safety factor. Hydraulic equipment suppliers often build cylinders for general applications with SF of minimum 3.5. What level of reliability is obtained with this safety factor in this case of monopile installation needs further study and quantification of uncertainties through a reliability analysis.

For this case study example, based on the results from the global analysis and for the extreme force of 800 kN, a hydraulic cylinder with the bore of 180 cm and rod diameter of 100 cm operating at 320 bar pressure is suitable [20]. If lower operating pressure, for instance 210 bar is chosen, a cylinder with piston diameter of 220 cm and bore of 160 cm must be used. The eye or clevis mountings at both ends is recommended to avoid bending on housing. This should be fitted with supporting beam to avoid jack-knife failure. The fatigue life assessment of this design will be addressed in future works.

Hydraulic control system design

The gripper mechanism as mentioned earlier has two main tasks: holding the monopile during hammering, and correcting the inclination angle. The vessel motions - in case of floating vessel - are also transferred to the monopile through the hydraulic actuators. These motions must be either compensated through a compensation system or the hydraulic system must be designed to operate with these motions. This can be done through an open loop system with the feedback from the operator (the easiest and most common method) or via a closed loop system. Moreover, the installation itself can also be automated - i.e. the inclination angle can be corrected automatically. For example high resolution cameras can be used to monitor the inclination angle, and to compensate the vessel motion via an integrated motion control system, like the Fugro Inclino system [21]. Other design concepts include motion compensated gripper which de-couples the vessel motions from the gripper in floating vessels [22].

Common industrial practice is to employ four hydraulic cylinders as shown in Fig. 2. Today's industrial practice is to keep the hydraulic cylinders in contact with the monopile through the whole phases of installation even during the hammering process. This approach reduces cylinder's life significantly due to bending moments applied on the rod and oil packing system. Industrial experiences show oil leakage after short time of operation, primarily caused by this design flaw.

One solution to overcome this design problem is to lock the connecting beam between the cylinder and the monopile. Normally, the hydraulic cylinder is used to move (through a piston rod) a sliding beam, which moves along a guide - see Fig. 6 (a). When the monopile inclination is corrected, the hydraulic cylinder is locked and the hammering process continues. This design exposes the structural integrity of all hydraulic system components all the time while the gripper is connected to the MP. The hydraulic system should be used to correct the inclination of the MP, but the system should be disconnected during the hammering process.

To cope with this design flaw, a simple modification of the hydraulic system can be implemented. Fig 6 (b) shows a schematic view of an alternative design. The hydraulic cylinder is used to correct the monopile inclination, and after that a hydraulic or mechanical system located at the back of the sliding beam (see detail X) can be used to lock this beam to the guide. This simple modification can extend the life of the hydraulic system components by reducing the dynamic force acting on cylinders during the hammering process.

The hydraulic control system can be designed in a way that

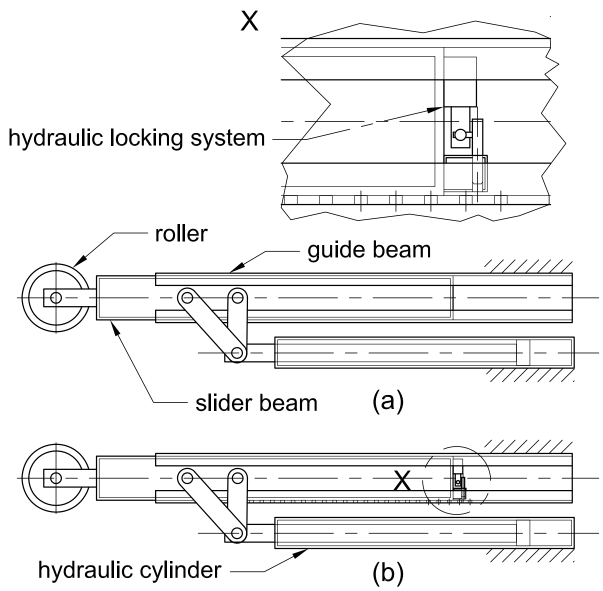

FIGURE 6: Schematic view of the hydraulic system used to correct MP inclination and hold MP in position. (a) Current design; (b) Alternative design.

both hammering phase and correction phase are addressed. Fig. 7 shows a typical design for gripper hydraulic system - note that the design is identical for all four cylinders. The 4/3 directional control valve is considered to have closed ports in the middle position to hold the MP during hammering. This should be used with the rolling beam shown in Fig. 6 (b), where an additional locking mechanism is considered to avoid cylinder damage. An accumulator and a auxiliary pump are also seen in this design which are discussed in the Step 3. A flow control valve at the pressure line is often needed to control the operation speed which is not shown in this figure.

It is important to note that the marine operations are subject to failure of system components. During monopile installation, it is possible that a mooring line breaks or the vessel experiences a sudden increase of wind or current loads and consequently the gripper contact force increases. These load cases can be analysed from global dynamic analysis from which the dynamic responses can be assessed. A practical and cost effective approach is to use control barrier in case of such accidental cases. For example a pressure relief valve or pressure switch on supply line of hydraulic cylinder is a practical solution. Another option is to use structural elements in supporting structures protecting the hydraulic actuators.

Step 3: Dynamic analysis of hydraulic system

The operational speed of a hydraulic cylinder depends on the fluid rate delivered from the hydraulic pump. The time that one cylinder reaches its end stroke is calculated from:

$$t = \frac{L\pi D^2}{4Q_{in}} \tag{3}$$

where Q_{in} is the input flow rate on one cylinder, D is the cylinder bore diameter, and L here is the cylinder stroke. Assuming that at any time only two cylinders are in operation - note that each two opposite cylinders can not operate simultaneously - the theoretical delivery of the pump, Q_P, is equal to $2Q_{in}$ and the theoretical pump displacement, D_P, is obtained from:

$$D_P = \frac{Q_P}{n_P} \tag{4}$$

in which n_P is the pump rotational speed. The term theoretical means that the efficiency is 100%. The power pack theoretical power is calculated from:

$$Power = \frac{D_P P_P n_P}{2\pi} \tag{5}$$

P_P in this equation is the pump output pressure.

For the case study problem, a power pack system and actuators are designed and presented in Table 2.

TABLE 2: Hydraulic power pack and actuator specification.

Parameter	Value	Unit
Cylinder bore dia., D	220	cm
Cylinder rod dia., d	160	cm
Cylinder stroke, L	20	cm
Time to reach end of stroke, t	5	sec.
Operating pressure, P_P	210	bar
Pump delivery, Q_P	182.5	l/min
Pump speed, n_P	1500	rpm
Theoretical power	64	kW

Since the loading varies largely in different environmental conditions, it is important to study the dynamic behaviour of the hydraulic system. This can be done by setting up an analytical model using the state equations of the system [23] or develop a model in commercial software like MATLAB, or 20-sim [24] as shown in Fig. 7. Such models are specially important when a fully automated system is considered.

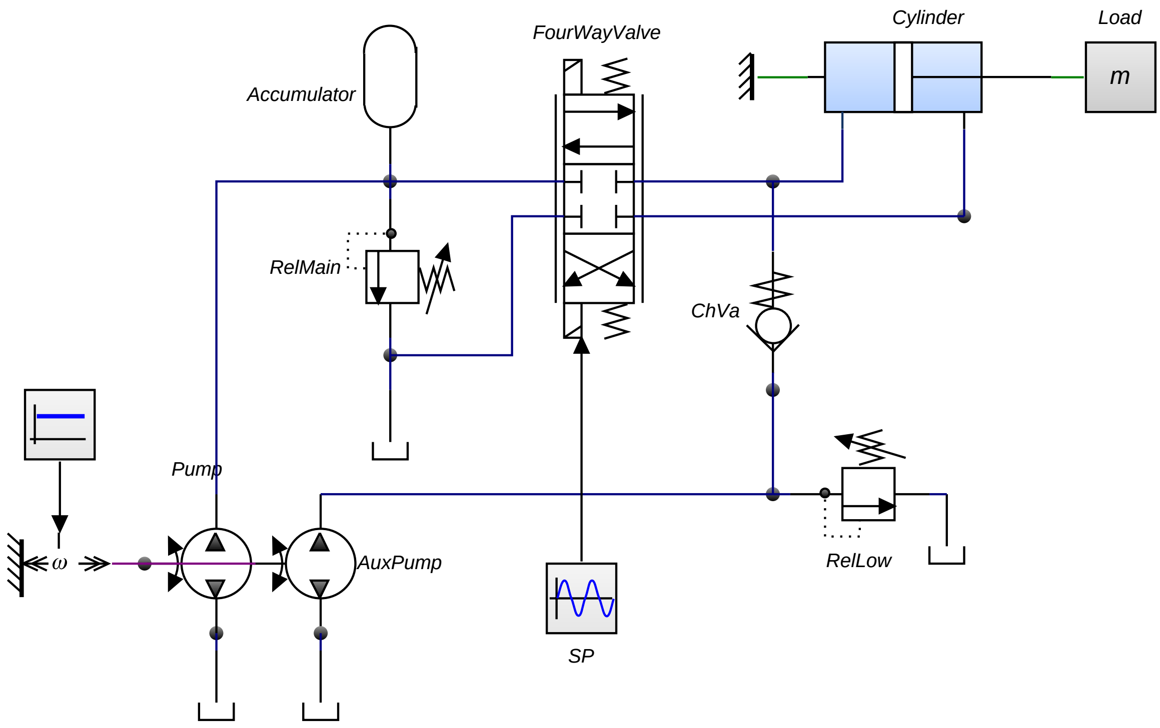

FIGURE 7: Sketch of the hydraulic system model.

Figures 8 and 9 illustrate the pump pressure, and required power variation for a given environmental condition and three penetration depths. The fluctuation observed in these figures occurs during the MP correction activities.

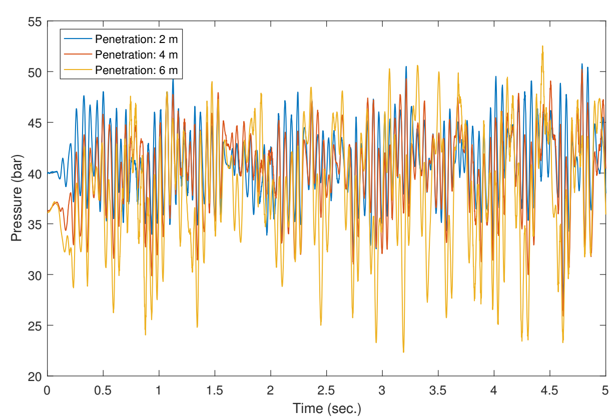

FIGURE 8: Pump pressure in environmental condition of, Hs=1.75 m , Tp=7.5 s, and different penetrations.

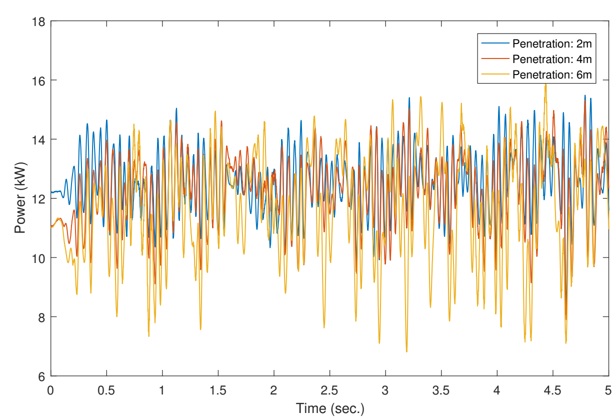

FIGURE 9: Hydraulic power variation in environmental condition of, Hs=1.75 m , Tp=7.5 s, and different penetrations.

High pressure variation observed in Fig. 8 reduces fatigue life of the actuator's components, in particular rings and seals. This can cause hydraulic oil leakage during operation. Higher variations are observed in higher penetrations. One solution to reduce the pressure variation is to correct the inclined angle more often. In higher penetrations it is therefore recommended to hammer the monopile in shorter spans and measure the inclined angle more often. For example inclined angle measurements can be for every 1 m hammering until penetration depth of 2 m, but from 2-6 m, it should be carried out in shorter spans, say for instance after every 50 cm hammering.

From the hydraulic system design side, the high pressure variation can be partially compensated by use of pressure accumulators on the pressure line as shown in Fig. 7.

The power variation, seen in Fig. 9, can be reduced by using an axillary pump. This large variation in power, primarily due to the vessel motions, must be considered in the power management of the vessel during the marine operation. It should be noted that in this paper, the valve dynamic, effect of fluid compressibility, and generated heat and heat transfer are not considered in this local hydraulic model, which are planned for the future works.

DISCUSSIONS & CONCLUSION

In this paper a systematic method was presented to design the hydraulic system used in gripper mechanism in installation of monopiles for offshore wind turbines. Monopiles are held by gripper device during hammering and the hydraulic cylinders are used to correct the inclination angle during installation operation. The force on the hydraulic cylinders is a function of wave height and period as well as the monopile penetration depth. As the depth increases, larger force is needed to correct the inclination angle, which is normally within the limit of 0.25 degree. The proposed design method starts with the global analysis where the forces on hydraulic cylinders are obtained for different weather conditions and penetrations. In the next step the hydraulic cylinders are chosen based on the limit sate design approach. The accidental forces should also be taken into account in global analysis. The hydraulic control system can be designed in a way to reduce the impact of the accidental cases. The last step is to study the dynamic behaviour of the hydraulic system through a local model. The proposed systematic method was illustrated through a case study of a monopile installation by means of a floating vessel.

In the proposed method, since the long-term load and load effects, both in terms of extreme and fatigue limits, are considered in designing the system, it is expected to achieve a system with higher reliability than deterministic method where design is based on empirical safety factors. The results from this approach can also be post-processed for fatigue analysis of the supporting structures. Moreover the dynamic model of the hydraulic system provides valuable information in terms of load variation as well as power variations. They can also be used to optimize the hydraulic system design in terms of fatigue life calculation, reliability and power loss. This data can be used in overall power management of the vessel during marine operation. In addition the results from this study will be further employed to build reliability analysis and life prediction models of hydraulic components as a part of reliability-based design of mechanical systems in marine applications - see for instance [25–27].

Future work is also devoted to study a more detailed model of the hydraulic system including the valve dynamics, fluid compressibility and heat transfer in the system.

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