Design and modeling of active heave compensation system

Introduction: The overall purpose of the project is to do virtual design, simulation and control of a hydraulically actuated mechanical system subjected to dynamic loading. This is formulated as a design task for the drawwork concept shown in Fig. 1.

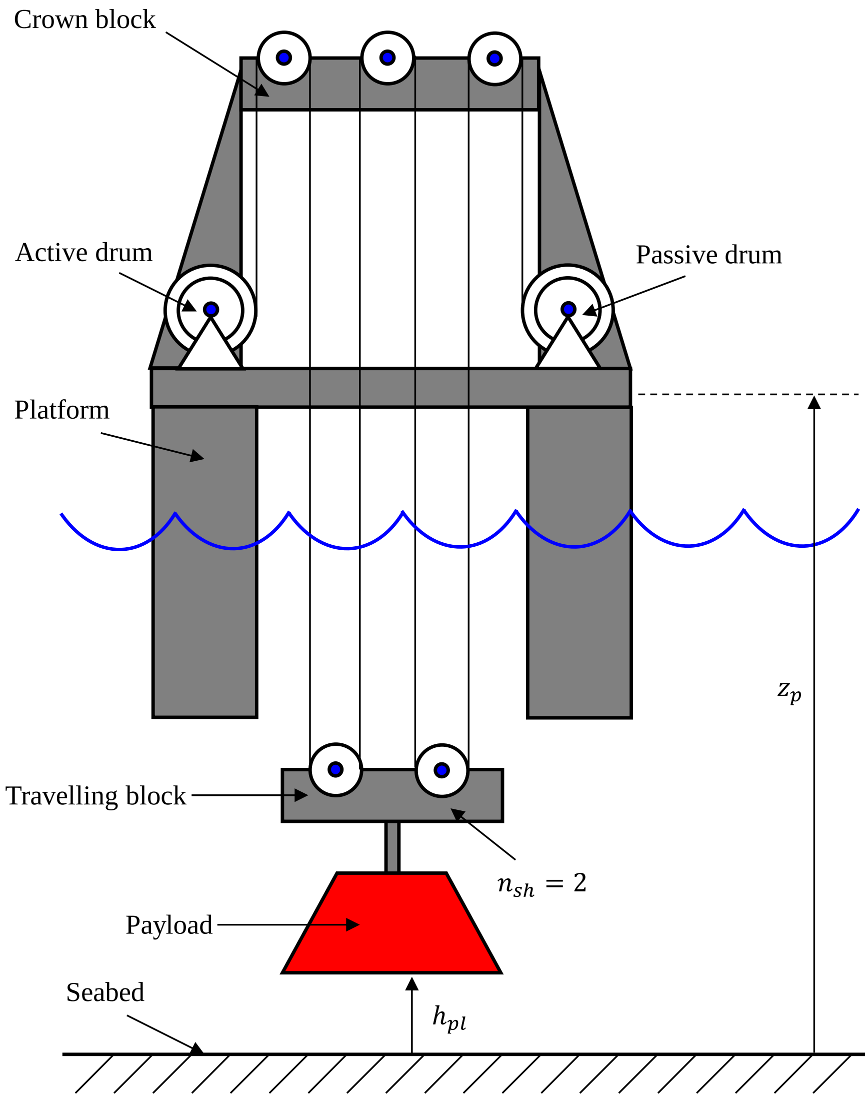

Figure 1 Design concept of drawwork, in this picture with $n_{sh} = 2$.

The main task of the drawwork is the lowering and hoisting of payloads to the seabed. When the payload is near the seabed the drawwork should be able to compensate for wave motion, i.e., land the payload as softly as possible on the seabed even though the vessel carrying the drawwork is subjected to wave induced heave motion. This manipulation of the payload should be done by means of the active drum. The intended power transmission is shown in Fig. 2.

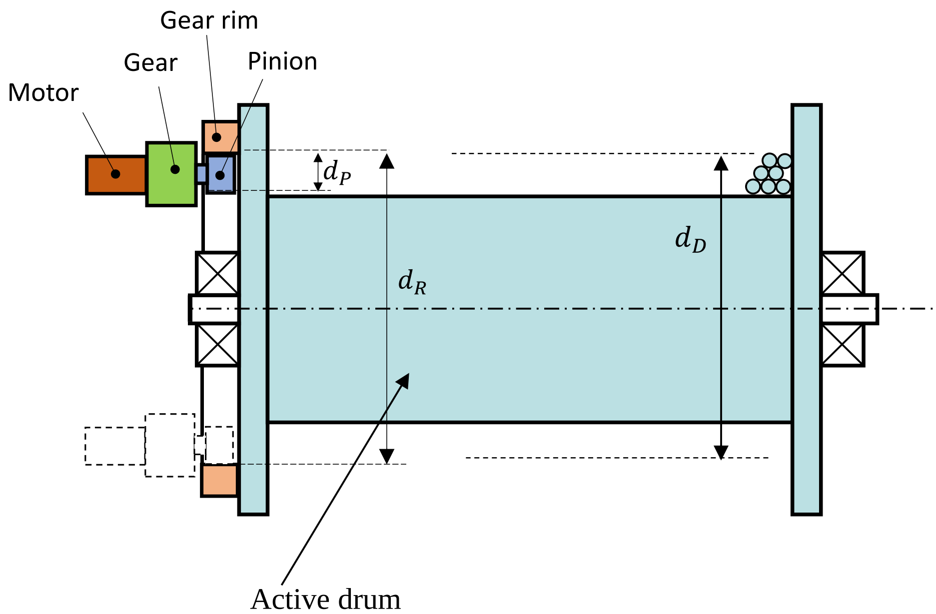

Figure 2 Power transmission, two gearmotors are shown.

The hydraulic gearmotors drive the drum via pinion - gear rim connections.
The hydraulic motors are driven by a hydraulic circuit, see Fig. 3, where both the number of motors, the number of servo valves and the number of hydraulic power units are to be determined.

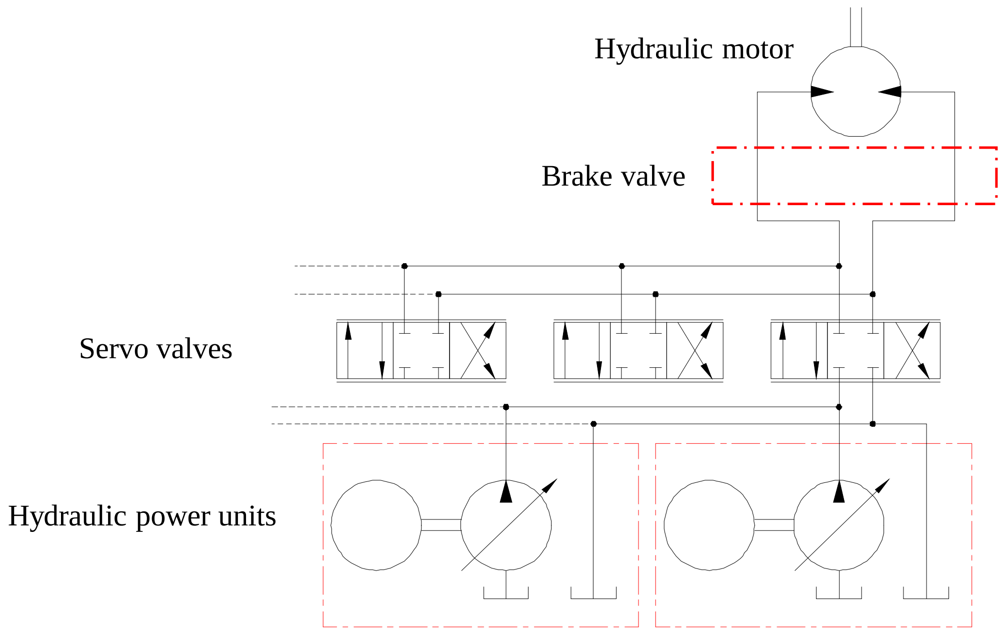

Figure 3 Hydraulic circuit – in this picture only a single motor.

Load case: A single load case must be investigated, see Tab. 1. It is described by parameters that define the absolute platform motion, z_p, see Fig. 4:

$$z_p = z(t) = z_w \cdot sin\left(\frac{2 \cdot \pi}{T_w} \cdot t\right) \tag{1}$$

The main task for the heave compensation system is to keep the payload at the same vertical position at all times, i.e., the payload is not being lowered og hoisted. Also, the distance to the seabed, h_{pl}, is so large that there is no danger of collision between payload and seabed.

Further, the external load on the motor shaft, M_M, can be computed according to the number of motors, n_M, the number of sheaves, n_{sh}, the ratio of the gearbox, i_g, the mass of the payload, m_{pl}, the drum diameter, d_R, the gear rim diameter, d_P, the pinion diameter, d_P, and the total equivalent friction coefficient, μ_{eq}. It can be assumed that the motors divide the load evenly.

$$M_M = \frac{m_{pl} \cdot g \cdot d_D \cdot d_P}{2 \cdot n_{sh} \cdot d_R \cdot 2 \cdot i_g \cdot n_M} \cdot \left(1 + \mu_{eq} \cdot \tanh\left[\frac{\omega_M}{\omega_0}\right]\right) \quad (2)$$

The mass of the payload includes the wire mass, mass of the travelling block etc. Also, the equivalent friction includes drag in the seawater, hydromechanical losses in the hydraulic motors and friction losses in the mechanical transmission (sheaves, gearbox, drum bearings, wire, etc.). The equivalent friction torque changes sign with the speed of the hydraulic motors, ω_M, and must be modeled so that it always acts against the motion.

Table 1 Load case data

z_w $[m]$	T_w $[s]$	n_{sh} $[-]$	i_g $[-]$	m_{pl} $[kg]$	d_D $[m]$	d_R $[m]$	d_P $[m]$	μ_{eq} $[-]$	ω_0 $[\frac{rad}{s}]$
1.2	10.0	3	7	24000	0.45	0.5	0.15	0.15	5

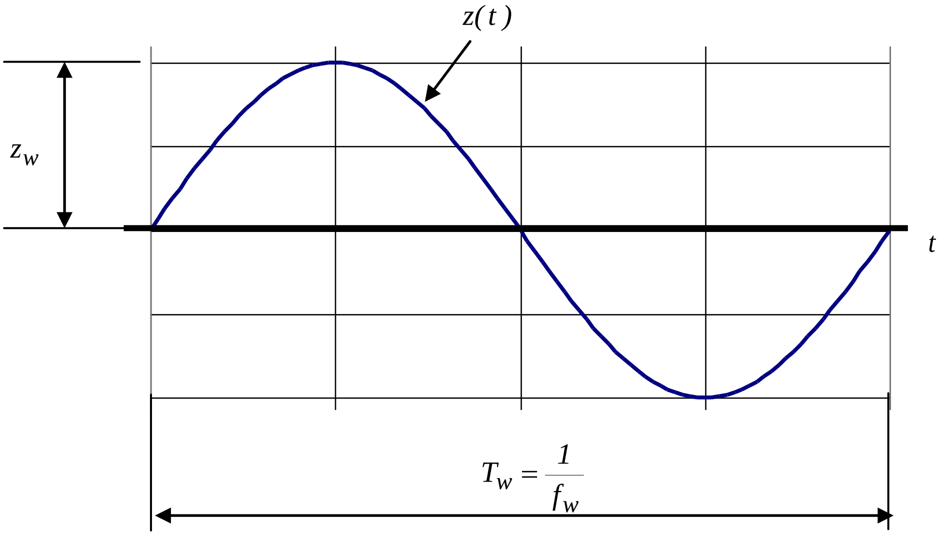

Figure 4 Illustration of wave motion.

Design criteria: Two design criteria are of importance:
- Costs should be minimized.
- Positioning accuracy should be optimized

Design parameters: The design parameters are limited to the hydraulic system:
- Hydraulic motor(s), i.e., quantity and type (the motors must be identical)
- Brake valve, i.e., type (optional)
- Servo valve(s), i.e., quantity and type(s)
- Hydraulic power unit(s), i.e., quantity and type(s)

The number of motors, servo valves and hydraulic power units are not fixed. The components must be selected from a limited choice of commercially available types described in the following PDF-files: *FixedMotor.pdf*, *BrakeValve.pdf*, *ServoValve.pdf*, and *HydraulicPowerUnit.pdf*.

Control
One or more servo valves together with the variable displacement pump of the hydraulic power unit(s) constitute the control elements. This means that there are two control signals: a common spool travel reference to the servo valves and a common pressure reference signal to the HPUs.
Feedback sensors include:

- vertical acceleration, velocity and position of the platform, $\ddot{z}_p$, $\dot{z}_p$ and z_p.
- angular velocity of hydraulic motor, ω_M

The sensors may be considered ideal, i.e., infinitely fast and accurate.

Hydraulic motor(s):
The hydraulic motors must be selected from the catalogue *FixedMotor.pdf*. They are not subjected to any radial force on their outputshafts but only torque. Selecting a suitable motor may include selecting a brake valve from *BrakeValve.pdf*, but not necessarily.
Because the hydromechanical losses are taken into account in the equivalent friction in Eq. (2) the hydromechanical efficiency of the motor should be ideal, i.e., $\eta_{hmM} = 1$. The volumetric losses of the motors should be modeled by means of a leakage orifice (in parallel with the motor) that ensures a volumetric efficiency $\eta_{vM} = 0.94$ at maximum speed (1600 ... 10000 rpm depending on motor size) and at a pressure difference of $\Delta p = 350\ bar$.
The dimensionless cost of the motors is given as:

$$C_M = w_M \cdot n_M \cdot \left(1 + \frac{D_M}{D_{M,max}}\right) \quad w_M = 2.0 \qquad (3)$$

where $D_{M,max} = 1000\,\frac{cm^3}{rev}$ is the maximum motor displacement and n_M is the number of motors.

Servo valve(s):
The servo valve(s) must be selected from the catalogue *ServoValve.pdf*. Assume that all electronics and power electronics is available so that a reference value can be computed and submitted to the valve infinitely fast. The valve dynamics is described by a transfer function between actual spool position and reference spool position (as computed by the control system). The transfer function should be 2nd order and the natural frequency, ω_v , and the dimensionless damping, ζ_v, should obtained from the catalogue.
The dimensionless cost of the servo valve(s) is given as:

$$C_{SV} = w_{SV} \cdot n_{SV} \cdot \left(1 + \frac{Q_{NL}}{Q_{NL,max}}\right) \quad w_{SV} = 2.5 \qquad (4)$$

where $Q_{NL,max} = 200\,\frac{l}{min}$ is the maximum nominal valve flow and n_{SV} is the number of servo valves.

Hydraulic power unit(s):
The hydraulic power unit(s) must be selected from the catalogue *HydraulicPowerUnit.pdf*. The HPU must be configured with a pressure control pump. Assume that the cooling capability corresponds to 50% of nominal power. Assume that all electronics and power electronics are available so that a reference value can be computed and submitted to the pressure-controlled pump infinitely fast. The HPU pump dynamics is described by a transfer function between actual pump pressure and reference pump pressure (as computed by the control system). The transfer function should be 2nd order with a natural frequency of $\omega_{HPU} = 6\,\frac{rad}{s}$ and a dimensionless damping of $\zeta_{HPU} = 1$.
The dimensionless cost of the HPUs is given as:

$$C_{HPU} = w_{HPU} \cdot n_{HPU} \cdot \left(1 + \frac{P_{HPU}}{P_{HPU,max}}\right) \qquad w_{HPU} = 3 \qquad (5)$$

where $P_{HPU,max} = 132\ kW$ is the maximum nominal power and n_{HPU} is the number of HPU's.

Project tasks

- Design the hydraulic system by choosing motor(s) (with or without brake valves), servo valve(s) and hydraulic power unit(s) and setting up a control scheme.
- Verify the performance of the system by means of one or more simulation models in Matlab/Simulink/Simscape.
- Introduce one or more failures during operation and simulate the consequences.
- Document the work in a report with a maximum of 15 pages in the main report (appendices can be added).