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.

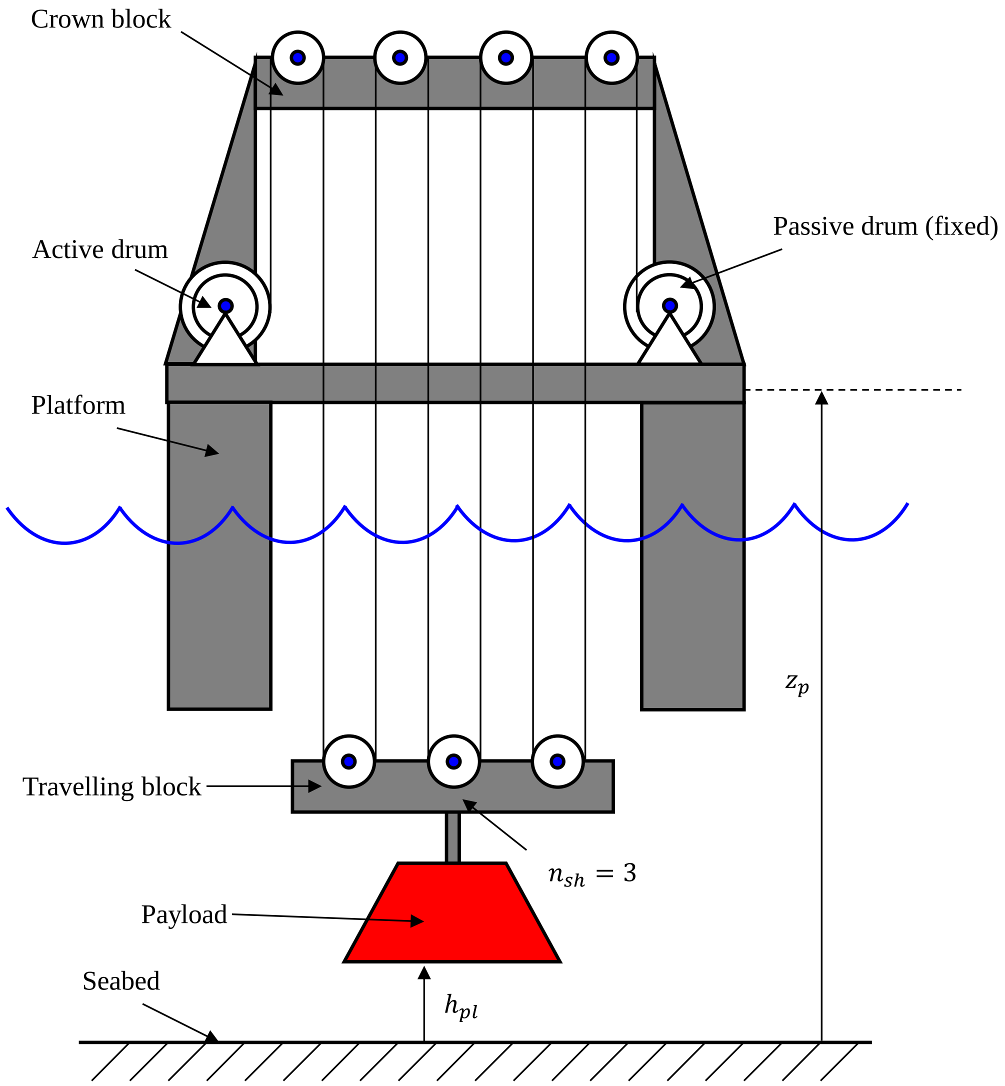

Figure 1 Design concept of drawwork, in this picture with $n_{sh} = 3$.

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.

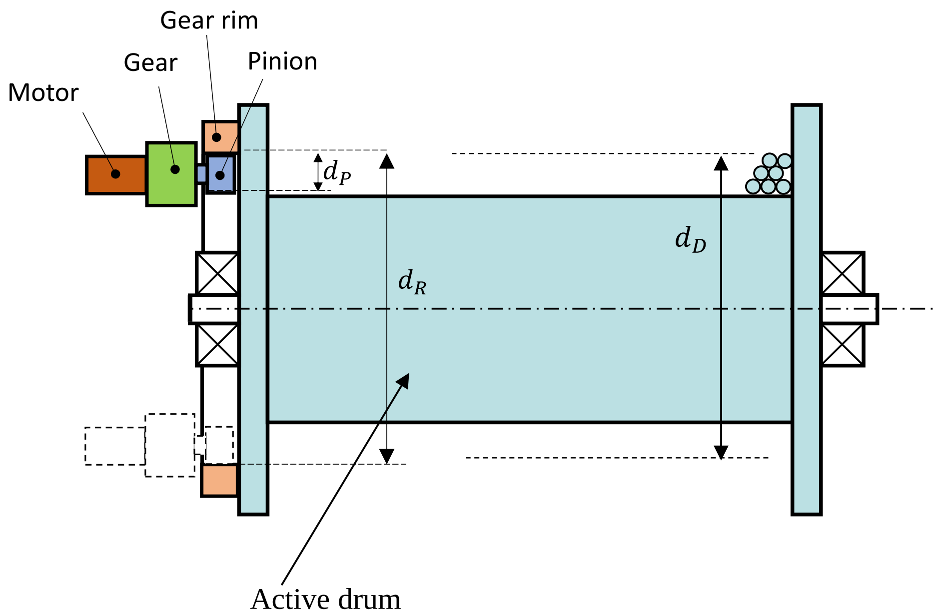

Figure 2 Power transmission, two gearmotors are shown.

Each hydraulic motor is mechanically connected to the drum via a gearbox and a pinion the is connected to the gear rim of the drum. The hydraulic motors can be driven by two different types of hydraulic circuit, see circuit A in Fig. 3 and circuit B in Fig. 4. Circuit A uses servo valves and circuit B uses pressure compensated proportional valves and counterbalance valves.

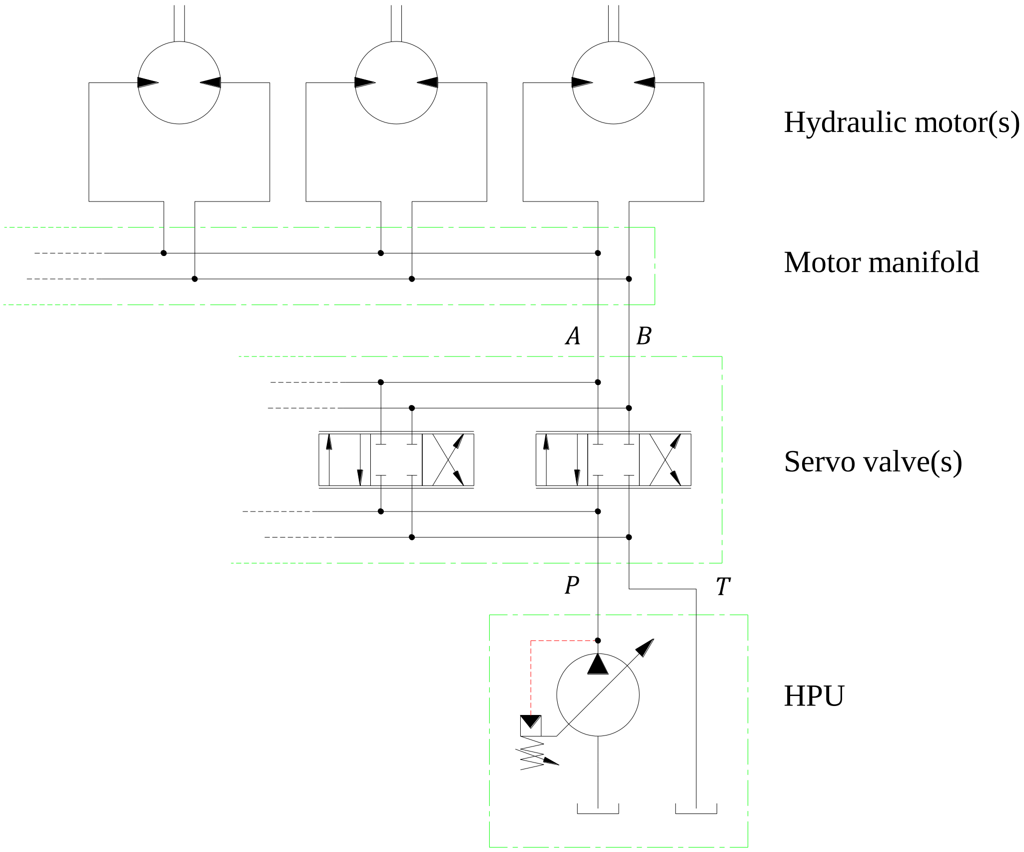

Figure 3 Hydraulic circuit A. It shows the overall architecture and how similar components should be connected in parallel.

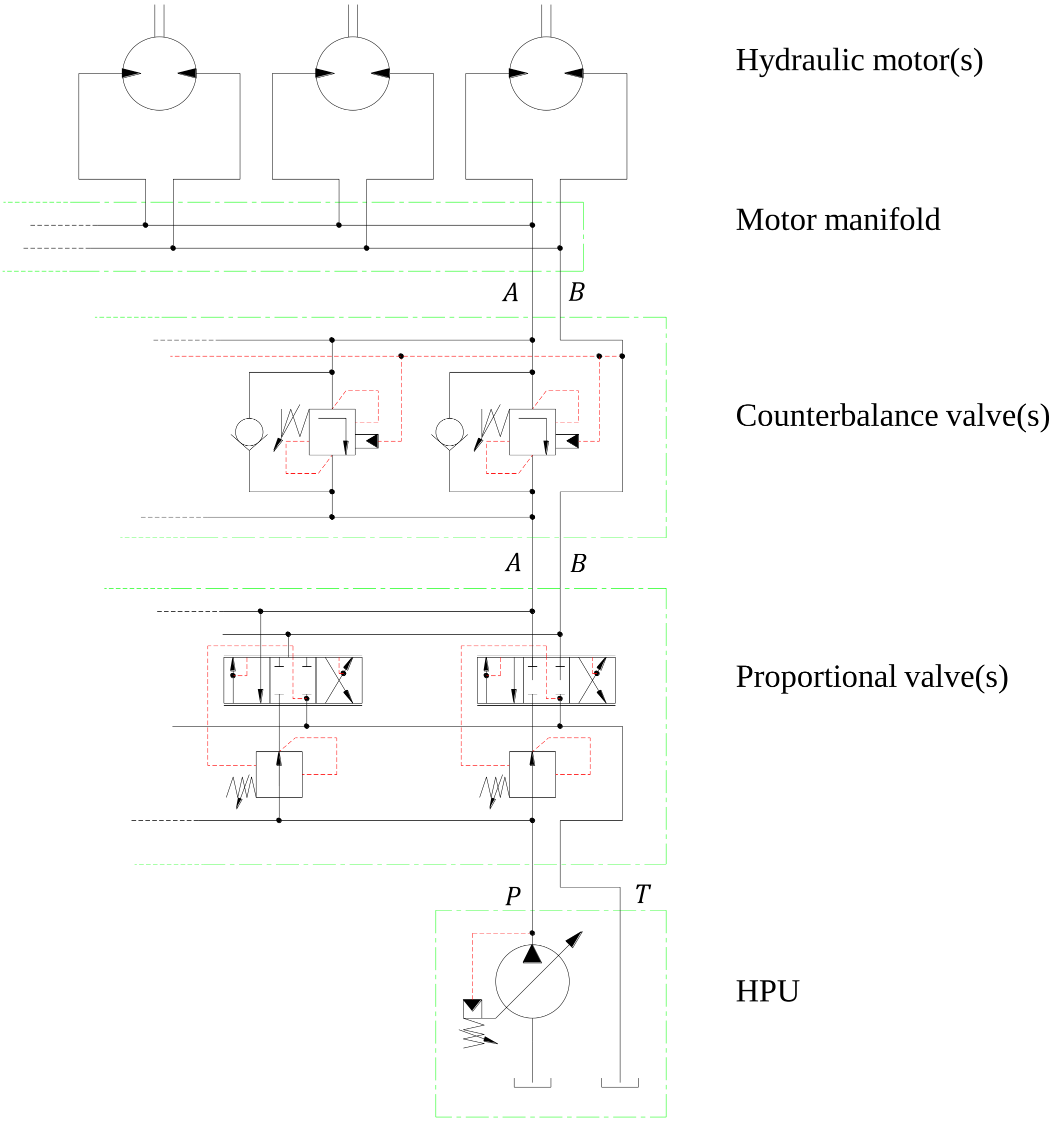

Figure 4 Hydraulic circuit B. It shows the overall architecture and how similar components should be connected in parallel.

Two design must be developed, one using circuit A and one using circuit B.

- In both design (A) and (B) the type and number of motors must be determined.
- In circuit A the the number and type of servo valves must be determined.
- In circuit B the number of proportional valves, the number and type of counterbalance valves must be determined.
- In both design (A) and (B), the pressure setting of the hydraulic power unit must be determined.

The hydraulic motors must be the same type and size. If more than one component is chosen they must be connected in parallel as shown in Fig. 3 and Fig. 4.

NOTE: It is not a requirement that there is more than one motor, more than one servo valve, more than one proportional valve, or more than one counterbalance valve.

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.

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_R, the pinion diameter, d_P, and the total equivalent friction coefficient, μ_{eq}. It can be assumed that the motors divide the load evenly.

$$M_M = \frac{m_{pl} \cdot g \cdot d_D \cdot d_P}{4 \cdot n_{sh} \cdot d_R \cdot i_g \cdot n_M} \cdot \left(1 + \mu_{eq} \cdot \tanh\left[\frac{\omega_M}{\omega_0}\right]\right) \tag{2}$$

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

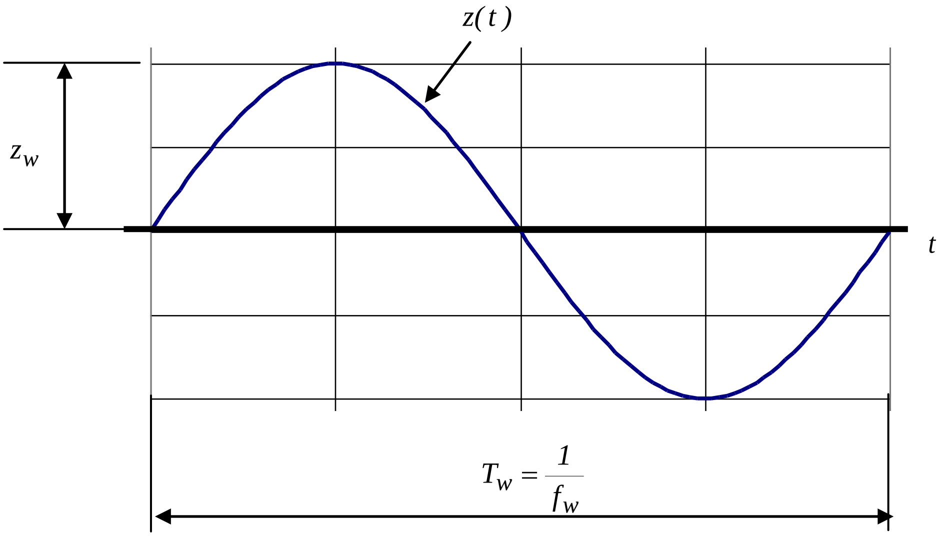

Figure 4 Illustration of wave motion.

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

Design parameters: The design parameters are limited to the hydraulic system:
- Hydraulic motor(s), i.e., quantity and type (the motors must be identical)
- Servo valve(s) (circuit A), i.e., quantity and type
- Proportional valve(s), (circuit B), i.e., quantity and type(s)
- Counterbalance valve(s), (circuit B), i.e., quantity and type(s).

The number of motors, servo valves, proportional valves and counterbalance valves are not fixed. The components must be selected from a limited choice of commercially available types, see further down.

Control

In circuit A, the control element is the servo valve and in circuit B the control element is the proportional valve. The pump pressure is adjustable, however, not during operation.
This means that there are two control signals: a common spool travel reference to the servo valves (circuit A) or the proportional valves (circuit B).
Feedback sensors include:

- vertical acceleration, velocity and position of the platform, $\ddot{z}_p$, $\dot{z}_p$ and z_p.
- angular position and velocity of hydraulic motor, θ_M and 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 output shafts but only torque.
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 a motor depends on the motor displacement, D_M, and it is given as:

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

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

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 a servo valve depends on the rated flow, Q_{rat}, and it is given as:

$$C_{SV} = w_{SV} \cdot \left(1 + \frac{Q_{rat}}{Q_{rat,max}}\right) \quad w_{SV} = 6.0 \tag{4}$$

where $Q_{rat,max} = 1500\,\frac{l}{min}$ is the maximum rated servo valve valve flow.

Proportional valve(s):
The proportional valve(s) must be selected from the catalogue *ProportionalValve.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 a proportional valve depends on the nominal flow, Q_{nom}, and it is given as:

$$C_{PV} = w_{PV} \cdot \left(1 + \frac{Q_{nom}}{Q_{nom,max}}\right) \quad w_{PV} = 4.0 \tag{5}$$

where $Q_{nom,max} = 1150 \frac{l}{min}$ is the maximum nominal proportional valve valve flow.

Counterbalance valve(s):
The counterbalance valve(s) must be selected from the homepage of Sun Hydraulics:

https://www.sunhydraulics.com/models/cartridges/load-holding/counterbalance#%7B%22view%22:%22single%22,%22page%22:1%7D

The counterbalance valve must be of type CB**. The valve can be considered infinitely fast.
The dimensionless cost of a counterbalance valve depends on the nominal capacity (nominal flow), Q_{cap}, and it is given as:

$$C_{CBV} = w_{CBV} \cdot \left(1 + \frac{Q_{cap}}{Q_{cap,max}}\right) \quad w_{CBV} = 1.0 \tag{5}$$

where $Q_{cap,max} = 480 \frac{l}{min}$ is the maximum nominal capacity for a counterbalance valve.

Hydraulic power unit(s):
The hydraulic power unit (HPU) can be considered an ideal pressure controlled HPU. It has a variable displacement pump that will deliver any flow up to $Q_{P,max} = 2000 \frac{l}{min}$. The pressure can be adjusted to any valve up to $p_P < p_{P,max} = 240\ bar$.
It can be assumed that the pump will not overheat.
The total steady state energy consumption over a wave period, E_{sys}, of the two design (circuit A) and (circuit B) must be computed based on the pump parameters, i.e.:

$$E_{sys} = \int_0^{T_W} p_P \cdot Q_P(t) \cdot dt \tag{5}$$

The cost of the HPU is not included in the overall costs of the two design.

Project tasks

- Design the two hydraulic system, circuit A and circuit B, by choosing motor(s), servo valve(s) or proportional valve(s) and counterbalance valve(s).

- Set up a control scheme that minimizes the absolute motion of the payload in the water.

- 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).