STEM Games 2019

Engineering

UNDERWATER HABITATS

a problem by

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1 Introduction

The choice of propulsion system type is one of the most important parts in submarine's design process since it determines the ability of fulfilling its purpose. Submarines that perform underwater research activities require endurance and certain flexibility in different littoral conditions. For example, conventional motor propulsion systems, such as diesel electric, when operational, require fresh air intake for driving electrical generator. In this way, a period while a submarine can remain submerged is limited by the power and endurance of the battery system and counts in hours or few days at most. Therefore, when it comes to endurance such systems might not be an optimal choice. Technology that overcomes this drawback is a nuclear power system which offers better endurance (even few months) and speed of the submarines but there are number of disadvantages (very expensive, operational limitations in shallow littoral waters etc.) that make them unsuitable for underwater research activities. The most suitable seems to be so called Air Independent Propulsion systems (AIP systems) which are marine propulsion technologies that allows non-nuclear submarine to operate without access to atmospheric oxygen.

One of the most interesting is definitely French AIP technology so called MESMA (Module d'Energie Sous-Marine Autonome). This is the AIP system that uses closed-turbine cycle for powering electrical generator and the heat for steam generation is provided by liquid ethanol (ili bioethanol) combustion in pure oxygen (sometimes mixed with argon). Combustion of pure ethanol (ili bioethanol) makes this system ecologically much more acceptable than other systems that use conventional fuels. In this way, virtually silent MESMA exploits all advantages of AIP systems for maintenance of underwater habitats and performing underwater research or training activities.

Today's assignment is to design a MESMA power generation system for submarine propulsion for a given submarine hull which consists of outer and inner shell, showed on Figure 1. Inside inner shell there is a reserved space of fixed volume (zadati ovdje ili zajedno sa skicom!) consisting of three compartments; fuel tanks compartment (Compartment 1), engineering room compartment (Compartment 2) and cargo place compartment (Compartment 3). Individual volumes of each compartment can be arbitrarily chosen but the sum of all compartment volumes is fixed to a constant value.

The scheme of MESMA power generation system is given in the assignment description and its performance (range, speed, efficiency, price?) (dopuniti, ako treba, kada tablica bodovanja bude gotova) will be evaluated for different depths according to the scoring table (možda nije dobar naziv) given later in the text. It is important to emphasize that one of evaluation categories is the size of the cargo place compartment so the care must be taken not to oversize the propulsion system which must be placed in other two compartments.

The assignment can be divided into two steps:

1. Design and dimension main components of the propulsion system,

2. Define operational characteristics of some components.

After the first step you will be given a full model of propulsion system with the components you selected which will help you define operational characteristics of some components. Detailed description of the system components is given in the next section.

Engineers, good luck!

Zadatak, Your task is to design submarine power generation system which can generate sufficient energy to travel xx km and generate minimum required power during xx hours when submarine is being loaded and unloaded. or: to design power generation system which can generate sufficient energy to travel 100 km at depth 50 m 20 km at depth 100 m and 20 km at depth 10 m Formule za nosivost, masu, otpor, maksimalna duljina izmjenjivača, maksimalna snaga turbine i drugo ovisno o veličini podmornice

2   Power generation system description

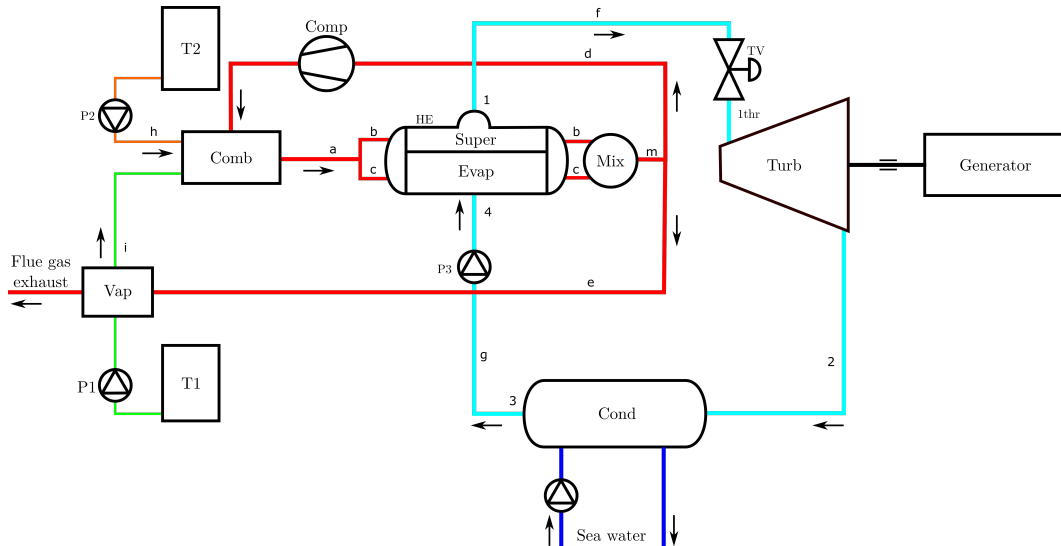Power generation system schematics are shown on figure 1.

Figure 1: Power generation system.

Submarine power generation system consists of two main circuits; flue gas circuit (red) and steam circuit (cyan). Steam circuit is used for steam production which expands in turbine "Turb" providing necessary power for the "Generator" used for charging the batteries for submarine propulsion. The flue gas circuit is used to provide necessary heat flow for evaporating and superheating the steam for the turbine. Flue gas is generated in the combustion chamber "Comb" in which energy is released by ethanol combustion in pure oxygen.

Liquefied oxygen and ethanol fuel are stored in separate tanks "T1" and "T2", respectively. While pump P2 delivers fuel stream "h" to combustion chamber "Comb", cryogenic pump P2 delivers liquefied oxygen stream "i" to oxygen vaporizer "Vap" where it vaporizes and enters combustion chamber "Comb" in stoichiometric ratio at which fuel combustion is, by assumption, complete without any excess oxygen. Flue gas temperature at the outlet from the combustion chamber (stream "a") is controlled by return stream of cooled flue gas "d" by compressor "Comp" and should never exceed maximum allowed flue gas temperature

Flue gas stream "a", which exits combustion chamber "Comb" enters heat exchanger "HE" used for water evaporation and steam superheating. Therefore, heat exchanger "HE" comprises two sections; an evaporation section "Evap" and superheating section "Super". Total flue gas mass flow which enters heat exchanger "HE" is divided into two flue gas streams "b" and "c". Stream "c" flows through evaporation ("Evap") section and stream "b" flows through superheating ("Super") section in parallel configuration. On heat exchanger exit, two flue gas parallel streams "b" and "c" are adiabatically mixed in mixing chamber "Mix" forming flue gas stream "m".Part of stream "m" is returned by compressor "Comp" to combustion chamber "Comb" as stream "d" to prevent overheating (maximum allowed flue gas temperature), while the rest (stream "e") is used for oxygen evaporation in heat exchanger "Vap" and evacuated outside submarine.

Heat exchanger "HE" is a pool boiling type apparatus which means that liquid water in the evaporating section is at rest and is evaporated by the flue gas that passes through a tube bundle in the evaporating section. The exact amount of steam that is evaporated in evaporation section "Evap" (which corresponds to the steam mass flow of stream "g") is then superheated in superheating section "Super" of heat exchanger "HE" producing the superheated steam "f". Superheated steam (stream "f") passes through throttling valve "TV" and enters the steam turbine "Turb" where mechanical power is produced by steam adiabatic expansion to condensation pressure p2. After the expansion, low pressure steam condenses in sea water cooled condenser "Cond" and is returned to high pressure evaporator "Evap" by condensate pump "P3".

The turbine drives an alternator-rectifier which supplies submarine propulsion system with direct current. (?) Required mechanical power for submarine propulsion and auxiliary systems should in every timestep be provided by turbine (no buffer). (dogovor s mentorskim teamom)

Following sections provide a detailed description of every part of power generation system that should be modeled.

2.1 Turbine

A Steam Turbine is a mechanical device in which thermal energy is extracted from pressurized steam by its adiabatic expansion and is transformed to mechanical work. Therefore, for a given mass flow rate of steam qm, power P generated by steam turbine is defined:

$$P = q_m(h_1 - h_2),\tag{1}$$

where

- q_m is steam mass flow rate,

- h_1 is steam specific enthalpy at turbine inlet,

- h_2 is steam specific enthalpy at turbine outlet.

Turbine steam swallowing capacity mass flow rate q_m at given pressures and inlet temperature can be expressed by Stodola's Ellipse equation:

$$q_m = \frac{K}{\sqrt{T}}(p_1^2 - p_2^2)^{\frac{1}{2}},\tag{2}$$

where:

- K is Stodola's coefficient,

- p_1 is turbine inlet pressure,

- p_2 is turbine outlet pressure,

- T is turbine inlet absolute temperature.

2.2 Steam turbine isentropic efficiency

Turbine isentropic efficiency η is defined as the ratio between enthalpy drop at actual turbine expansion and enthalpy drop at isentropic expansion between inlet and outlet turbine pressure p_1 and p_2.

$$\eta = \frac{h_1 - h_2}{h_1 - h_{2s}}.\tag{3}$$

Steam turbine isentropic efficiency depends upon mean steam volume flow rate qv according to the following equation:

$$\eta = aq_v^2 + bq_v + c,\tag{4}$$

where q_v is geometric mean of volume flow rates at turbine inlet ($q_{v,in}$) and at outlet in case of isentropic expansion ($q_{v,out,s}$).

$$q_v = \sqrt{q_{v,in} \cdot q_{v,out,s}},\tag{5}$$

Turbine efficiency is highest at its nominal design point for which pressures (p_1, p_2) and inlet temperature T1 are given in table 1, while nominal flow rate is defined by Stodola's ellipse equation 2. Turbine can operate at other conditions (off design turbine performance with lower, i.e. higher mass flow rates than the nominal) with the cost of reduced efficiency, provided that these conditions are not outside permitted values.

2.3   Off design turbine performance

Lower mass flow rate

Steam throttling is a common way for reduction of power produced by steam turbine. By reducing fuel pump "P2" and oxygen pump "P1" speed accordingly when power demand is lowered, fuel consumption and evaporated steam mass flow rate are decreased. In conjunction with steam flow rate decrease, throttling valve "TV" orifice is constricted to reduce turbine inlet pressure $p_{1,thr}$ to a value which satisfies Stodola's Ellipse law 2 for given decreased mass flow rate q_m. In a throttling process, superheated steam which exits heat exchanger "HE" is throttled to lower pressure $p_{1,thr}$, while steam enthalpy is conserved.

$$h_{1,thr} = h_1, \tag{6}$$

$$p_{1,thr} = p_1, \tag{7}$$

$$T_{1,thr} = T(p_{1,thr}, h_1) < T_1. \tag{8}$$

Side effect of throttling is a temperature reduction which corresponds to the state of conserved enthalpy h_1 at throttled pressure $p_{1,thr}$. Pressure and temperature are both part of a Stodola's Ellipse law 2 which means they need to simultaneously satisfy Stodola's Ellipse law 2 and enthalpy conservation of a throttling process from state "1".

By throttling, turbine inlet pressure and inlet temperature are reduced which decreases available isentropic power of turbine. Turbine efficiency described by equation 4 is also decreased at partial turbine load.

Higher mass flow rate

In case that power demand is increased, turbine can be overloaded by steam mass flow q_m greater than nominal. This is achieved by fuel pump "P2" and oxygen pump "P1" speed increase which causes increased fuel consumption and evaporated steam mass flow. When turbine is overloaded, steam flow bypasses few initial turbine stages which enables increased steam consumption, although with reduced turbine efficiency described by equation (4). To accurately model this effect is out of scope of this assignment. Therefore, it is assumed in overload working mode that turbine inlet values of pressure and temperature remain nominal, i.e. there is no change at throttling valve due to increased steam mass flow rate q_m. The only effect of increased q_m is turbine efficiency reduction as described by equation 4.

Restrictions of steam turbine operational parameters are following:

- Steam temperature at turbine inlet should not exceed temperature $\vartheta_{turb,max}$,

- Define operational characteristics of some components.

Turbine design parameters are given in **??**.

2.4   Heat exchangers

Heat exchanger is a device whose purpose is to transfer heat from warmer fluid stream to colder stream. There are three heat exchangers in the power generation system: heat exchanger "HE" which is divided to superheating and evaporative section, condenser "Cond" and oxygen vaporizer "Vap". Condenser "Cond" and vaporizer "Vap", whose performance will not be evaluated in this task are assumed to be regulated accordingly in each time step with the current need of the power generation system.

Heat exchanger "HE"

Heat exchanger "HE" is used for steam evaporation and superheating in two corresponding sections by hot flue gas from combustion chamber ("b" and "c" stream). It is shell and tube type with flue gas passing through tubes in each section.

Heat flow exchanged in each of two heat exchanger sections is described as:

$$\Phi = A \cdot k \cdot LMTD, \tag{9}$$

where:

$LMTD$ is logarithmic mean temperature difference, k is overall heat transfer coefficient, A is total heat exchange area of the section. Assume superheating section of heat exchanger is counter-flow.

In addition to LMTD method, rate of heat transfer can be determined using NTU method. For counter-flow heat exchanger effectiveness is calculated as:

$$\pi_1 = \frac{1 - exp((1 - \pi_3)\pi_2)}{1 - \pi_3 exp(-(1 - \pi_3)\pi_2)}, \tag{10}$$

Dimensionless parameters π_1,π_2,π_3 are defined as follows:

$$\pi_1 = \frac{\theta_1' - \theta_1''}{\theta_1' - \theta_1'}, \tag{11}$$

$$\pi_2 = \frac{kA}{C_1}, \tag{12}$$

$$\pi_3 = \frac{C_1}{C_2}, \tag{13}$$

where C is stream heat capacity.

Index 1 corresponds to stream with lower heat capacity, index 2 to stream with higher heat capacity, superscript $'$ to stream inlet state in heat exchanger and $''$ to outlet state.

Flue gas, which is divided into two streams "b" and "c" enters evaporation "Evap" and superheating "Super" section at equal temperature and flows through tubes in one pass. Flue gas mass flow through evaporative and superheating section of heat exchanger is defined as:

$$q_{m,fg,evap} = evap_{fr} \cdot q_{m,fg}, \tag{14}$$

$$q_{m,fg,super} = (1 - evap_{fr}) \cdot q_{m,fg}, \tag{15}$$

Where: $evap_{fr}$ is flue gas fraction of stream "c" to total flue gas stream "a" total flue gas mas flow and is assumed constant throughout system operation. Its value can be selected between 0 and 1. $q_{m,fg}$ is total flue gas mass flow rate (stream "a") . $q_{m,fg,evap}$ is evaporation section mass flow rate (stream "b") $q_{m,fg,super}$ is superheating section mass flow rate (stream "c") Heat transfer coefficient on tube side can be calculated by using following simplified convection models.

If the flow is transient or turbulent $(2300 < Re < 5e6)$, mean Nusselt number is defined as:

$$Nu = \frac{f/8 \cdot (Re - 1000) \cdot Pr}{1 + 12.7\sqrt{f/8} \cdot (Pr^{2/3} - 1)}, \tag{16}$$

where f is the friction factor, defined as

$$f = \frac{1}{(1.82 \log_{10} Re - 1.64)^2}. \tag{17}$$

If the flow is laminar $(Re < 2300)$, mean Nusselt number is defined as:

$$Nu = 1.86 \left(Pe \frac{d}{L} \right)^{1/3}. \tag{18}$$

In above equations non-dimensional numbers are

- $Nu = \frac{\alpha d}{\lambda}$ Nusselt number

- $Re = \frac{\rho w d}{\mu}$ Reynolds number

- $Pr = \frac{\mu c_p}{\lambda}$ Prandtl number

- $Pe = RePr$ Peclet number

Nusselt, Prandtl and Reynolds numbers and flue gas velocity are calculated using mean flue gas temperature properties between pipe inlet and outlet. All physical properties except density are assumed independent of pressure and its values for pure gases are given in appendix. Flue gas physical properties are calculated as molar average of gases which form flue gas. Density is calculated using ideal gas equation of state.

Evaporation occurs on shell side of evaporative section of heat exchanger. Evaporative heat transfer coefficient on shell side is assumed constant and its value is $\alpha_{ev} = 3500 \frac{W}{m^2 K}$. In superheater section of exchanger, heat transfer coefficient on shell side is also assumed constant and its value is $\alpha_{super} = 70 \frac{W}{m^2 K}$.

Evaporated steam mass flow q_m is determined by heat flow Φ_e exchanged in evaporative part "Evap" of heat exchanger "HE":

$$\Phi_e = q_m(h'' - h_4), \tag{19}$$

where h'' is saturated (dry) steam specific enthalpy, and h4 is enthalpy of subcooled liquid at condensate pump outlet. The same amount of heat flow is received from stream "b" flue gas whose enthalpy decreases.

$$\Phi_e = q_{m,fg,evap}(h_{fg,comb} - h_{fg,evap,out}), \tag{20}$$

where $h_{fg,comb}$ is flue gas specific enthalpy at combustion chamber "Comb" outlet $h_{fg,evap,out}$ is flue gas specific enthalpy at evaporative section "Evap" outlet. Flue gas mean molar heat capacities are given in appendix It is assumed that mass flow q_m in current time step flows through superheating section "Super" of "HE", turbine "Turb", condenser "Cond" and condensate pump "P3".

Evaporated steam mass flow q_m, whose value is determined by equation (13) is then superheated at constant pressure in superheater section "Super" of heat exchanger "HE" by exchanged heat flow:

$$\Phi_s = q_m(h_1 - h''), \tag{21}$$

The same amount of heat flow is received from stream "c" flue gas whose enthalpy decreases.

$$\Phi_s = q_{m,fg,super}(h_{fg,comb} - h_{fg,super,out}), \tag{22}$$

where $h_{super,out}$ is flue gas specific enthalpy at superheating section "Super" exit. Both flue gas streams are adiabatically mixed at heat exchanger outlet in mixing chamber "Mix". Outlet temperature from the mixing chamber is defined by equation:

$$\Theta_{fg,ret} = \frac{q_{m,fg,evap} \cdot h_{fg,evap,out} + q_{m,fg,super} \cdot h_{fg,super,out}}{q_{m,fg} \cdot c_{p,fg}(\Theta_{fg,ret})}, \tag{23}$$

where:

- $q_{m,fg}$ is temperature after stream "b" and "c" mixing at which part of flue gas is returned to combustion chamber

- $c_{p,fg}(\Theta_{fg,ret})$ is mean specific heat capacity of flue gas between temperature $\Theta_{fg,ret}$ and $0°C$.

Data for available tubes are given in table 2.

Value of exchanged heat at both sections depends upon stream inlet conditions, overall heat transfer coefficient k and overall heat transfer area A. Heat transfer coefficient k and area A are influenced by number of tubes in each section and its length which define tube side heat exchanger coefficient and total heat transfer area.

Type and length of tubes at evaporative and superheating section are equal. Number of tubes for each section and flue gas fraction which enters each section can be selected independently.

While designing a heat exchanger, to prevent tube erosion, one must consider that product of flue gas density, ρ, and squared velocity, w^2, never exceed $6800Pa$, i.e.

$$\rho w^2 < 6800Pa. \tag{24}$$

Condenser

Condenser "Cond" is a heat exchanger which is used for condensation of steam which exits turbine "Turb" at condensation pressure p_2. During condensation, following heat flow is released:

$$\Phi_{cond} = q_m(h_2 - h'), \tag{25}$$

where h' is enthalpy of saturated liquid water at condensation pressure p_2.

Heat flow Φ_{cond} is rejected to sea water inside condenser "Cond". It can be assumed that condenser "Cond" in the system is properly designed and sea water mass flow at pump "P4" is automatically adjusted to absorb heat flow Φ_{cond}. Therefore, sizing of condenser "Cond" and sea water pump "P4" and its control is not a part of this task and is assumed as ideal. Condensed water always exits condenser "Cond" at saturated liquid state. Maximum sea water temperature at which submarine has to operate is $30°C$, and minimal temperature difference between condensing steam and sea water for given condenser is $15°C$.

Combustion chamber

Combustion chamber "Comb" is enclosed space in which ethanol burns in liquid oxygen. It is assumed to be perfectly insulated. During the combustion, lower heating value of ethanol is converted into flue gas enthalpy. Combustion pressure is at least 1 bar above static pressure evaluated at submarine centerline. Pressure loss from combustion chamber "Comb" to heat exchanger "HE" can be neglected.

In addition to ethanol combustion, combustion chamber "Comb" is used for mixing newly formed flue gas and cooled flow gas which passed heat exchanger "HE" and is returned to chamber "Comb" to prevent overheating. Combustion chamber inlet streams are pure oxygen stream "i" and ethanol stream "h", both at $10°C$ and returning flue gas stream "d" at heat exchanger exit temperature $\Theta_{fg,ret}$. The only outlet stream from the combustion chamber "Comb" is flue gas stream "a" which consists of return flue gas "d" and flue gas newly formed by combustion.

$$q_{m,fg} = q_{m,fg,new} + q_{m,fg,ret}, \tag{26}$$

Flue gas outlet temperature is defined by first law of thermodynamics applied to combustion chamber "Comb":

$$\Theta_{fg,comb} = \frac{\dot{H}_{O2,in} + \dot{H}_{ethanol,in} + \dot{H}_{fg,ret} + q_{n,ethanol}\Delta H_{md}}{C_{fg}}, \tag{27}$$

where

- ΔH_{md} is ethanol lower heating value [kJ/kmol]

- $\dot{H}_{O2,in}$ is enthalpy of oxygen stream "i" at $10°C$

- $\dot{H}_{ethanol,in}$ is enthalpy of ethanol stream "h" at $10°C$

- $\dot{H}_{fg,ret}$ is enthalpy of returning flue gas at temperature $\Theta_{fg,ret}$

- $q_{n,ethanol}$ is molar flow rate of ethanol stream "h"

- C_{fg} is mean isobaric heat capacity of flue gas stream "a" which exits combustion chamber "Comb"

Flue gas composition is the same in all system parts and is defined by complete combustion of ethanol in pure oxygen.

2.5 Fuel and oxygen pumps

Fuel pump "P2" is used to transport fuel to combustion chamber. Pump "P2" has a variable frequency drive which allows precise control of fuel mass flow to combustion chamber. It is assumed that combustion of delivered fuel mass flow is complete and instantaneous. It is also assumed that control of oxygen pump "P1" is linked to pump "P2" in a way that oxygen supply is in exact stoichiometric ratio needed for complete ethanol combustion. Therefore, oxygen pump "P1" dimensioning and control is not a part of this assignment. Fuel pump "P2" speed determines the fuel combustion rate and influences evaporation and superheating of steam in heat exchanger "HE" and finally power P extracted at steam turbine "Turb".

Fuel pump

Fuel pump "P2" is positive displacement pump with variable speed drive. It is used to transport fuel from fuel tank "T2" to combustion chamber "Comb". The total differential head a pump must generate is a sum of static head difference and frictional head losses:

$$h_{tot} = h_{stat} + h_{fr}. \tag{28}$$

There is no height difference between fuel tank "T2" and combustion chamber "Comb". Fuel is stored at pressure of 1 atm. Frictional head loss is calculated based on fuel volume flow rate:

$$h_{fr} = k_{pipe} \cdot q_{v,fuel}^2, \tag{29}$$

where

- k_{pipe} is pipe friction coefficient $\left[\frac{mh^2}{l^2}\right]$

- $q_{v,fuel}^2$ is fuel volume flow rate

It is assumed that positive displacement pump volume flow rate depends only on pump speed and does not depend on pump head. Pump slippage is neglected. Pump volume flow rate with respect to pump speed is defined:

$$q_v = \frac{N}{N_{max}} q_{v,max}, \tag{30}$$

where

- N is pump speed

- N_{max} is maximum pump speed

- $q_{v,max}$ is maximum pump volume flow rate.

Pump head is limited to h_{max}. Pump speed is limited at given submarine depth such that pump total head does not exceed maximum head:

$$h_{stat} + k_{pipe}\left(\frac{N}{N_{max}} q_{v,max}\right)^2 \leq h_{max}. \tag{31}$$

Pump parameters are given in table **??**.

Compressor

Part of flue gas at heat exchanger exit after mixing at mixing chamber "Mix" is returned to combustion chamber "Comb" by compressor "Comp". This flue gas return should ensure that flue gas stream "a" always exits combustion chamber at temperature below $\theta_{fg,max}$. Compressor "Comp" can provide any flue gas mass flow rate between minimum and maximum flow rate defined for that compressor type. It can be assumed that sufficient mass of flue gases is always available for return at mixing region "Mix".

Compressor is controlled with respect to fuel pump speed. Dependence between pump speed and return flue gas flow rate provided by the compressor is linear.

For fully defining correlation between compressor and pump, compressor loads L_{Comp} at two relative pump speed $N_{rel,1}$ and $N_{rel,2}$ have to be provided. Compressor load can have a value between zero and one and is defined as:

$$L_{Comp} = \frac{q_{m,ret} - q_{m,ret,min}}{q_{m,ret,max} - q_{m,ret,min}}, \tag{32}$$

where

- $q_{m,ret}$ is flue gas mass flow rate which compressor "Comp" returns to combustion chamber "Comb"

- $q_{m,ret,min}$ is minimal compressor mass flow rate

- $q_{m,ret,max}$ is maximal compressor mass flow rate

Compressor load at some relative pump speed N_{rel} is than:

$$L_{Comp} = L_{Comp,1} + \frac{L_{Comp,2} - L_{Comp,1}}{N_{rel,2} - N_{rel,1}} \left(N_{rel} - N_{rel,1} \right) \tag{33}$$

and is bounded between values 0 and 1.

Condensate pump

Condensate pump "P3" is used for return of condensed steam at low condensation pressure p_3 condenser "Cond" outlet to heat exchanger "HE" at high evaporation pressure p_4. (mislim da ovo možemo pojednostaviti na sljedeći način) Neglecting a pump losses, condensate pump power is defined by:

$$P_{cp} = q_m(h_4 - h') = q_v(p_4 - p_3), \tag{34}$$

where

- h' is enthalpy of dry saturated water at condensation pressure p_3

- h_4 is actual enthalpy after water compression from dry saturated state at pressure p_3 to pressure p_4

2.6   Constants

- $\theta_{fg,max} = 800°C$

- $Re_{T_{crit}} = 3000$

- $Re_{L_{crit}} = 2300$

- $\alpha_{ev} = 3500 \frac{W}{m^2 K}$

- $\alpha_{super} = 70 \frac{W}{m^2 K}$

- $\theta_{fuel,in} = 10°C$

- $\theta_{O_2,in} = 10°C$

- $M_{ethanol} = 46 \frac{kg}{kmol}$

- $M_{CO_2} = 44 \frac{kg}{kmol}$

- $M_{H_2O} = 18 \frac{kg}{kmol}$

- $\Delta h_{md} = 1366940 \frac{kJ}{kmol}$

- $C_{mp,O_2} =$

- $C_{mp,C_2H_6O} =$

- $\rho_{sea\ water} = 1000 \frac{kg}{m^3}$

- $\eta_{cond\ pump} = 0.75$

3 Beamforming system introduction

An underwater habitat connects wirelessly to a ship on the surface by means of ultrasonic hydrophones installed at the top of the habitat. In order to increase communications range, multiple hydrophones can be used to make a beamforming system. A flat rectangular area, 45 cm x 45 cm in dimension, is provisioned for this purpose. The area lies in the y-z plane of the reference coordinate system, as depicted in Figure 2.
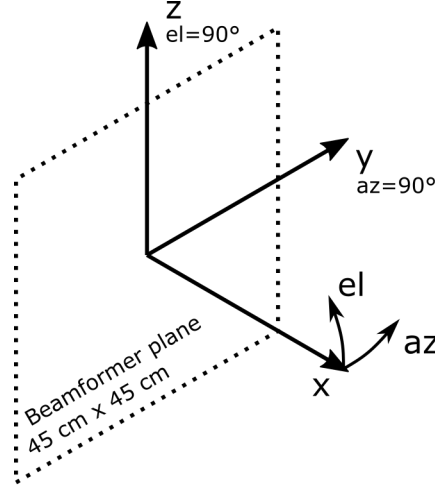

Figure 2: Reference coordinate system with available beamformer area

Hydrophone operates in frequency range of 20 kHz to 90 kHz. The communication link uses a carrier frequency of 30 kHz. Hydrophone has a cosine radiation pattern, described by expression

$$D(\phi, \theta) = \cos^2(\theta)$$

with ϕ being azimuth angle, and θ elevation angle. The radiation pattern is presented in the reference coordinate system in Figure 3.

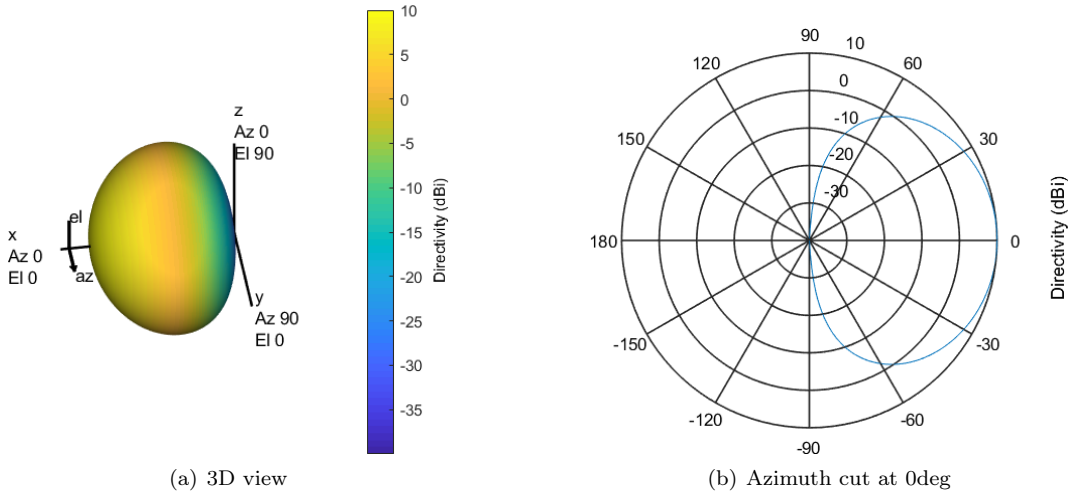(a) 3D view   (b) Azimuth cut at 0deg

Figure 3: Radiation pattern of a hydrophone element

Physically, hydrophones are circular, 5 cm in diameter. All hydrophones in the beamforming system are installed with their radiation maximum pointing to zenith, i.e. perpendicular to sea surface, as described in Figure 3. At the other end, the ship is equipped with a single hydrophone mounted on ship's hull, and pointing at nadir, i.e. perpendicular to seabed.

Consider the signal narrowband. The ship is in the farfield of the beamforming system. Assume the communication link is being established in the Adriatic Sea, with average salinity of 38‰, and sea temperature of 20 °C.

Task 1

Design a single linear antenna array along the y-axis to achieve the best link between the habitat and the ship in four given scenarios.

Scenario 1 The ship is directly above the beamforming system.

Scenario 2 The ship is located along the longitudinal (y) axis of the beamforming system, at an elevation of $30°$.

Scenario 3 The ship is located along the longitudinal (y) axis of the beamforming system, at an elevation of $60°$.

Scenario 4 The ship is located along the longitudinal (y) axis of the beamforming system, at an elevation of $70°$.

(a) Scenario 1
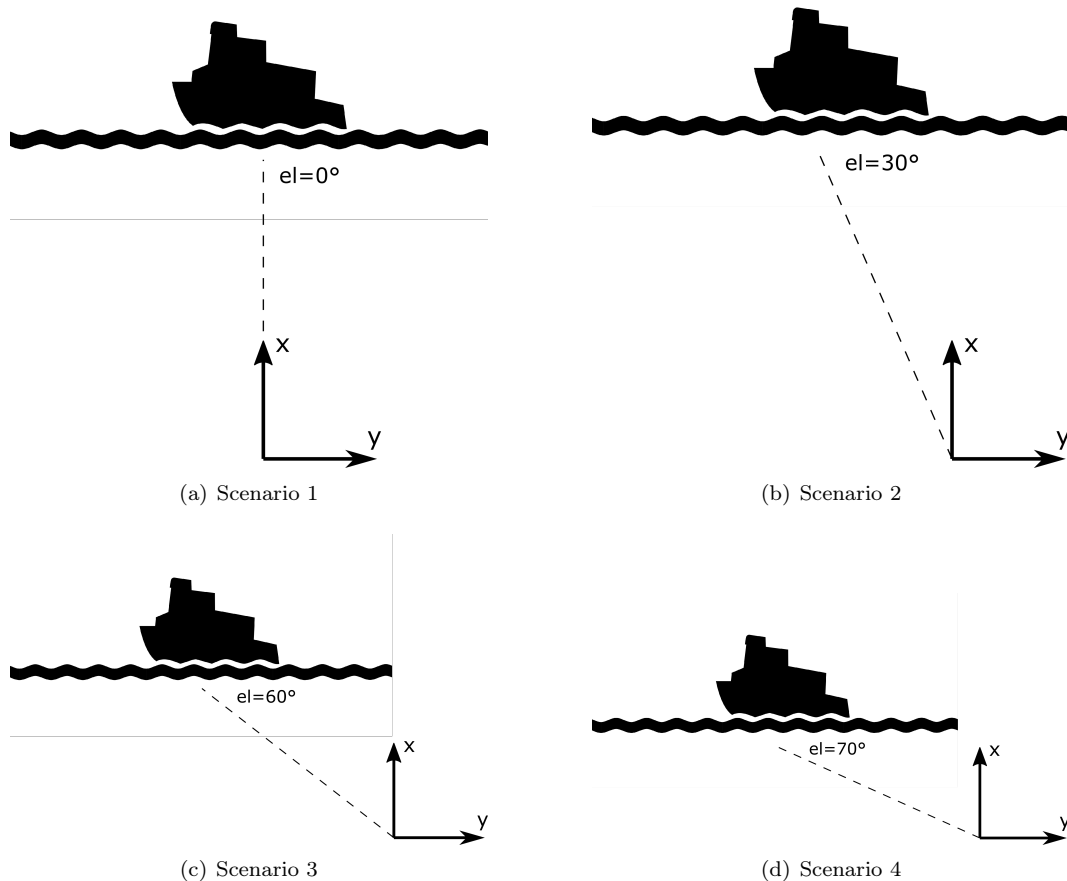
(b) Scenario 2

(c) Scenario 3

(d) Scenario 4

Figure 4: Scenarios for Task 1

A beamforming system is defined by y coordinates of each hydrophone element. Number of elements is arbitrary, given the limitations introduced in Introduction. The same beamforming system is used for all four scenarios. For each of the four scenarios, provide complex weights that are applied to each of the hydrophone elements to achieve optimal beamforming for the particular scenario.

Output data

elements1.csv

The file contains a one-dimensional array representing the position of each hydrophone element along the y axis in metres.

scenarioX.csv *e.g. scenario1.csv, . . .*

> The file contains a one-dimensional array of complex numbers representing the weights applied to each hydrophone element. Weights are given for elements in the same order as they are listed in elements1.csv. Array length is equal to the number of elements in the beamforming system. All numbers are inside, or at the edge of the unit circle.
>
> Complex numbers should be stored either in
>
> - cartesian notation: a+bi, or
> - polar notation: P*exp(Ri)

Scoring

The beamforming system you designed is evaluated for each of the given scenarios. For the first three scenarios, points are awarded for the directivity the beamforming system achieves in the direction of the ship, after applying the given beamforming weights. If the ship is in the main lobe of the beamforming system, the number of awarded points is equal to

$$D_{\text{target}} - \Delta\phi$$

where D_{target} is the directivity in the direction of the ship in dBi, and $\Delta\phi$ the angular distance between the ship and the direction of beamformer's maximum directivity in degrees. If the ship is within the strongest sidelobe of the beamforming system, the number of awarded points is equal to

$$D_{\text{target}} - \frac{\Delta\phi}{10}$$

The ship is considered to be within a radiation lobe if it is within 6 dB of the considered radiation lobe's maximum radiation.

In scenario 4, the beamforming system is evaluated with the given beamforming weights, and its directivity compared to that of the same beamforming system with uniform excitation. The points are then awarded as

$$D_{\text{target,weighted}} - D_{\text{target,uniform}}$$

The highest number of points for this scenario is 20.

For all scenarios, the following rules apply: A beamforming system constructed with just one element is awarded no points. A beamforming system acheiving negative directivity (in dBi) in the direction of the ship is awarded no points.

Task 2

You will now design a planar beamforming system, to enable beamforming toward a ship that is moving in both dimension along the sea surface. You are given a single scenario (*Scenario 5*) in which the ship is located at an azimuth angle of 45°and at elevation of 60°, as seen from the reference coordinate system.

Output data

elements2.csv

> The file contains 2D coordinates of beamforming elements. One row of the file represents one beamforming element. The first column is the y-coordinate, and the second column is the z-coordinate.

scenario5.csv

> The file contains a one-dimensional array of complex numbers representing the weights applied to each hydrophone element. Weights are given for elements in the same order as they are listed in elements2.csv. Array length is equal to the number of elements in the beamforming system. All numbers are inside, or at the edge of the unit circle.
>
> Complex numbers should be stored either in
>
> - cartesian notation: a+bi, or
> - polar notation: P*exp(Ri)

Scoring

The team will achieve points that correspond to directivity in dBi in the direction of the ship.

A beamforming system constructed with just one element is awarded no points. A beamforming system acheiving negative directivity (in dBi) in the direction of the ship is awarded no points.

Evaluation and results

Evaluations are performed automatically, and will run continuously during the competition day. You can see the results of evaluation for all teams on a scoring board, and thus monitor how everyone progresses. The scoring board can be found on `https://engineering.stemgames.hr`.

You submit your output files to your team's Google Drive folder. After the simulation for your beamformer design is complete, you will find evaluator's log file on Google Drive.

The input files found on Google Drive at the end of the competition day will be taken as team's final solution. No changes will be accepted after that time.

At the end of the competition day, and after the final evaluations complete, the total number of points awarded to the team will be normalized to the best team. Hence, the best team will receive 10 points, and the points received by other teams will be linearly scaled to the best team.