Your task is to design 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

**Power generation system description:**

Power generation system schematics are shown on figure 1.

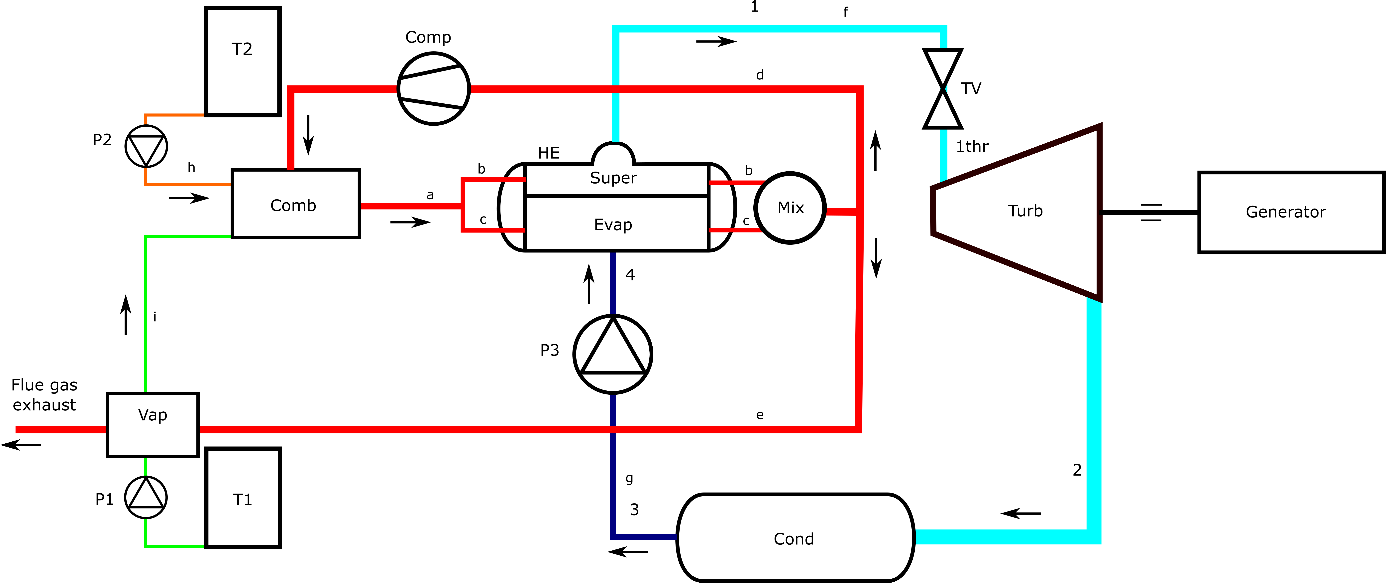


Figure 1. Power generation system

Energy for power generation is released inside combustion chamber “Comb” 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 complete without any excess oxygen. Flue gas temperature is controlled by return stream of cooled flue gas “d” by compressor “Comp” and should never exceed 800 °C.

Flue gas stream “a”, which exits combustion chamber “Comb” enters heat exchanger “HE” used for water evaporation and steam superheating. Total flue gas mass flow which enters heat exchanger is divided into two flue gas streams “b” and “c” which flow through evaporation (“Evap”) and superheating (“Super”) sections of heat exchanger “HE” in parallel configuration. Napraviti shemu izmjenjivača. On heat exchanger exit, two parallel streams “b” and “c” of flue gas are adiabatically mixed in “Mix” part of “HE” forming flue gas stream “m”. Part of stream “m” is returned by compressor “Comp” to combustion chamber “Comb” as stream “d” to prevent overheating (max 800 °C), while the rest (stream “e”) is used for oxygen evaporation in “Vap” and evacuated outside submarine.

Steam mass flow (stream “g”) which is evaporated in evaporation section “Evap” is superheated in superheating section “Super” of heat exchanger “HE”. Superheated steam (stream “f”) passes through throttling valve “TV” to steam turbine “Turb” inlet 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)

**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:

where

qm is steam mass flow rate,

h1 is steam specific enthalpy on turbine inlet,

h2 is steam specific enthalpy on turbine outlet.

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

where:

K is Stodola’s coefficient,

p1 is turbine inlet pressure

p2 is turbine outlet pressure

Tis turbine inlet absolute temperature

**Steam turbine isentropic efficiency**

Turbine isentropic efficiency is defined as the ratio between enthalpy drop at actual turbine expansion and enthalpy drop at isentropic expansion between inlet and outlet turbine pressure p1 and p2.

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

where qv is geometric mean of volume flow rates at turbine inlet (qv,in) and at outlet in case of **isentropic** expansion (qv,out,s).

Turbine efficiency is highest at its nominal design point for which pressures (p1, p2) and inlet temperature T1 are given in table 1 and nominal flow rate is defined by Stodola’s ellipse equation. Turbine can operate at other conditions, provided that these conditions are not outside permitted values with the cost of reduced efficiency.

**Off design turbine performance**

Steam throttling is a common way for reduction of power produced at steam turbine. When power demand is lowered, by reducing fuel pump “P2” and oxygen pump “P1” fuel consumption and evaporated steam mass flow rate decreases. In conjunction with steam flow rate decrease, throttling valve “TV” orifice is constricted to reduce turbine inlet pressure to a value which satisfies Stodola’s Ellipse law for given mass flow rate. In a throttling process, superheated steam which exits heat exchanger “HE” is throttled to lower pressure, while steam enthalpy is conserved.

Side effect of throttling is a temperature reduction which corresponds to the state of conserved enthalpy at throttled pressure and which is also part of a Stodola’s Ellipse law, so both pressure and temperature reduction has to be taken in account when calculating steam state which satisfies Stodola’s Ellipse law.

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

In case that power demand is increased turbine can be overloaded by steam mass flow 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. To accurately model this effect is out of scope of this assignment. Therefore, at overload working mode with increased steam flow rate, at turbine inlet values of pressure and temperature remain nominal, i.e. there is no change at throttling valve and turbine efficiency is reduced as described by equation.

Turbine parameters are given in table 1.

**Heat exchangers**

Heat exchanger is a device whose purpose is to convey 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” whose performance will not be evaluated in this task.

**Condenser and evaporator**

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). Heat exchanger “HE” It is shell and tube type with flue gas passing through tubes in both sections. Shema

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

where:

LMTD is logarithmic mean temperature difference,

k is overall heat transfer coefficient,

A is total heat exchange area.

Assume superheating section of heat exchanger is counter-flow.

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

Where:

x is evaporation to total flue gas mas flow ratio which is assumed constant throughout system operation and its value can be selected between 0 and 1.

qm,fg is total flue gas mass flow rate (stream “a”) .

is evaporation section mass flow rate (stream “b”)

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 turbulent for which critical Reynolds number equals 3000, Nusselt number is defined as:

For turbulent model, Nusselt, Prandtl and Reynolds numbers are calculated using mean flue gas temperature properties.

\* Laminar model?, combination lam i turb

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 Alfa = 3500 W/m^2K. In superheater part of exchanger, heat transfer coefficient on shell side is assumed constant and its value is Alfa = 70 W/m^2K.

Mass flow of the steam which enters the turbine is mass flow which is evaporated in evaporative section. Evaporated steam mass flow is determined by heat exchanged in evaporative part of heat exchanger:

where h’’ is saturated (dry) steam specific enthalpy, and h5 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.

where

is flue gas specific enthalpy at combustion chamber “Comb” exit

is flue gas specific enthalpy at evaporative section “Evap” exit.

Evaporated steam mass flow , whose value is determined by heat exchanged in evaporative section “Evap” is then superheated at constant pressure in superheater section “Super” of heat exchanger “HE” by exchanged heat flow:

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

where

is flue gas specific enthalpy at superheating section “Super” exit.

Both flue gas streams are adiabatically mixed at heat exchanger outlet. Exit temperature is defined by equation:

where:

is temperature after stream “b” and “c” mixing at which part of flue gas is returned to combustion chamber

is specific heat capacity of flue gas at temperature and pressure

Data for available tubes is 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 and area 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 ratio which enters each section can be chosen independently.

**Condenser**

Condenser “Cond” is a heat exchanger which is used for condensation of steam which exits turbine “Turb” at condensation pressure p2. During condensation, following heat flow is released:

where  
h’ is enthalpy of saturated liquid water at condensation pressure p2.

Heat flow 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” (Dodati morsku vodu i pumpu kroz kondenzator na shemi) is automatically adjusted to absorb heat flow . 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.

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 40 °C. Therefore, care should be taken at condensation pressure selection.

**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 occurs at 60 bars. In addition to ethanol condensation, combustion chamber is used for mixing newly formed flue gases and cooled flow gas which is returned after it passes through heat exhanger “HE”. 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 . The only outlet stream is flue gas stream “a” which consists of return flue gas “d” and flue gas newly formed by combustion. Shema

Flue gas outlet temperature is defined by first law of Thermodynamics applied to combustion chamber “Comb”:

where

is ethanol lower heating value

is enthalpy of oxygen stream “i” at 10°C

is enthalpy of ethanol stream “h” at 10°C

is enthalpy of returning flue gas at temperature

is molar flow rate of ethanol stream “h”

is 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.

**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 total ethanol combustion. Therefore, oxygen pump “P1” dimensioning and control is not a part of this assignment.

By controling fuel pump “P2” speed, rate of fuel combustion is determined which influences evaporation and superheating of steam in heat exchanger “HE” and finally power P extracted at steam turbine “Turb”.

**Fuel pump**

Opis pumpe (cijela podmornica vozi se prema dobavi pumpe) Frekvencijski upravljana, brzina vrtnje

At nominal pump speed, pump characteristic H-Q and P-Q curves are defined as:

At other pump speeds, pump flow rate, head and consumed power are linked to nominal values according to:

All pump speeds are normalized between 0 and 1 where 1 is maximum pump speed.

For pumps speed which are not nominal, H-Q and P-Q curves can be expressed as:

Pump head at given volume flow rate is limited by equation:

Additionally, pump heat cannot exceed value at any volume flow rate Q.

For desired fuel volume flow rate, pipe friction head loss is defined by:

where is pipe friction coefficient [m\*h^2/m^6].

Intersection of pump characteristic at given speed and pipe characteristic defines fuel volume flow rate. Required pump speed can be determined by solving system of equations, or graphically from pump characteristic curve charts.

\*power?, max power?, static head?

Pump parameters are given in table 3

**Compressor**

Part of flue gas at exits heat exchanger exit after mixing at “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 800 °C. 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. Connection between pump speed and return flue gas flow rate is linear. Pump speed at which compressor flow rate is minimal and at which is maximal has to be selected.

where:

is minimal fuel gas mass flow rate that compressor can provide,

is maximal fuel gas mass flow rate that compressor can provide

is pump speed at which compressor provides minimum mass flow rate

is pump speed at which compressor provides maximum mass flow rate

is pump speed

is fuel gas mass flow rate that compressor can provides at pump speed *N*

Outside selected pump speed interval, return flue gas flow rate which compressor provides is constant and equal to minimal and maximal flow rate.

If < ,

at N < , ,

at N > , .

If >

at N > , ,

at N < , .

Minimum and maximum pump speeds values can be selected outside 0 and 1 interval irrespective to actual achievable possible pump speeds. (to achieve constant mass flow between actual pump speeds if wanted).

Example: , , for all possible pump speeds *N* (between 0 and 1).

**Condensate pump**

Condensate pump “P3” is used for return of condensed steam at low condensation pressure p1 condenser “Cond” outlet to heat exchanger “HE” at high evaporation pressure p1.

Condensate pump power is defined by:

Pump isentropic efficiency is defined as:

where

is enthalpy of dry saturated water at condensation pressure p2

is enthalpy after isentropic water compression from pressure p2 at dry saturated state to pressure p1,

is actual enthalpy after water compression from pressure p2 at dry saturated state to pressure p1

Condensate pump efficiency is assumed constant 90 %.

Table 1. Turbine parameters

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Turbine nr. | Nominal power [kW] | Nominal inlet pressure [bar] | Nominal outlet pressure [bar] | Nominal inlet temperature [°C] | Stodola coefficient [kg\*K^0.5/(bar\*s)] | Efficiency coef. a [s^2m^-6] | Efficiency coef b [sm^-3] | Efficiency coef. c [-] | Max allowed temp [°C] |
| 1 | 100 | 18 | 0.6 | 250 | 0.2873 | -301,61 | 73.681 | -3.75 | 320 |
| 2 | 120 | 20 | 1 | 230 | 0,34665 | -356.4 | 86.513 | -4.5 | 330 |
| 3 | 80 | 20 | 0,4 | 260 | 0,18462 | - 497,9 | 98.541 | -4.125 | 310 |
| 4 | 40 | 17 | 0.6 | 255 | 0,11392 | -2155,7 | 204.71 | -4.05 | 300 |
|  |  |  |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |  |  |

+ Price, dimensions, mass… limits

Table 2. Heat exchanger tube

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Tube nr. | Outer diameter [mm] | Thickness [mm] | Price/m [eur] | Material | Thermal conductivity [W/mK] |
| 1 | 20 | 1 | 10 | Fe | 50 |
| 2 | 20 | 2 | 20 | Cu | 310 |
| 3 | 24 | 2 | 15 | Fe | 50 |
| 4 | 24 | 2 | 25 | Cu | 310 |
|  |  |  |  |  |  |
|  |  |  |  |  |  |
|  |  |  |  |  |  |

Table 3 Pump parameters

|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Pump nr. | a1 | b1 | c1 | a2 | b2 | c2 | d | a3 | b3 | c3 | nom speed | min  speed | max speed | max  head |
| 1 | -27.32 | 5.26 | 5.24 | -59.28 | 18.64 | 13.34 | 1.26 | 26.09 | -36.67 | 13.24 | 0.7 | 0.35 | 1 | 8.2 |
| 2 | -223.5 | 14.3 | 4.7 | -533.52 | 55.92 | 13.34 | 0.42 | 213.48 | -100.01 | 12.0 | 0.7 | 0.35 | 1 | 8 |
| 3 |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 4 |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |

Table 4. Compressor parameters

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Compressor nr. | Minimum flue gas flow rate [kg/h] | Maximum flue gas flow rate [kg/h] |  |  |
| 1 | 0.5 | 2 |  |  |
| 2 | 1 | 10 |  |  |
| 3 | 0.8 | 20 |  |  |
| 4 | 10 | 50 |  |  |
| 5 |  |  |  |  |

Zadati:

Steam properties, flue gas properties…

Hint: koristiti Coolprop, XSteam – poračunavanje svojstava pare i dimnih plinova (u excelu /matlabu / pythonu)

ethanol enthalpy of combustion: -1366,94 kJ/mol

To design submarine, it is necessary to select following parameters:

Selected turbine:

Evaporation pressure:

Condensation pressure:

Type of heat exchanger tubes (tube nr.):

Number of tubes:

Single tube length:

Evaporation flue gas ratio (qm,fg,evap/qm,fg total) :

Selected pump:

Selected compressor:

Compressor control coefficients:

Fuel pump speed during transport:

Fuel pump speed during submarine loading:

Total fuel mass

Total oxygen mass

At loading pump speed, at least xx kW of power should be generated by turbine.

Team will be graded by time required to travel given distance and money spent on system and loaded fuel.

If loaded fuel is not enough to travel full given distance, number of points will be scaled by fraction of traveled distance.

If maximum and minimum allowed pressures, temperatures or flue gas velocity limits are broken, penalty…

If flue gas condensation inside heat exchanger occurs, penalty.

Preporučeni način rješavanja:

Odabrati nominalnu snagu turbine prema karakteristici brzina podmornice – snaga. Odabrati snagu kod koje je optimalni omjer snage i brzine. Paziti na to da su tlakovi i temperature unutar dopuštenih.

Pretpostaviti da će turbina raditi u nominalnom režimu kod vožnje podmornice (najveća učinkovitost). Izračunati maseni protok pare u tom načinu rada prema Stodolinom zakonu.

Izračunati entalpiju h4 (izlaz iz pumpe kondenzata).

Izračunati potreban izmijenjeni toplinski tok na isparivaču – umnožak protoka pare i razlike entalpije suho zasićene pare i h4.

Pretpostaviti ulaznu i izlaznu temperaturu dimnih plinova u isparivač. Prema potrebnom izmijenjenom toplinskom toku izračunati potreban protok dimnih plinova kroz sekciju izmjenjivača.

Odabrati vrstu cijevi i broj cijevi tako da brzina plinova kroz cijev bude unutar dozvoljenih granica. Prema brzini strujanja i svojstvima na srednjoj temperaturi plinova izračunati koeficijent prijelaza topline na strani plinova.

Izračunati srednju logaritamsku temperaturnu razliku i koeficijent prolaza topline kroz cijevi. Izračunati potrebnu površinu za izmjenu potrebnog toplinskog toka. Prema dobivenoj površini izračunati duljinu cijevi. Prema potrebi korigirati broj cijevi i pretpostavljene temperature kako bi se dobile realne duljine cijevi.

Izračunati potreban toplinski tok koji se treba izmijeniti na superheateru prema masenom protoku pare koji množi razliku entalpije koja ulazi u turbinu i entalpije suho zasićene pare.

Duljina cijevi pregrijača jednaka je odabranoj duljini isparivača. Koristiti NTU metodu za proračun protusmjernog izmjenjivača (dodati opis metode u zadatak). Odabrati broj cijevi i maseni protok dimnih plinova kako bi pri izmjenjenom toplinskom toku izlazna temperatura dimnih plinova bila blizu one koje smo pretpostavili kod isparivača.

Proračunati temperaturu dimnih plinova nakon miješanja pri izlasku iz isparivača i pregrijača. Prema količini plinova koji je proračunat kroz isparivač i pregrijač, izračunati njihov omjer koji je konstantan pri svim uvjetima rada sustava.

Proračunati potreban omjer dimnih plinova nastalih izgaranjem i onih koji se vračaju u ložište kako bi temperatura plinova koji ulaze u isparivač i pregrijač bila jednaka pretpostavljenoj.

Proračunati količinu goriva potrebnu za izgaranje pri čemu je maseni protok koji izlazi iz ložišta jednak zbroju protoka kroz isparivač i pregrijač.

Odabrati pumpu koja može dobaviti potreban protok goriva i kompresor koji može dobaviti potrebni protok dimnih plinova koji se vraćaju u ložište.

GOTOV PRORAČUN PRI NAZIVNOJ SNAZI

Smanjiti brzinu pumpe na neku pretpostavljenu vrijednost kad turbina daje potrebnu snagu pri punjenju podmornice.

Odabrati količinu povrata dimnih plinova kako bi temperatura na izlazu iz ložišta bila u prihvatljivim granicama. Prema protoku u ovom opterećenju i u nazivnom opterećenju, odabrati linearnu regulaciju kompresora.

Podijeliti protok plinova prema kroz isparivač i pregrijač prema prethodno određenom omjeru.

Prijedlog: nakon što su dizajnirali sustav pri nazivnom opterećenju, dati im model pumpe, ložišta, isparivača, pregrijača, mješanja i povrata dimnih plinova u ložište. Kad imaju model, ovisno o odabranoj brzini vrtnje vide maseni protok pare i temperaturu pregrijanja pare.

Zatim s dobivenim protokom i stanjem pare ulaze u proračun djelomičnog opterećenja turbine. Proračunavaju tlak i temperaturu nakon prigušivanja, efikasnost turbine i snagu na turbini. Ako je snaga niža od potrebne ili previsoka, mijenjaju brzinu pumpe, modelom dobivaju novo stanje i opet proračunavaju rad turbine.