GEB

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

Co-funded by the
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Geothermal Power Plants: Course objectives

- To understand the energy conversion systems that can be used to exploit geothermal energy as a function of the characteristics of the different geothermal reservoirs.

- To use the principles of thermodynamics to analyze different aspects of the utilization of geothermal energy for electric power generation.

- To understand the basis of the use the Second Law Analysis and the Exergy Analysis to optimize the working parameters of geothermal power plants.

- To analyze the environmental impact of the use of geothermal energy for power generation.

- To introduce students to novel geothermal power production systems (EGS, hybrid systems, …)

Geothermal Power Plants: Prerequisite

Completed the following courses:

- Thermodynamics for geothermal energy
- Geology for geothermal energy
- Deep geothermal engineering

Geothermal Power Plants: Learning outcomes

- Identify appropriate system for power production from geothermal resources
- Analyze from a thermodynamic point of view different types of geothermal power plants
- Understand the specific aspects that influence the design and economics of geothermal power plants
- Select adequate geothermal power plant system depending on the characteristics of the resource
- Optimize the working parameters of simplified models of geothermal power plants using the exergy analysis.
- Evaluate ways to mitigate environmental effects, and meeting regulations
- Understand basic economic aspects of geothermal power plants

Geothermal Power Plants: Course content (syllabus)

- Geothermal power generating systems. Thermodynamics of the energy conversion processes.
- High-enthalpy geothermal resources for power generation: Dry-steam and flash steam power plants.
- Low-enthalpy geothermal resources for power generation: Binary cycle power plants
- Advanced geothermal energy conversion systems. Hybrid geothermal power systems.
- Energy and exergy analysis applied to geothermal power systems. Working parameters optimization.
- Environmental aspects for geothermal power systems
- Economic aspects for geothermal power systems
- Case studies of geothermal power systems

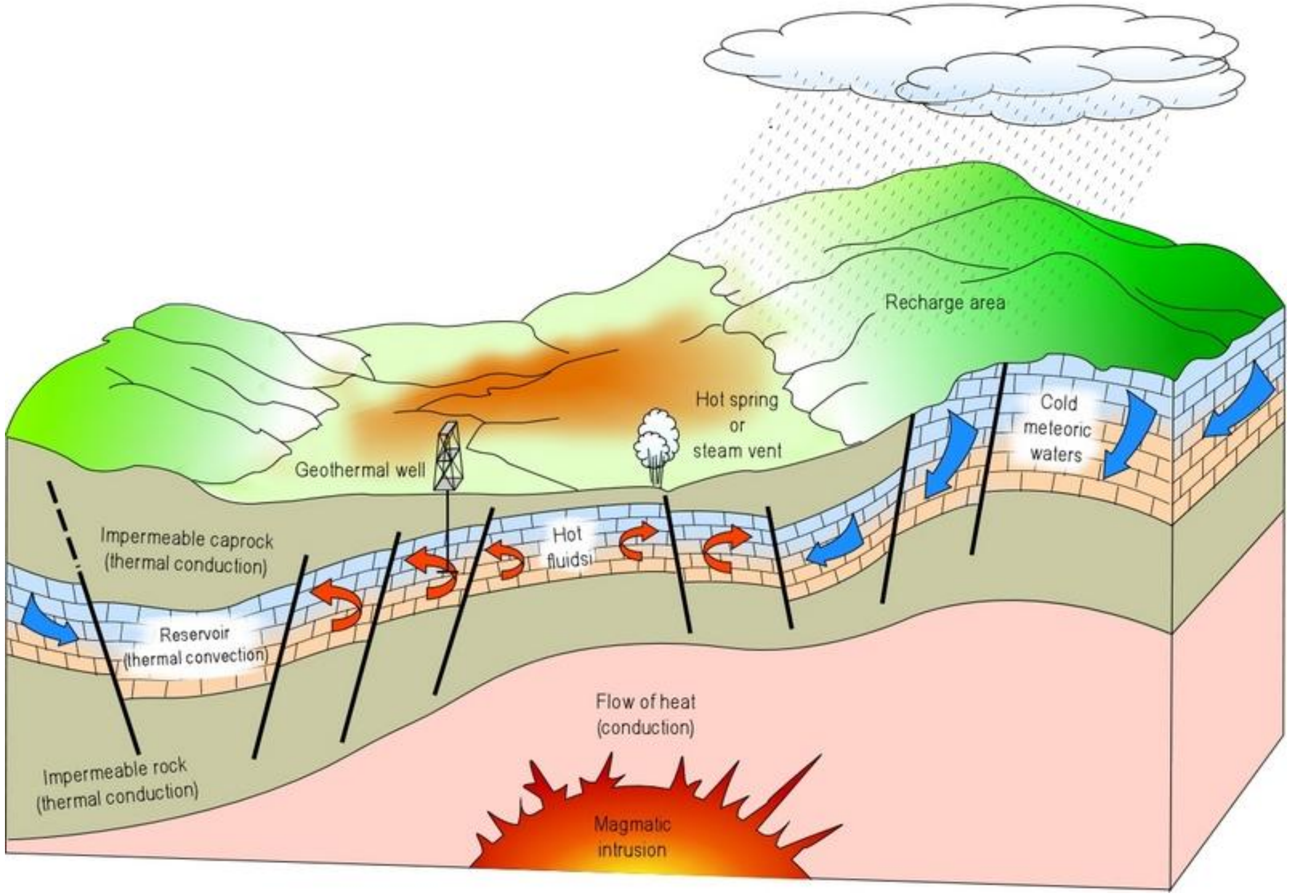
Geothermal reservoir

A geothermal reservoir is a volume of rocks in the subsurface which exploitation in terms of heat can be economically profitable.

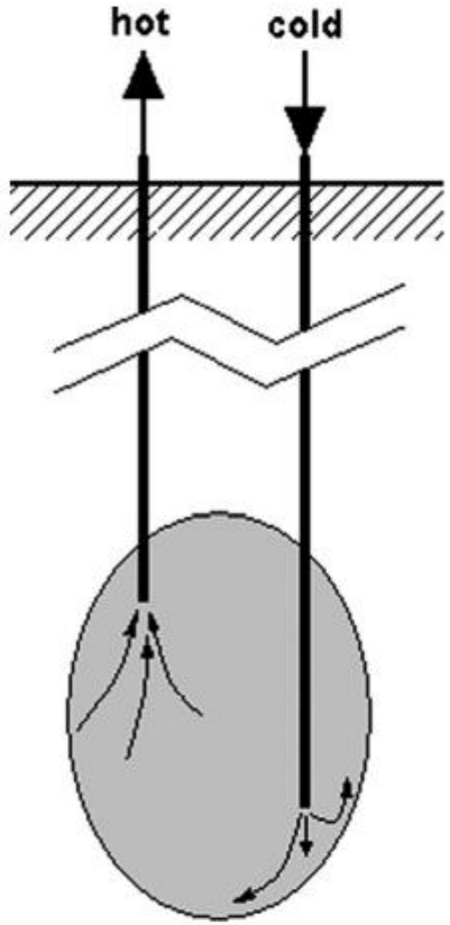
1.- High temperature at low depths

2.- Presence of water

- Permeable rocks
- Impermeable caprocks
- Natural recharge mechanisms



Geothermal ‘doublet’



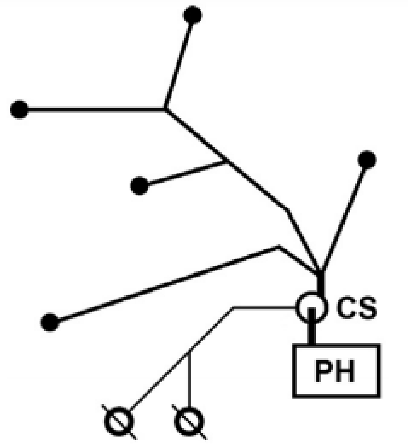


Figure 5.1 Two-phase gathering system: cyclone separator (CS) at the powerhouse (PH). Filled circles = production wells; open circles = injection wells.

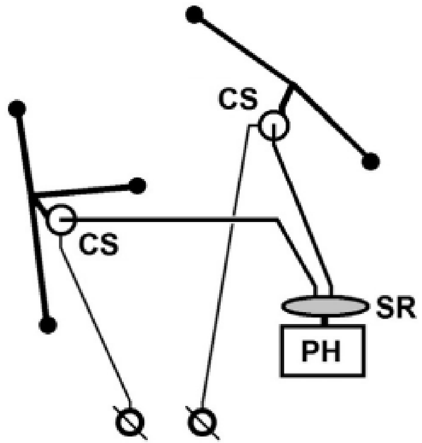
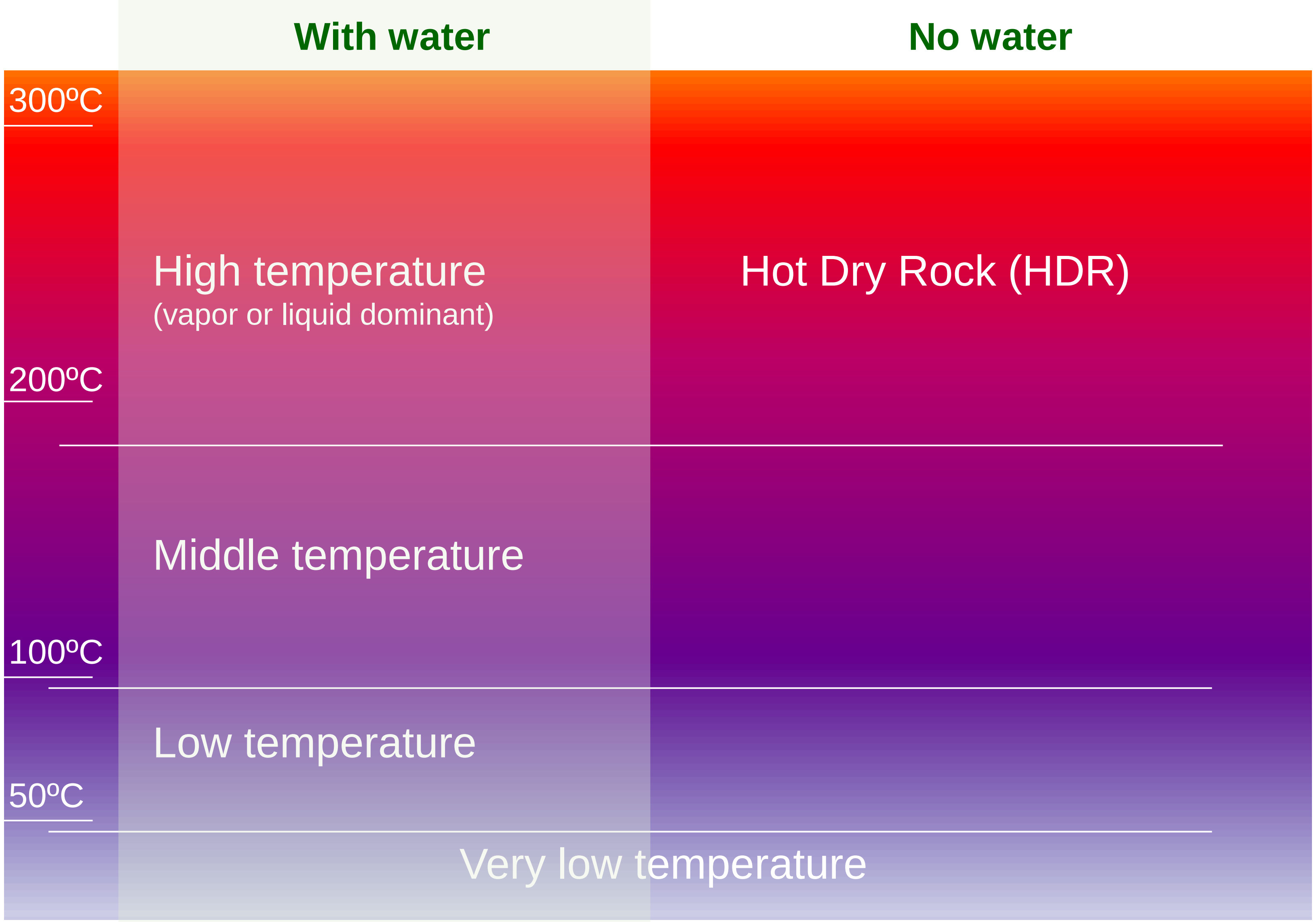


Figure 5.2 Gathering system with satellite separator stations: steam pipelines to a steam receiver (SR) at the powerhouse.

Reservoir clasification



With water

No water

300ºC

200ºC

100ºC

50ºC

High temperature
(vapor or liquid dominant)

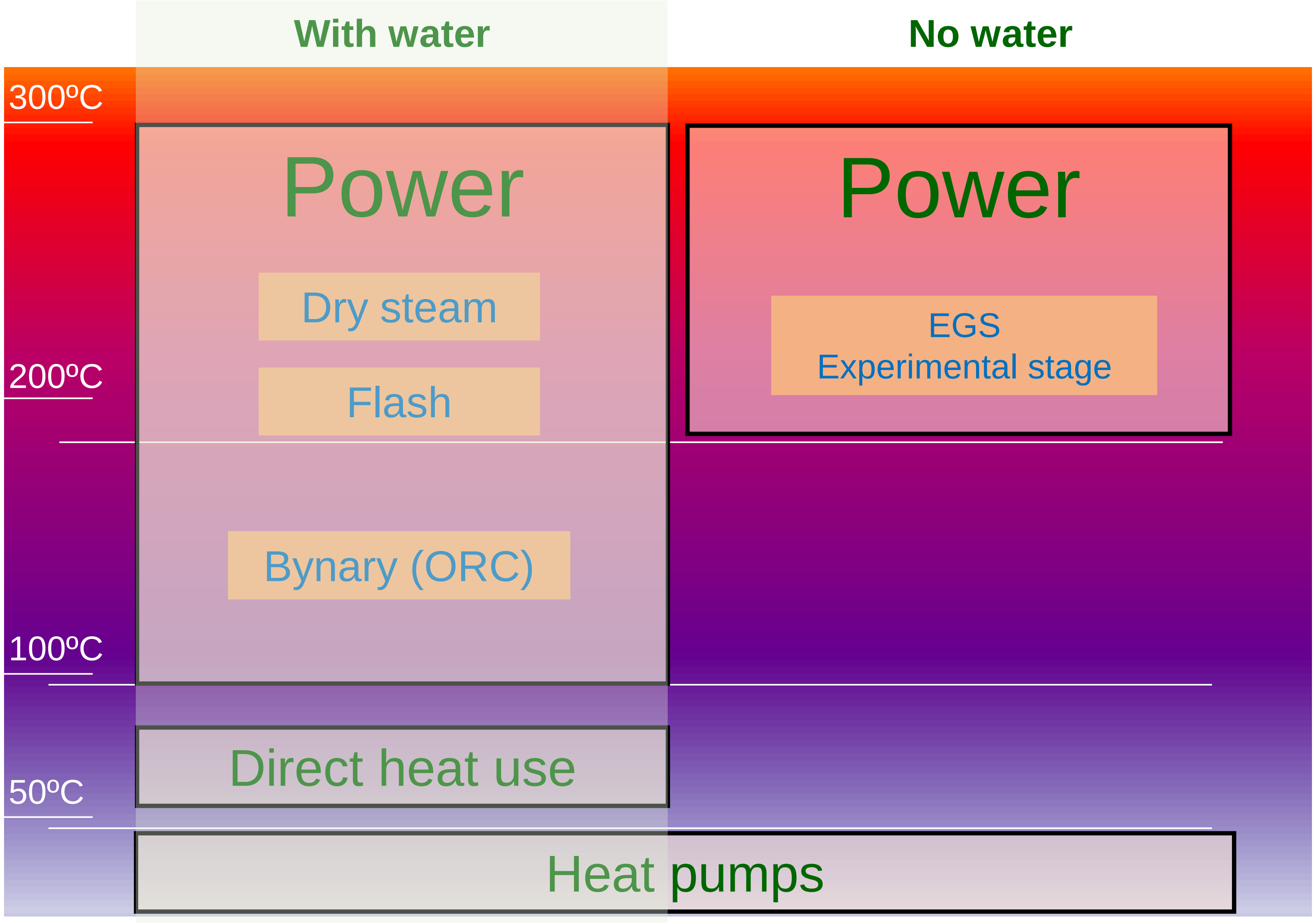
Hot Dry Rock (HDR)

Middle temperature

Low temperature

Very low temperature

Reservoir explotaition

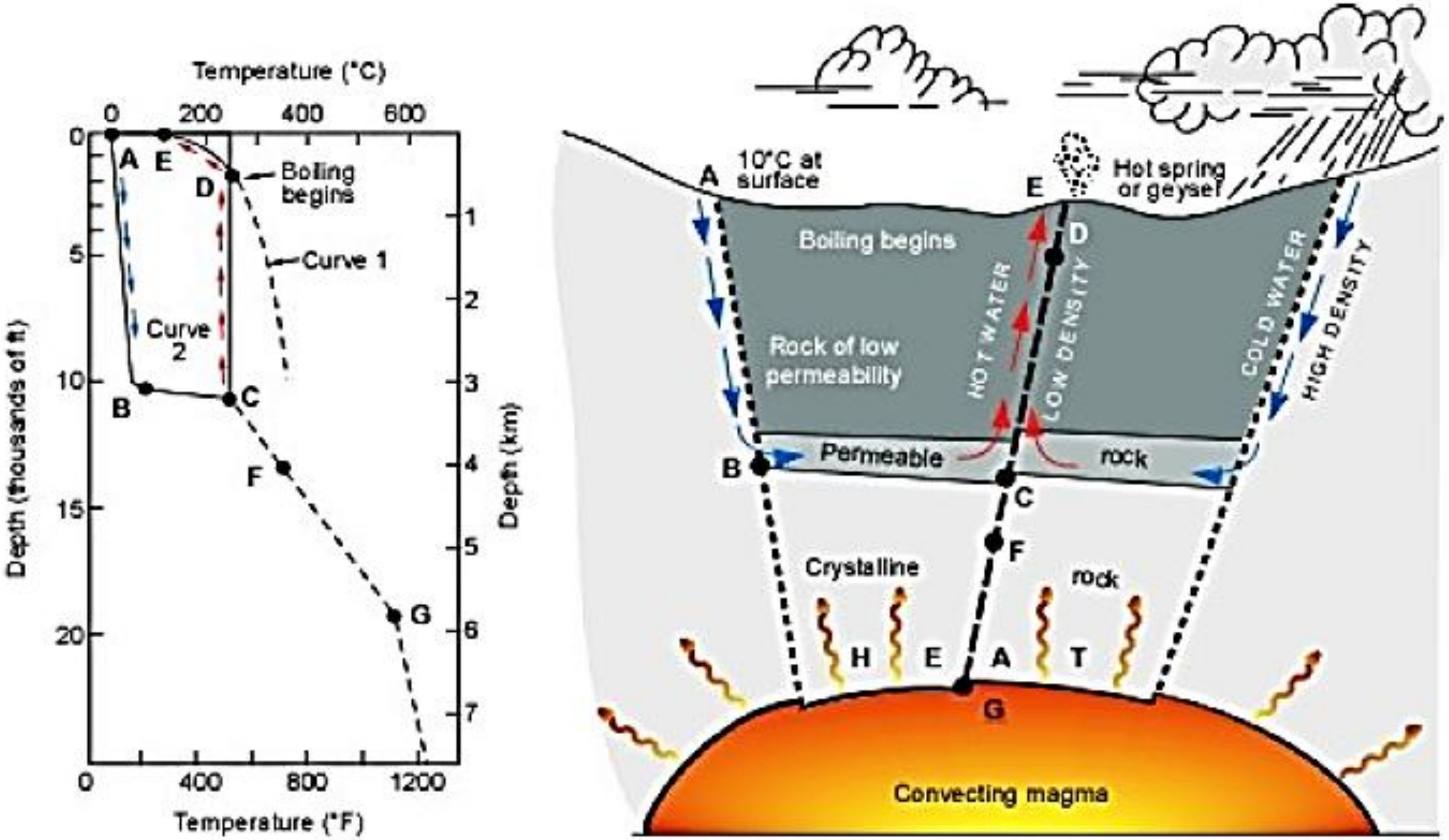


With water	No water
Power	Power
Dry steam Flash Bynary (ORC)	EGS Experimental stage

Direct heat use

Heat pumps

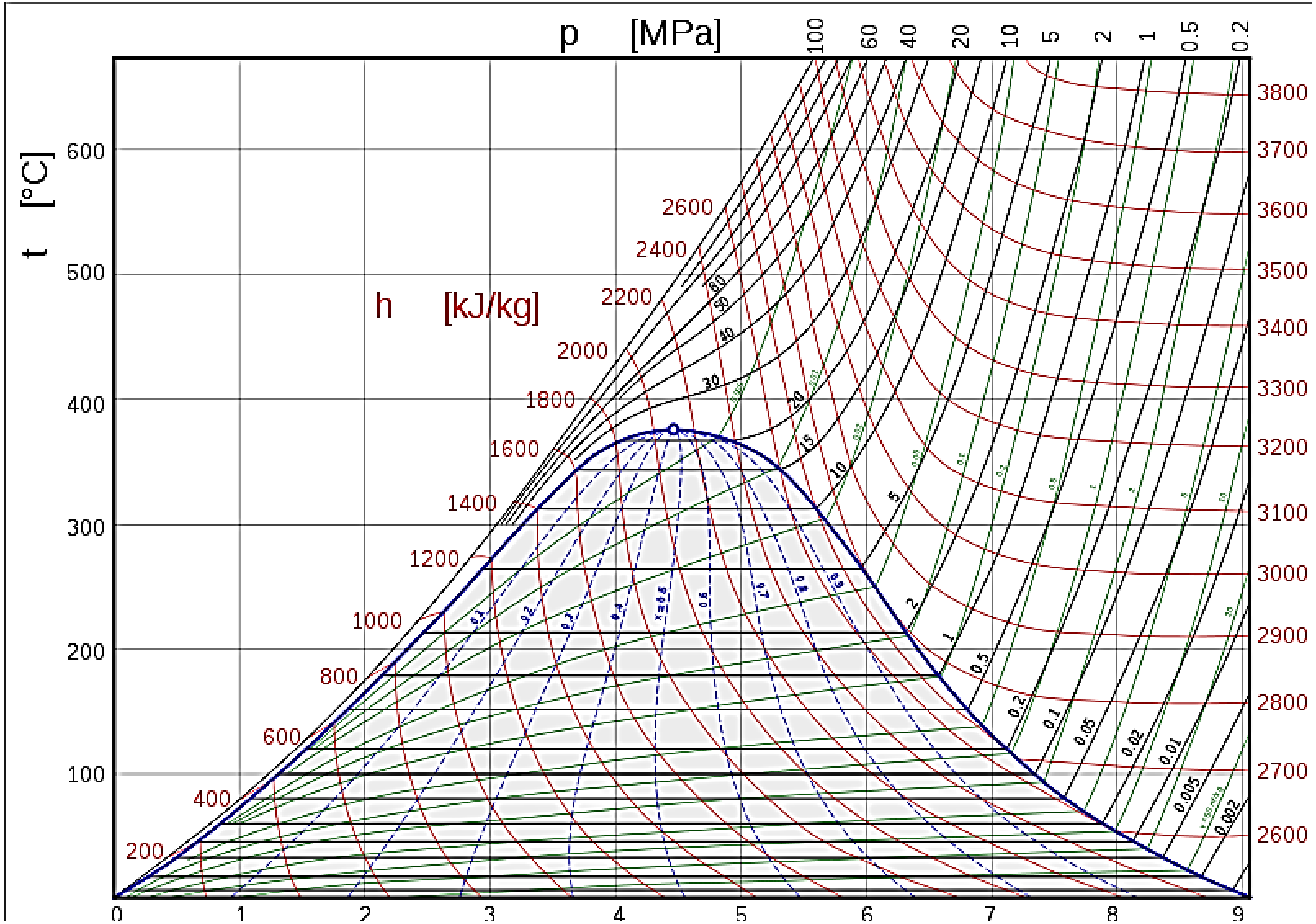
Geothermal reservoir



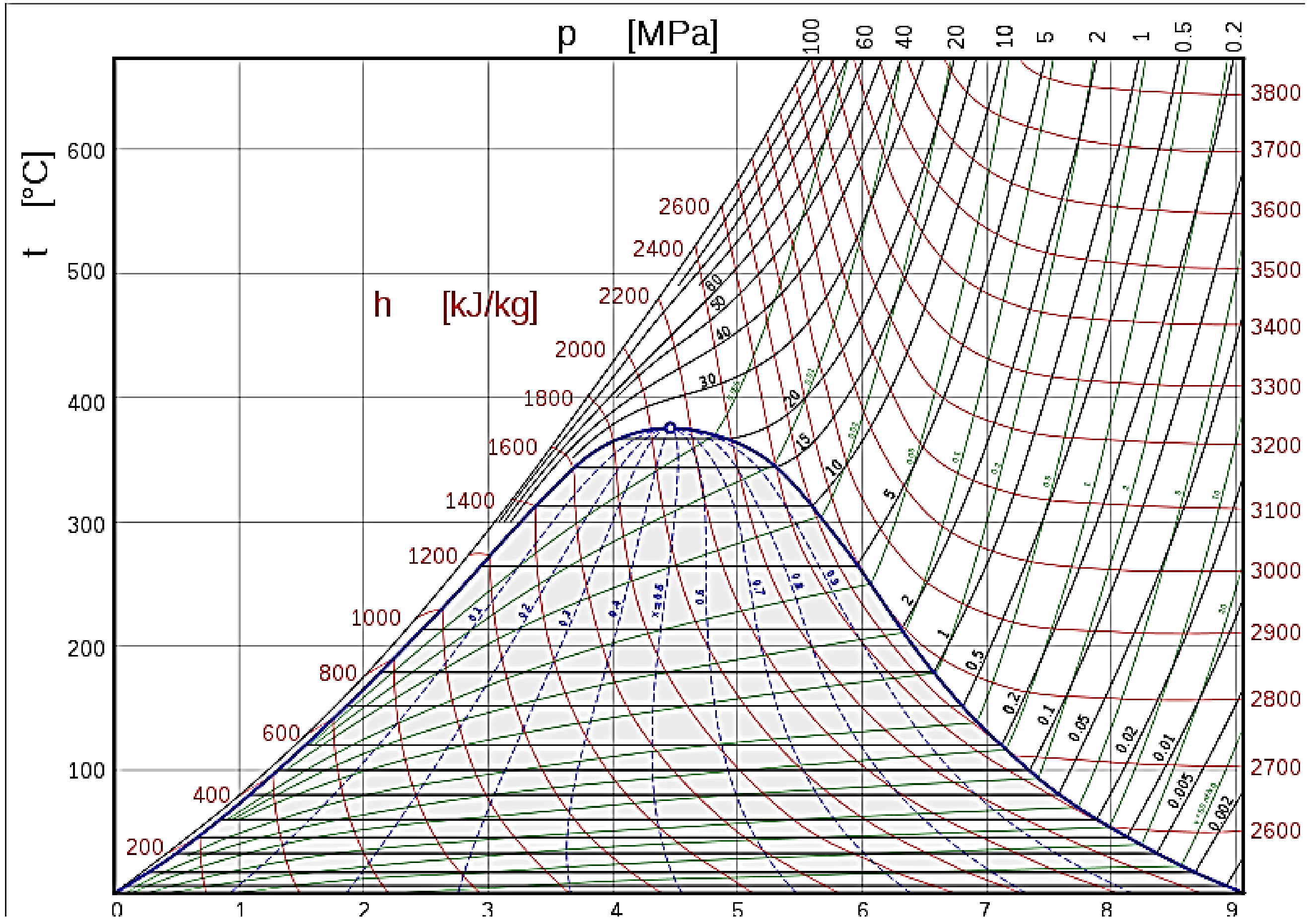
Model of a geothermal system. Curve 1 is the reference curve for the boiling point of pure water. Curve 2 shows the temperature profile along a typical circulation route from recharge at point A to discharge at point E

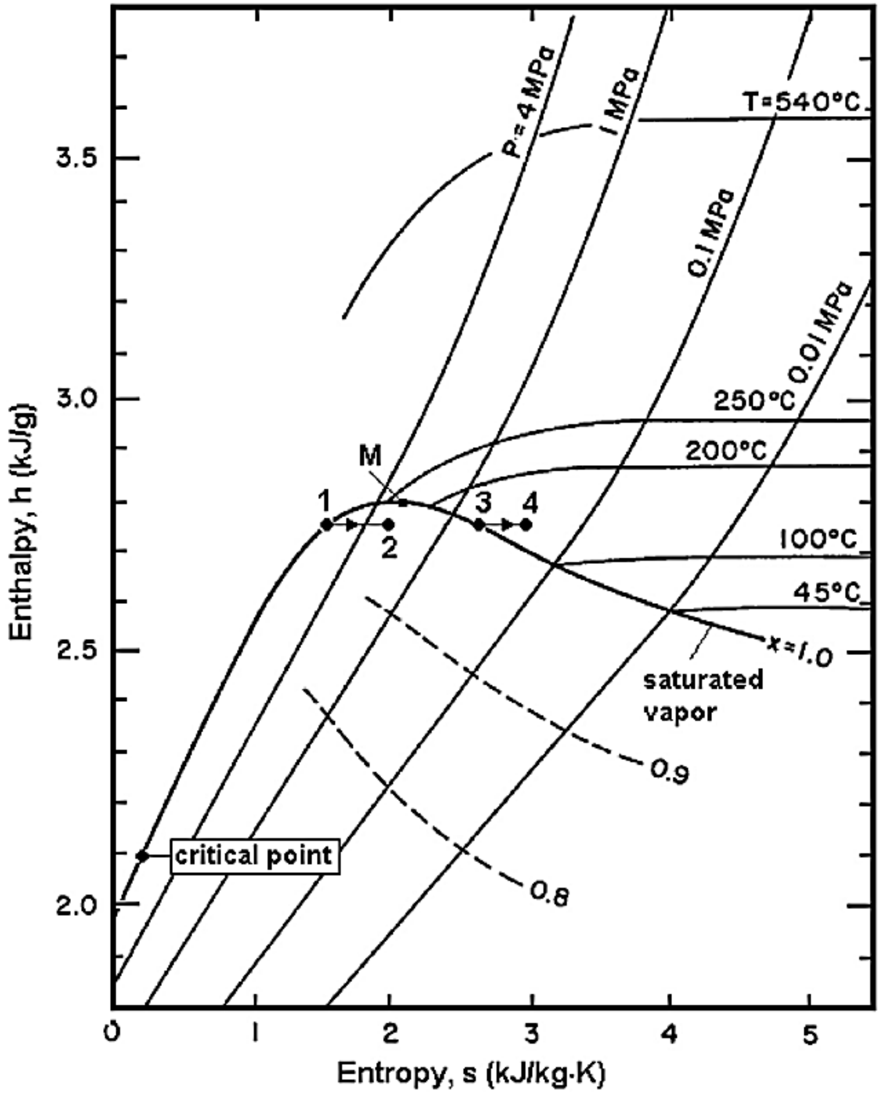
(From White, 1973).

Fluid properties. Water

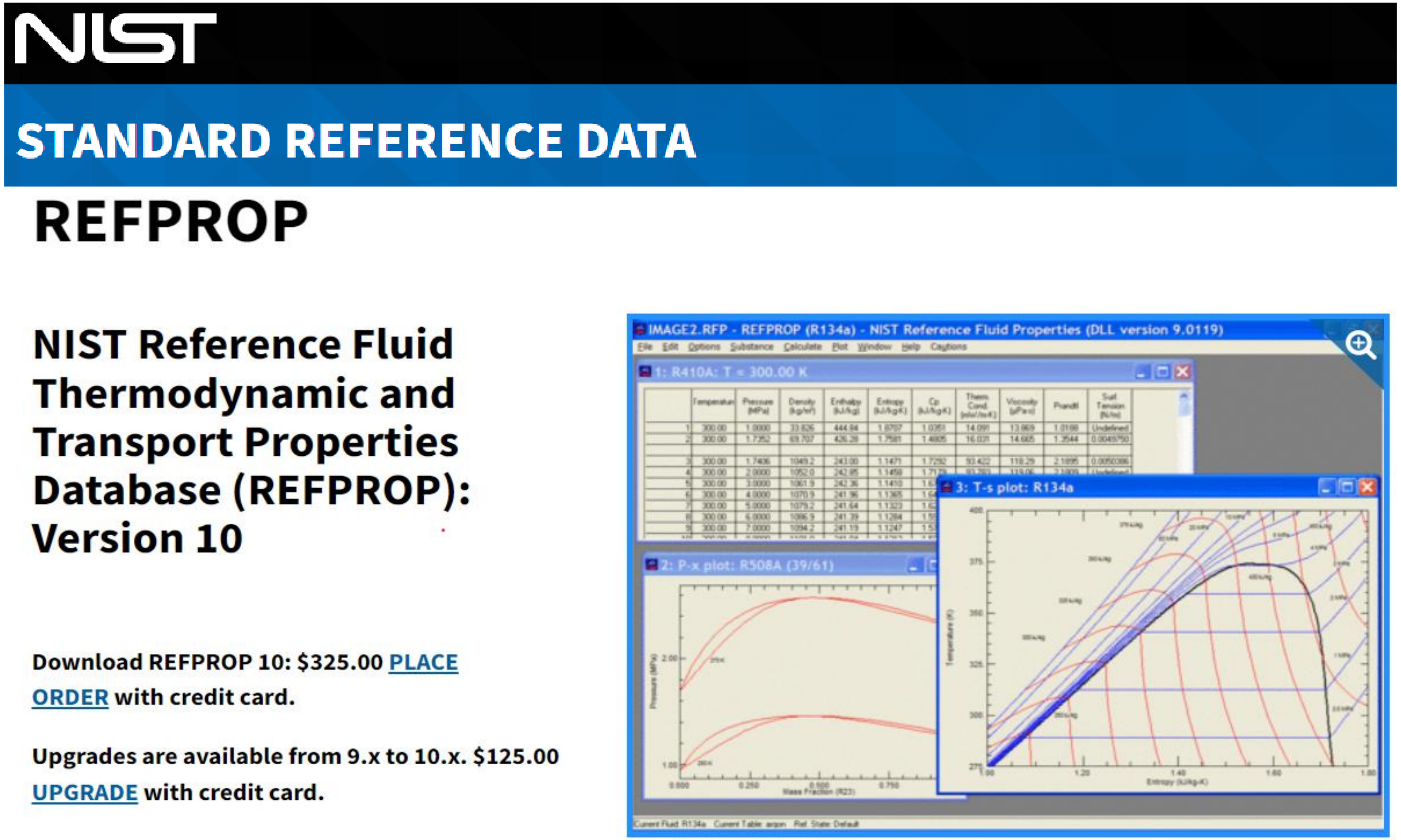


Fluid properties. Water





Fluid properties



STANDARD REFERENCE DATA

REFPROP

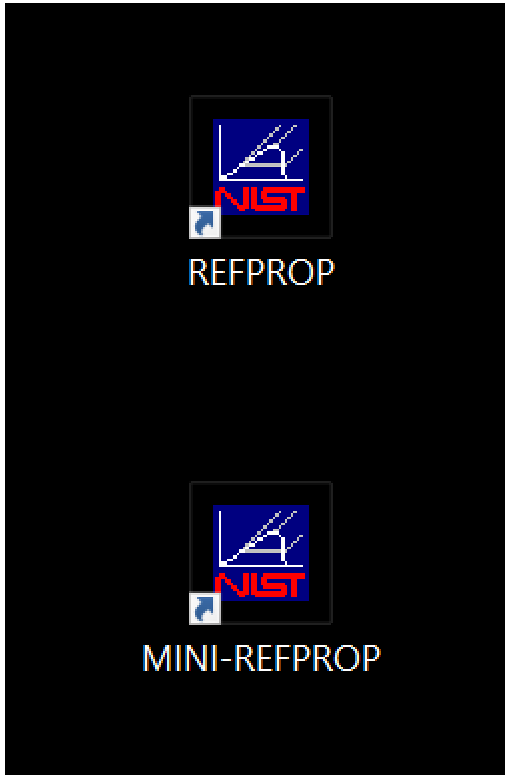
NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP): Version 10

Download REFPROP 10: $325.00 PLACE ORDER with credit card.

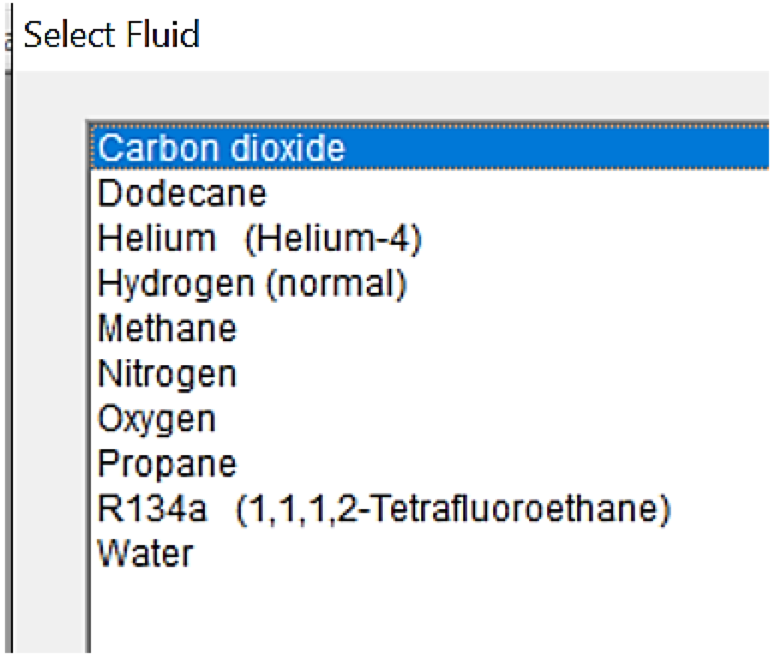
Upgrades are available from 9.x to 10.x. $125.00 UPGRADE with credit card.

https://www.nist.gov/srd/refprop

Fluid properties



- mini-REFPROP is a free sample version of the full REFPROP program ☺
- Contains only a limited number of pure fluids



Select Fluid

Carbon dioxide
Dodecane
Helium (Helium-4)
Hydrogen (normal)
Methane
Nitrogen
Oxygen
Propane
R134a (1,1,1,2-Tetrafluoroethane)
Water

- Not possible to use it linked to Excel (Add-in) ☹

https://trc.nist.gov/refprop/MINIREF/MINIREF.HTM

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Geothermal Energy Capacity Building in Egypt (GEB)

Thermodynamics: Energy analysis

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MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

- Objectives:

1. Conservation of mass principle.
2. Conservation of energy principle applied to control volumes (first law of thermodynamics).
3. Energy balance of common steady-flow devices such as nozzles, diffusers, compressors, turbines, throttling valves, mixing chambers and heat exchangers.

1. CONSERVATION OF MASS

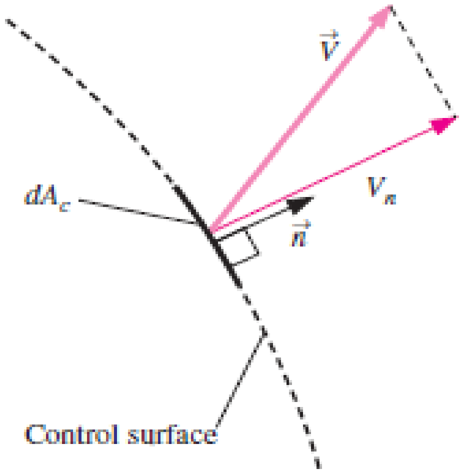
❑ Mass and Volume Flow Rates:

$$\text{Mass flow rate, } \dot{m} = \int_{A_c} \rho v_n dA_c$$

$$\boxed{\dot{m} = \rho v_n A_c}$$

$$\text{Volume flow rate, } \dot{V} = \frac{\dot{m}}{\rho} = \int_{A_c} v_n dA_c$$

$$\boxed{\dot{V} = v_n A_c}$$



The normal velocity V_n for a surface is the component of velocity perpendicular to the surface.

$\hat{n}$: normal unit vector

$\vec{V}$: Flow velocity

$\overrightarrow{V_n}$: normal flow velocity

A_c: cross − sectional area of flow

1. CONSERVATION OF MASS

❑ Conservation of Mass Principle:

The conservation of mass principle for a control volume can be expressed as: *The net mass transfer to or from a control volume during a time interval Δt is equal to the net change (increase or decrease) in the total mass within the control volume during Δt. That is,*

$$\left(\begin{array}{c}\textbf{Total mass}\\ \textbf{entering the CV during } \Delta t\end{array}\right) - \left(\begin{array}{c}\textbf{Total mass}\\ \textbf{leaving the CV during } \Delta t\end{array}\right) = \left(\begin{array}{c}\textbf{Net change in}\\ \textbf{mass within CV during } \Delta t\end{array}\right)$$

$$\boxed{\sum m_{in}\Big|_{CS} - \sum m_{out}\Big|_{CS} = \Delta m_{CV}}$$

In a rate form:

$$\boxed{\sum \dot{m}_{in}\Big|_{CS} - \sum \dot{m}_{out}\Big|_{CS} = \frac{dm_{CV}}{dt}}$$

1. CONSERVATION OF MASS

$$\frac{dm_{CV}}{dt} = \frac{d(\rho V_{CV})}{dt} = \frac{d}{dt}\int_{CV} (\rho dV + V d\rho)$$

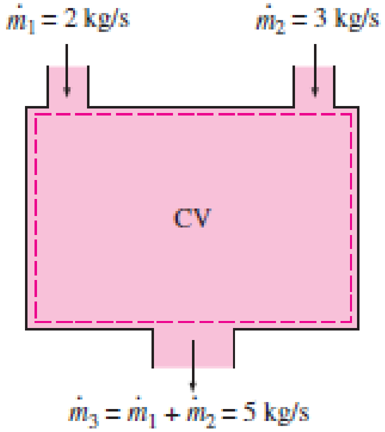
- ○ The rate of change of the mass within the control volume (CV) is due to the change of its volume dV and the change of the density of the fluid $d\rho$.

- ○ $\frac{dm_{CV}}{dt} = 0$ if there is no change in volume dV and no change in density $d\rho$.

1. CONSERVATION OF MASS

❑ Mass Balance for Steady-Flow Processes:

During a steady-flow process, the total amount of mass contained within a control volume does not change with time (m_{CV} = constant).

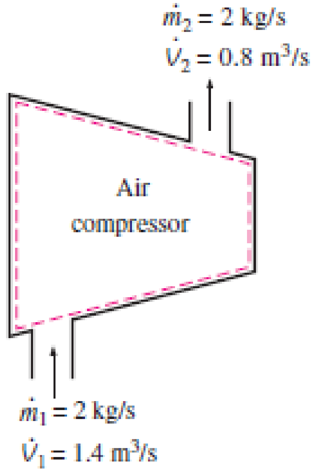


Conservation of mass principle for a two-inlet–one-outlet steady-flow system.

$$\sum \dot{m}_{in}\Big|_{CS} = \sum \dot{m}_{out}\Big|_{CS}$$

- ○ For steady-incompressible flow, i.e $\boldsymbol{\rho = \text{constant}}$:

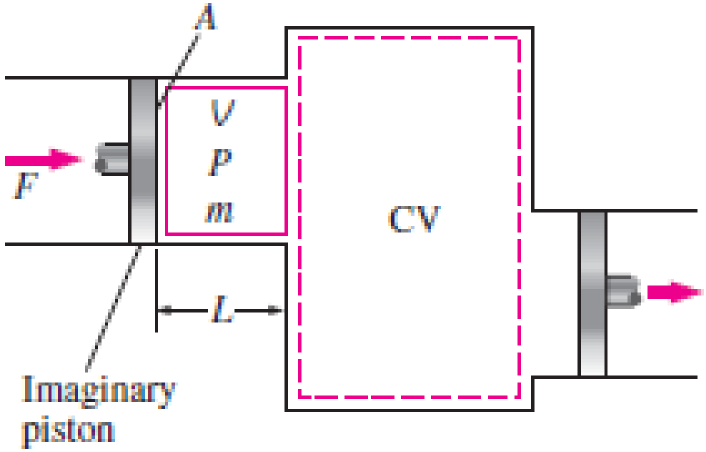
$$\sum \dot{V}_{in}\Big|_{CS} = \sum \dot{V}_{out}\Big|_{CS}$$



During a steady-flow process, volume flow rates are not necessarily conserved although mass flow rates are.

2. FLOW WORK AND THE ENERGY OF A FLOWING FLUID

- Unlike closed systems, control volumes involve mass flow across their boundaries, and some work is required to push the mass into or out of the control volume. This work is known as the *flow work*, or *flow energy*, and is necessary to maintain a continuous flow through a control volume.



Schematic for flow work.

$$\mathbf{W_{flow} = F.L = p.A.L = pV\ (J)}$$

$$\mathbf{w_{flow} = p}v\ \mathbf{(J/kg)}$$

2. FLOW WORK AND THE ENERGY OF A FLOWING FLUID

❑ For closed system:

$$\mathbf{e = u + ke + pe}$$

❑ For open system (control volume):

The energy contained in a flowing fluid is $\boldsymbol{\theta}$

$$\boldsymbol{\theta} = \mathbf{e} + \underbrace{\mathbf{p}\boldsymbol{v}}_{\text{flow work}} = \mathbf{p}\boldsymbol{v} + \mathbf{u} + \mathbf{ke} + \mathbf{pe}$$

$$\boldsymbol{\theta} = \mathbf{h + ke + pe}$$

❑ Energy Transport by Mass:

- ○ Amount of energy transport: $\mathbf{E_{mass}} = \mathbf{m}\boldsymbol{\theta} = \mathbf{m(h + ke + pe)}$
- ○ Rate of energy transport: $\dot{\mathbf{E}}_{\mathbf{mass}} = \dot{\mathbf{m}}\boldsymbol{\theta} = \dot{\mathbf{m}}\mathbf{(h + ke + pe)}$

3. ENERGY ANALYSIS OF STEADY-FLOW SYSTEMS

❑ First law of thermodynamics for open-steady flow systems:

$$\boxed{\dot{\mathbf{E}}_{\mathbf{in}} - \dot{\mathbf{E}}_{\mathbf{out}} = \frac{\mathbf{dE_{system}}}{\mathbf{dt}}}$$

Zero, for steady-state steady-flow process

$$\dot{\mathbf{E}}_{\mathbf{in}} = \dot{\mathbf{E}}_{\mathbf{out}}$$

$$\dot{\mathbf{Q}}_{\mathbf{in}} + \dot{\mathbf{W}}_{\mathbf{in}} + \dot{\mathbf{E}}_{\mathbf{mass,in}} = \dot{\mathbf{Q}}_{\mathbf{out}} + \dot{\mathbf{W}}_{\mathbf{out}} + \dot{\mathbf{E}}_{\mathbf{mass,out}}$$

$$\dot{\mathbf{Q}}_{\mathbf{in}} + \dot{\mathbf{W}}_{\mathbf{in}} + \sum_{\mathbf{in}}^{\mathbf{CS}} \dot{\mathbf{m}}\boldsymbol{\theta} = \dot{\mathbf{Q}}_{\mathbf{out}} + \dot{\mathbf{W}}_{\mathbf{out}} + \sum_{\mathbf{out}}^{\mathbf{CS}} \dot{\mathbf{m}}\boldsymbol{\theta}$$

$$\boxed{\dot{\mathbf{Q}}_{\mathbf{in}} + \dot{\mathbf{W}}_{\mathbf{in}} + \sum_{\mathbf{in}}^{\mathbf{CS}} \dot{\mathbf{m}}\left(\mathbf{h} + \frac{\mathbf{v}^{2}}{\mathbf{2}} + \mathbf{gz}\right) = \dot{\mathbf{Q}}_{\mathbf{out}} + \dot{\mathbf{W}}_{\mathbf{out}} + \sum_{\mathbf{out}}^{\mathbf{CS}} \dot{\mathbf{m}}\left(\mathbf{h} + \frac{\mathbf{v}^{2}}{\mathbf{2}} + \mathbf{gz}\right)}$$

3. ENERGY ANALYSIS OF STEADY-FLOW SYSTEMS

❑ Special cases:

1. Single stream ($\dot{m}_{in} = \dot{m}_{out} = \dot{m}$:

$$\dot{Q}_{in} + \dot{W}_{in} + \dot{m}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \dot{W}_{out} + \dot{m}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

2. Single stream per unit $\dot{m}$ (single stream per unit mass per unit time):

$$q_{in} + w_{in} + h_{in} + \frac{v_{in}^2}{2} + gz_{in} = q_{out} + w_{out} + h_{out} + \frac{v_{out}^2}{2} + gz_{out}$$

3. Single stream per unit $\dot{m}$ with negligible kinetic and potential energies:

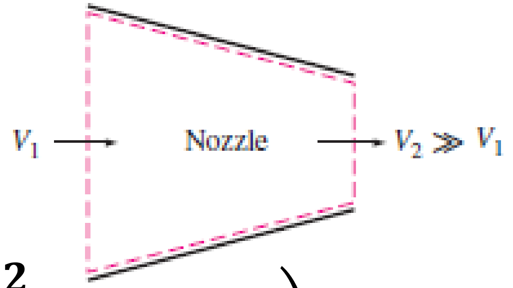
$$q_{in} + w_{in} + h_{in} = q_{out} + w_{out} + h_{out}$$

or

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{out} - h_{in}$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

➢ Nozzle:



- *Nozzle* is a device that increases the velocity of a fluid at the expense of its pressure.
- *Nozzle* can be used with compressible or incompressible fluid flow.
- Energy balance for single stream:

$$\dot{Q}_{in} + \underset{\text{Zero}}{\dot{W}_{in}} + \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \underset{\text{Zero}}{\dot{W}_{out}} + \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g(z_2 - z_1)\right]$$

For single stream and neglected change in potential energy:

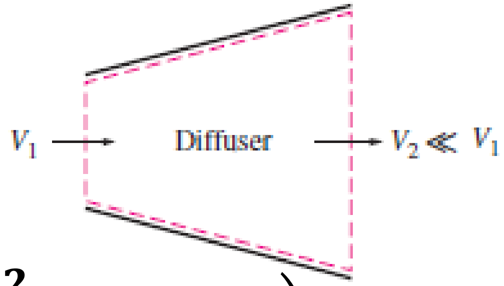
$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic nozzle:

$$(h_2 - h_1) = \left(\frac{v_1^2 - v_2^2}{2}\right)$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

Diffuser:



- *Diffuser* is a device that increases the pressure of a fluid by slowing it down.
- *Diffuser* can be used with compressible or incompressible fluid flow.
- Energy balance for single stream:

$$\dot{\mathbf{Q}}_{\mathbf{in}} + \underset{\text{Zero}}{\cancel{\dot{\mathbf{W}}_{\mathbf{in}}}} + \dot{\mathbf{m}}_{\mathbf{in}}\left(\mathbf{h}_{\mathbf{in}} + \frac{\mathbf{v}_{\mathbf{in}}^{2}}{2} + \mathbf{g}\mathbf{z}_{\mathbf{in}}\right) = \dot{\mathbf{Q}}_{\mathbf{out}} + \underset{\text{Zero}}{\cancel{\dot{\mathbf{W}}_{\mathbf{out}}}} + \dot{\mathbf{m}}_{\mathbf{out}}\left(\mathbf{h}_{\mathbf{out}} + \frac{\mathbf{v}_{\mathbf{out}}^{2}}{2} + \mathbf{g}\mathbf{z}_{\mathbf{out}}\right)$$

$$\left(\dot{\mathbf{Q}}_{\mathbf{in}} - \dot{\mathbf{Q}}_{\mathbf{out}}\right) = \dot{\mathbf{m}}\left[\left(\mathbf{h}_{2} - \mathbf{h}_{1}\right) + \left(\frac{\mathbf{v}_{2}^{2} - \mathbf{v}_{1}^{2}}{2}\right) + \mathbf{g}\left(\mathbf{z}_{2} - \mathbf{z}_{1}\right)\right]$$

For single stream and neglected change in potential energy:

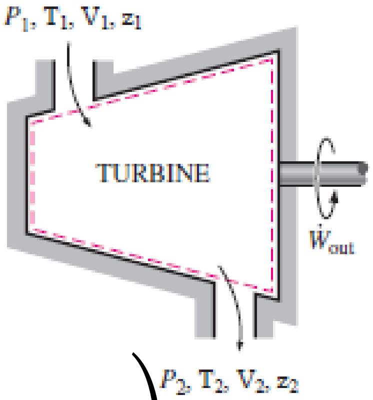
$$\left(\dot{\mathbf{Q}}_{\mathbf{in}} - \dot{\mathbf{Q}}_{\mathbf{out}}\right) = \dot{\mathbf{m}}\left[\left(\mathbf{h}_{2} - \mathbf{h}_{1}\right) + \left(\frac{\mathbf{v}_{2}^{2} - \mathbf{v}_{1}^{2}}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic diffuser:

$$\left(\mathbf{h}_{2} - \mathbf{h}_{1}\right) = \left(\frac{\mathbf{v}_{1}^{2} - \mathbf{v}_{2}^{2}}{2}\right)$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

Turbine:



- *Turbine* is a device that produces power from the fluid.
- *Gas turbine, steam turbine and wind turbine* use a compressible fluid flow.
- *Water turbine* uses an incompressible fluid flow.
- Energy balance for single stream fluid flow:

$$\dot{Q}_{in} + \underset{\text{Zero}}{\dot{W}_{in}} + \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \dot{W}_{out} + \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) - \dot{W}_{out} = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g(z_2 - z_1)\right]$$

For single stream and neglected change in potential energy:

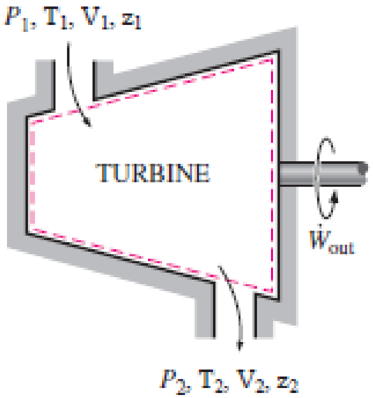
$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) - \dot{W}_{out} = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic turbine:

$$\dot{W}_{out} = \dot{m}\left[(h_1 - h_2) + \left(\frac{v_1^2 - v_2^2}{2}\right)\right]$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

Turbine:



- Energy balance for single stream-*incompressible* fluid flow:

$$\Delta h = \frac{\Delta p}{\rho}$$

$$(\dot{Q}_{in} - \dot{Q}_{out}) - \dot{W}_{out} = \dot{m}\left[\left(\frac{p_2 - p_1}{\rho}\right) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g(z_2 - z_1)\right]$$

For single stream and neglected change in potential energy:

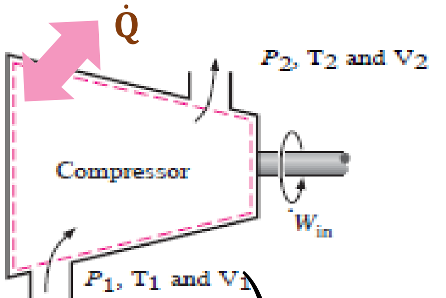
$$(\dot{Q}_{in} - \dot{Q}_{out}) - \dot{W}_{out} = \dot{m}\left[\left(\frac{p_2 - p_1}{\rho}\right) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic turbine:

$$\dot{W}_{out} = \dot{m}\left[\left(\frac{p_1 - p_2}{\rho}\right) + \left(\frac{v_1^2 - v_2^2}{2}\right)\right]$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

➢ Compressor:



- *Compressor* is a device that delivers power to a compressible fluid.
- Energy balance for single stream-*compressible* fluid flow:

$$\dot{Q}_{in} + \dot{W}_{in} + \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \underset{\text{Zero}}{\dot{W}_{out}} + \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) + \dot{W}_{in} = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g(z_2 - z_1)\right]$$

For single stream and neglected change in potential energy:

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) + \dot{W}_{in} = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic compressor:

$$\dot{W}_{in} = \dot{m}\left[(h_2 - h_1) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

➢ Pump:

- *Pump* is a device that delivers power to an incompressible fluid.
- Energy balance for single stream-*incompressible* fluid flow:

$$\dot{Q}_{in} + \dot{W}_{in} + \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \underset{\text{Zero}}{\cancel{\dot{W}_{out}}} + \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) + \dot{W}_{in} = \dot{m}\left[\left(\frac{p_2 - p_1}{\rho}\right) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g(z_2 - z_1)\right]$$

For single stream and neglected change in potential energy:

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) + \dot{W}_{in} = \dot{m}\left[\left(\frac{p_2 - p_1}{\rho}\right) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic pump:

$$\dot{W}_{in} = \dot{m}\left[\left(\frac{p_2 - p_1}{\rho}\right) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

➢ Throttling valve:

- *Throttling valves* are any kind of flow-restricting devices that cause a significant pressure drop in the fluid.
- *Throttling valves* can be used with compressible or incompressible fluid flow.
- Energy balance for single stream fluid flow:

$$\dot{Q}_{in} + \underset{\text{Zero}}{\dot{W}_{in}} + \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \underset{\text{Zero}}{\dot{W}_{out}} + \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

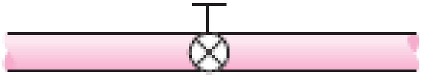
$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}\left[\left(h_2 - h_1\right) + \left(\frac{v_2^2 - v_1^2}{2}\right) + g\left(z_2 - z_1\right)\right]$$

For single stream and neglected change in potential energy:

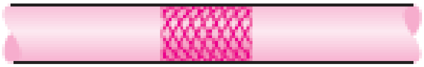
$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}\left[\left(h_2 - h_1\right) + \left(\frac{v_2^2 - v_1^2}{2}\right)\right]$$

For single stream, neglected change in potential energy and adiabatic:

$$\left(h_2 - h_1\right) = \left(\frac{v_1^2 - v_2^2}{2}\right)$$



(*a*) An adjustable valve



(*b*) A porous plug

(*c*) A capillary tube

4. SOME STEADY-FLOW ENGINEERING DEVICES

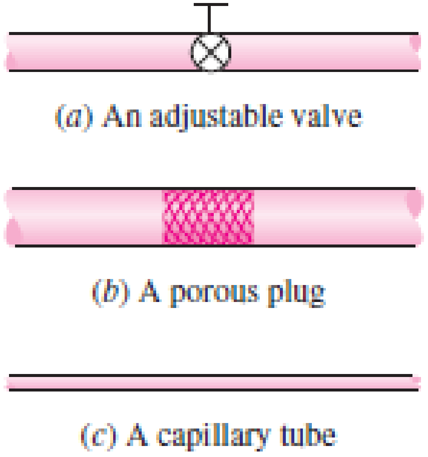
➢ Throttling valve:

For single stream, neglected change in potential energy, adiabatic and constant velocity:

$$h_2 = h_1$$

For single stream, neglected change in potential energy, adiabatic, constant velocity and ideal gas:

$$T_2 = T_1$$



4. SOME STEADY-FLOW ENGINEERING DEVICES

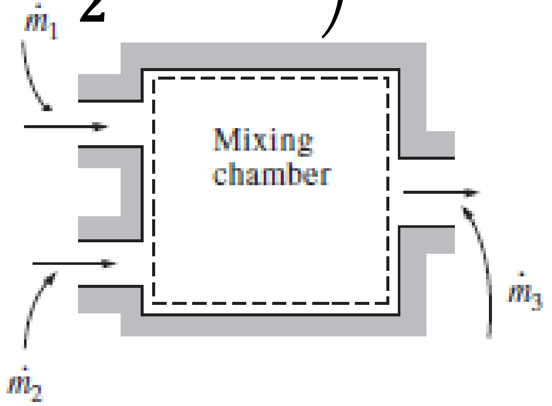
Mixing chamber:

- *Mixing chamber* is a section where the mixing process takes place.
- Energy balance:

$$\dot{Q}_{in} + \dot{W}_{in} + \sum \dot{m}_{in}\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) = \dot{Q}_{out} + \dot{W}_{out} + \sum \dot{m}_{out}\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)$$

- Mass balance:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}$$



For instance, if the mixing chamber has two inlets and one outlet:

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

$$(\dot{Q}_{in} - \dot{Q}_{out}) + (\dot{W}_{in} - \dot{W}_{out}) = \dot{m}_3\left[h_3 + \left(\frac{v_3^2}{2}\right) + gz_3\right] - \dot{m}_1\left[h_1 + \left(\frac{v_1^2}{2}\right) + gz_1\right] - \dot{m}_2\left[h_2 + \left(\frac{v_2^2}{2}\right) + gz_2\right]$$

For neglected change in potential and kinetic energies, adiabatic and no work interaction:

$$\dot{m}_3 h_3 = \dot{m}_1 h_1 + \dot{m}_2 h_2$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

➢ Heat exchanger:

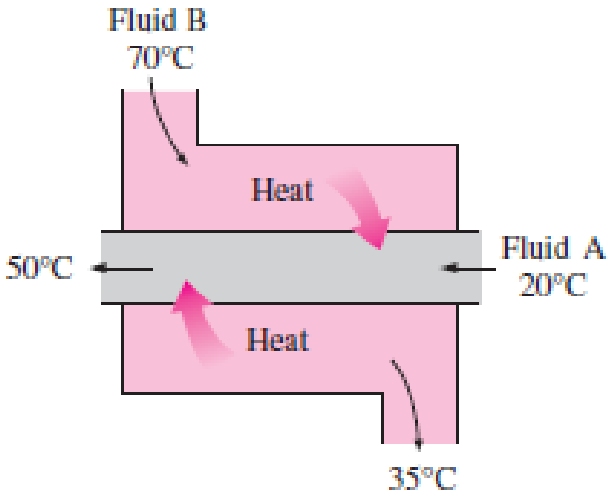
- *Heat exchangers* are devices where two moving fluid streams exchange heat without mixing.

- Energy balance:

$$\dot{Q}_{in} + \underbrace{\dot{W}_{in}}_{\text{Zero}} + \sum \dot{m}_{in}\overbrace{\left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right)}^{\theta_{in}} = \dot{Q}_{out} + \underbrace{\dot{W}_{out}}_{\text{Zero}} + \sum \dot{m}_{out}\overbrace{\left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right)}^{\theta_{out}}$$

- Mass balance:

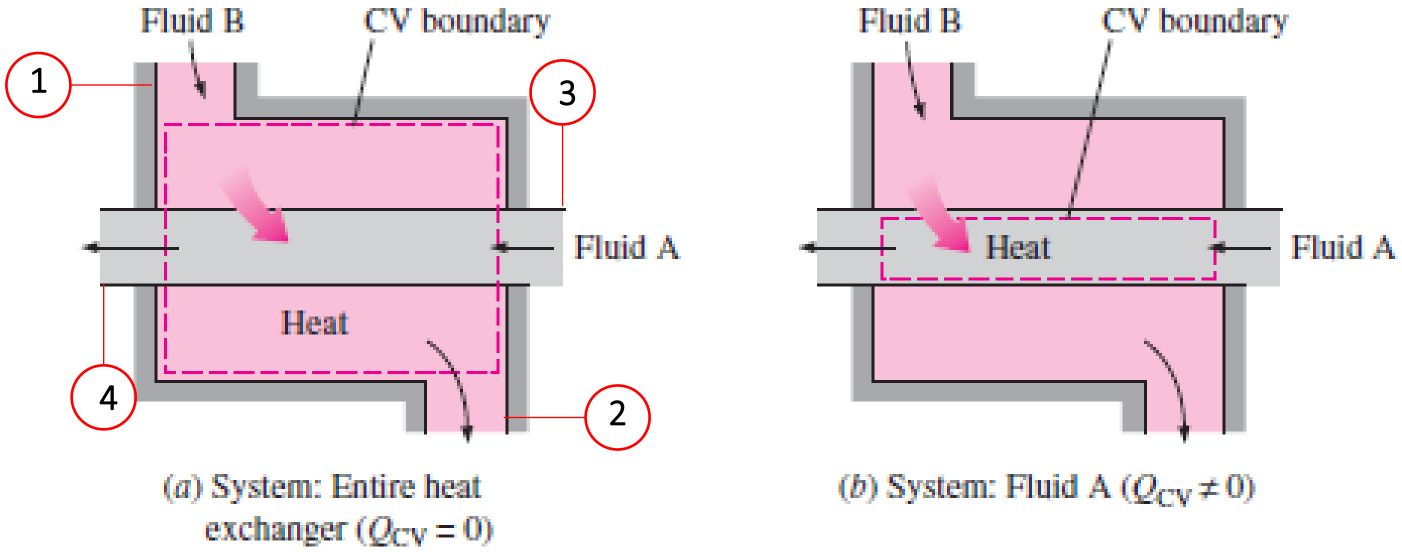
$$\sum \dot{m}_{in} = \sum \dot{m}_{out}$$



4. SOME STEADY-FLOW ENGINEERING DEVICES

Heat exchanger:

The heat transfer associated with a heat exchanger may be zero or nonzero depending on how the control volume is selected.



(a) System: Entire heat exchanger ($Q_{CV} = 0$)

(b) System: Fluid A ($Q_{CV} \neq 0$)

- Energy balance:

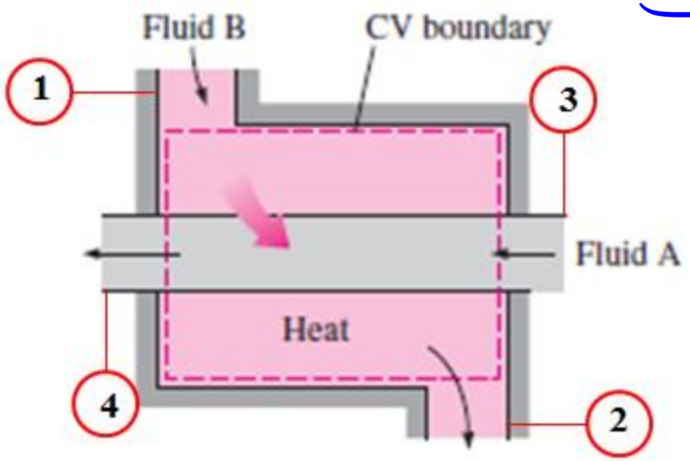
$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}_B\theta_2 + \dot{m}_A\theta_4 - \dot{m}_B\theta_1 - \dot{m}_A\theta_3$$

4. SOME STEADY-FLOW ENGINEERING DEVICES

Heat exchanger:

For neglected change in potential and kinetic energies:

$$\left(\dot{Q}_{in} - \dot{Q}_{out}\right) = \dot{m}_B h_2 + \dot{m}_A h_4 - \dot{m}_B h_1 - \dot{m}_A h_3 = \underbrace{\dot{m}_A\left(h_4 - h_3\right)}_{\dot{Q}_{BA}} - \underbrace{\dot{m}_B\left(h_1 - h_2\right)}_{\dot{Q}_{BA}}$$



For neglected change in potential and kinetic energies and adiabatic process:

$$\dot{m}_B h_2 + \dot{m}_A h_4 = \dot{m}_B h_1 + \dot{m}_A h_3$$
$$\dot{m}_A\left(h_4 - h_3\right) = \dot{m}_B\left(h_1 - h_2\right) = \dot{Q}_{BA}$$

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Geothermal Energy Capacity Building in Egypt (GEB)

Thermodynamics: Exergy analysis

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EXERGY: A MEASURE OF WORK POTENTIAL

- **Objectives:**

1. **Exergy.**

2. **Reversible work.**

3. **Exergy destruction.**

4. **Second-law efficiency.**

5. **Exergy balance.**

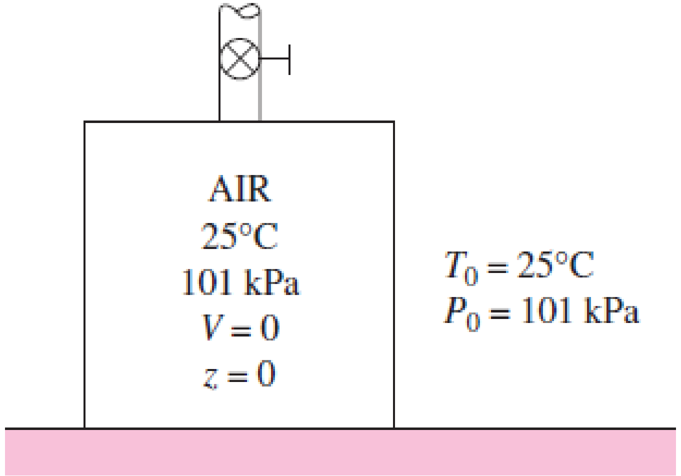
EXERGY: WORK POTENTIAL OF ENERGY

- *The work potential is the amount of energy we can extract as useful work.* This work potential is exergy, which is also called the availability or available energy.
- The system must be in the dead state at the end of the process to maximize the work output.
- A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment. At the dead state, a system is at the temperature and pressure of its environment (in thermal and mechanical equilibrium); it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level); and it does not react with the environment (chemically inert).

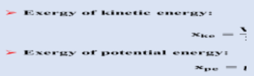
EXERGY: WORK POTENTIAL OF ENERGY

- A system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. This represents the useful work potential of the system at the specified state and is called exergy.

- *It is important to realize that exergy does not represent the amount of work that a work-producing device will actually deliver upon installation. Rather, it represents the upper limit on the amount of work a device can deliver without violating any thermodynamic laws.*



A system that is in equilibrium with its environment is said to be at the dead state


Exergy of kinetic energy:
Exergy of potential energy:

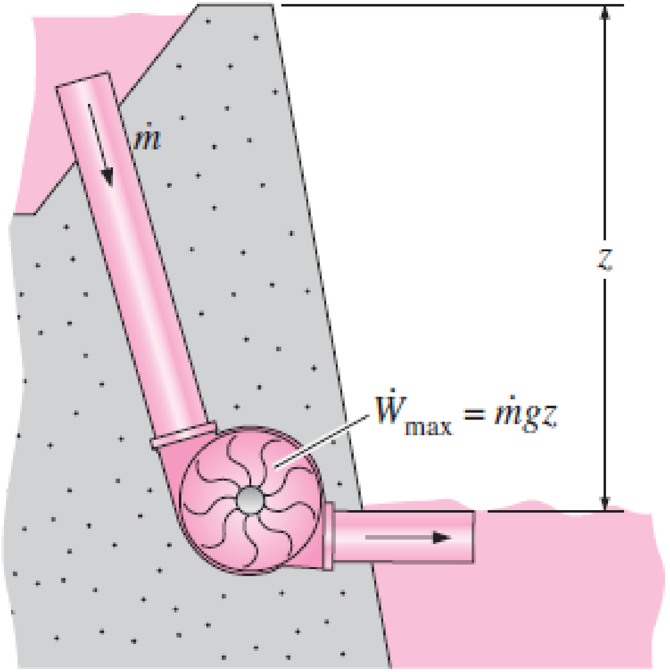
Exergy (Work Potential) Associated with Kinetic and Potential Energy

- Exergy of kinetic energy:

$$\mathbf{ex_{ke}} = \frac{\mathbf{V^2}}{\mathbf{2}} \quad \& \quad \mathbf{EX_{ke}} = \mathbf{m}\frac{\mathbf{V^2}}{\mathbf{2}}$$

- Exergy of potential energy:

$$\mathbf{ex_{pe}} = \mathbf{gZ} \quad \& \quad \mathbf{EX_{pe}} = \mathbf{mgZ}$$



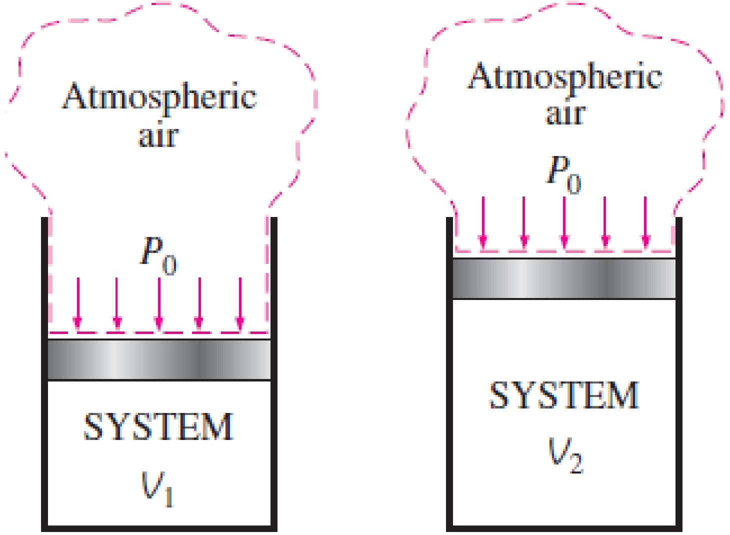
REVERSIBLE WORK AND IRREVERSIBILITY

$$\mathbf{W_{surr} = p_o(V_2 - V_1)}$$

Useful work $$\mathbf{W_u = W - W_{surr} = W - p_o(V_2 - V_1)}$$

- Reversible work W_{rev} is defined as the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the *specified initial and final states*.

- The difference between the reversible work W_{rev} and the useful work W_u is due to the irreversibilities present during the process, and this difference is called irreversibility I.



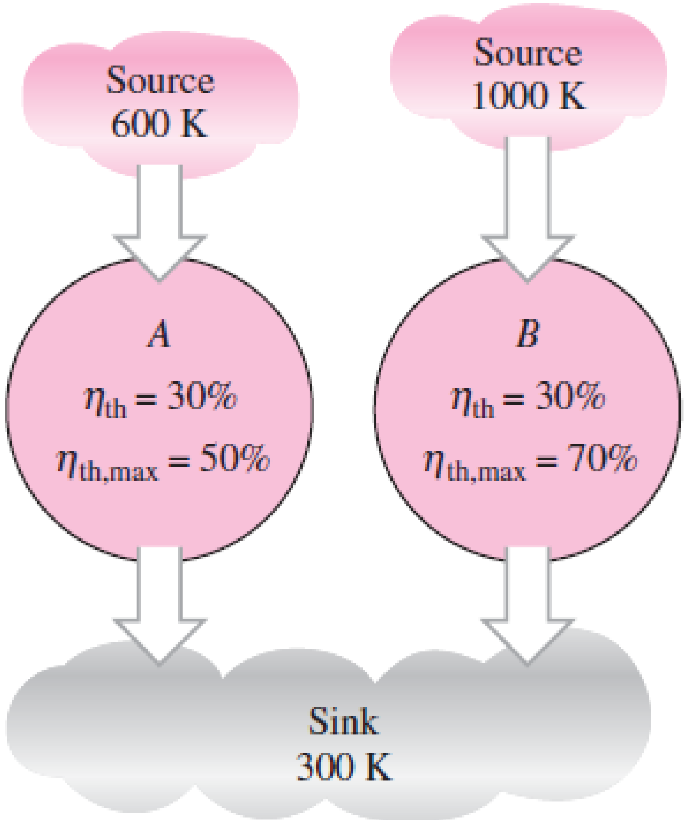
As a closed system expands, some work needs to be done to push the atmospheric air out of the way (W_{surr})

$$\mathbf{I = W_{rev,out} - W_{u,out}} \quad \textbf{or} \quad \mathbf{I = W_{u,in} - W_{rev,in}}$$

SECOND-LAW EFFICIENCY, η_{II}

- Thermal efficiency η_{th} and the coefficient of performance COP for devices as a measure of their performance. They are defined on the basis of the first law only, and they are sometimes referred to as the first-law efficiencies η_{I} . The first law efficiency, however, makes no reference to the best possible performance, and thus it may be misleading.

- *The second-law efficiency η_{II} as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions.*



$$\eta_{rev,A} = \left(1 - \frac{T_L}{T_h}\right) = 1 - \frac{300}{600} = 50\%$$

$$\eta_{rev,B} = \left(1 - \frac{T_L}{T_h}\right) = 1 - \frac{300}{1000} = 70\%$$

SECOND-LAW EFFICIENCY, η_{II}

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}}$$

$$\eta_{II,A} = \frac{0.3}{0.5} = 60\% \quad \text{and} \quad \eta_{II,B} = \frac{0.3}{0.7} = 43\%$$

With respect to the first law, both are similar in performance since they have the same efficiency. With respect to the second law however, Engine A is better in performance than Engine B

$$\eta_{I,B} = \eta_{I,A}$$

$$\eta_{II,B} < \eta_{II,A}$$

SECOND-LAW EFFICIENCY, η_{II}

$$\eta_{II} = \frac{W_u}{W_{rev}} \quad \text{(Work – producing devices)}$$

$$\eta_{II} = \frac{W_{rev}}{W_u} \quad \text{(Work – consuming devices)}$$

$$\eta_{II} = \frac{COP}{COP_{rev}} \quad \text{(Refrigerators and heat pumps)}$$

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{Exergy supplied}}$$

EXERGY CHANGE OF A SYSTEM

Exergy of a Fixed Mass: Non-flow (or Closed System) Exergy

$$-\delta Q - \delta W = dU$$

$$\delta W = \delta W_u + \delta W_{surr} = \delta W_u + p_o dV$$

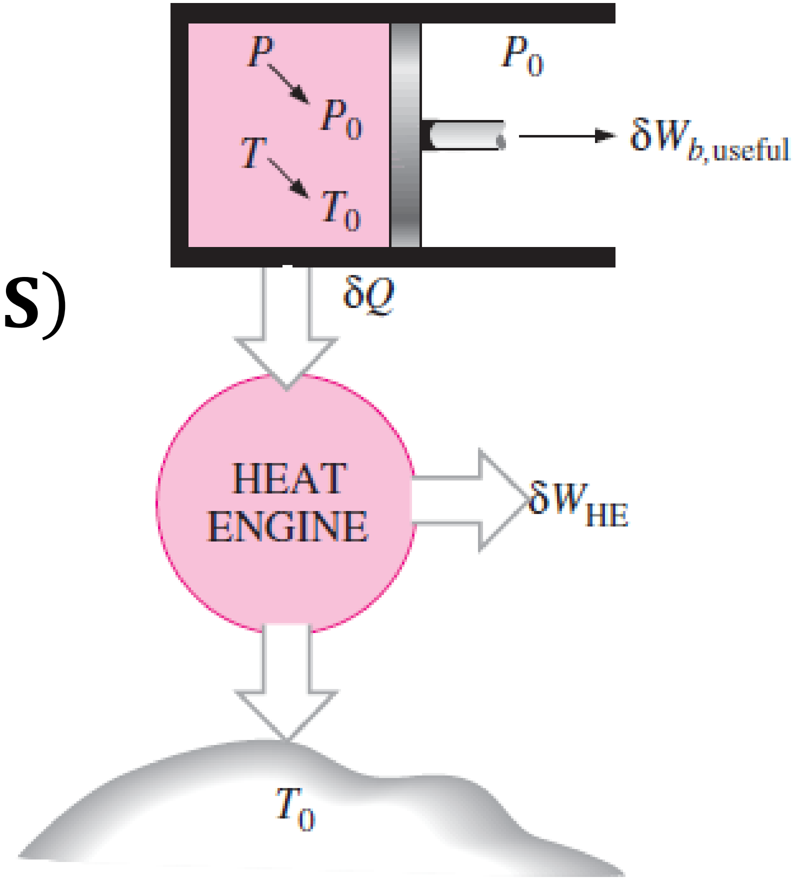
$$\delta W_{HE} = \left(1 - \frac{T_o}{T}\right)\delta Q = \delta Q - T_o \frac{\delta Q}{T} = \delta Q - (-T_o dS)$$

$$\delta Q = \delta W_{HE} - T_o dS$$

$$-\delta W_{HE} + T_o dS - \delta W_u - p_o dV = dU$$

$$\therefore \quad \delta W_{total,u} = \delta W_{HE} + \delta W_u$$

$$\delta W_{total,u} = -dU - p_o dV + T_o dS$$



EXERGY CHANGE OF A SYSTEM

By integration:

$$\mathbf{W_{total,u} = (U - U_o) + p_o(V - V_o) - T_o(S - S_o)}$$

- If the closed system possess kinetic and potential energies:

$$\mathbf{W_{total,u} = (U - U_o) + p_o(V - V_o) - T_o(S - S_o) + m\frac{V^2}{2} + mgZ}$$
$$\mathbf{= EX\ (Exergy)}$$

- Per unit mass:

$$\mathbf{\frac{EX}{m} = \phi = (u - u_o) + p_o(v - v_o) - T_o(s - s_o) + \frac{V^2}{2} + gZ}$$

$$\mathbf{\frac{EX}{m} = \phi = (e - e_o) + p_o(v - v_o) - T_o(s - s_o)}$$

EXERGY CHANGE OF A SYSTEM

- The exergy change of a closed system during a process is simply the difference between the final and initial exergies of the system,

$$\Delta EX = EX_2 - EX_1 = m(\phi_2 - \phi_1) = (E_2 - E_1) + p_o(V_2 - V_1) - T_o(S_2 - S_1)$$

$$= (U_2 - U_1) + p_o(V_2 - V_1) - T_o(S_2 - S_1) + m\left(\frac{V_2^2 - V_1^2}{2}\right) + mg(Z_2 - Z_1)$$

- Per unit mass:

$$\Delta\phi = \phi_2 - \phi_1$$

$$= (u_2 - u_1) + p_o(v_2 - v_1) - T_o(s_2 - s_1) + \left(\frac{V_2^2 - V_1^2}{2}\right) + g(Z_2 - Z_1)$$

The exergy of a closed system is either positive or zero

EXERGY CHANGE OF A SYSTEM

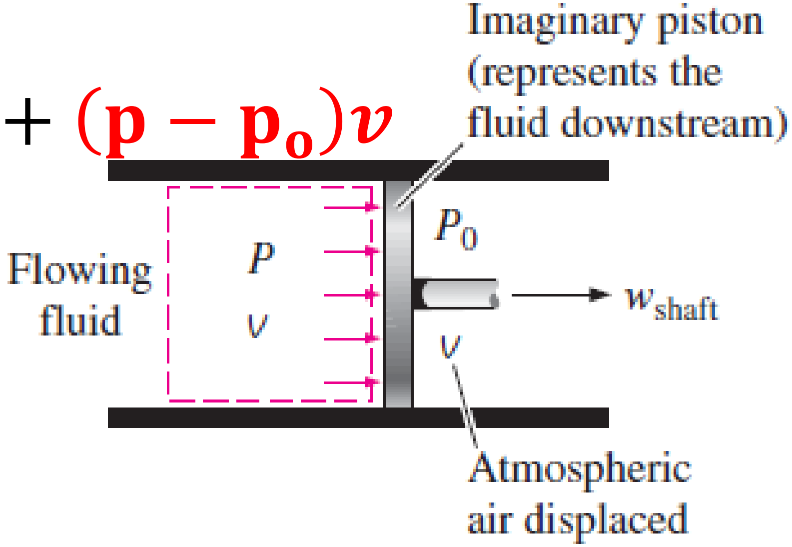
Exergy of a Flow Stream: Flow (or Stream) Exergy

$$\mathbf{ex_{flow} = pv - p_o v = (p - p_o)v}$$

$$\mathbf{ex_{flowing\ fluid}\ \psi = ex_{non-flowing\ fluid}\ \phi + ex_{flow}}$$

$$\psi = (u - u_o) + p_o(v - v_o) - T_o(s - s_o) + \frac{V^2}{2} + gZ + (p - p_o)v$$

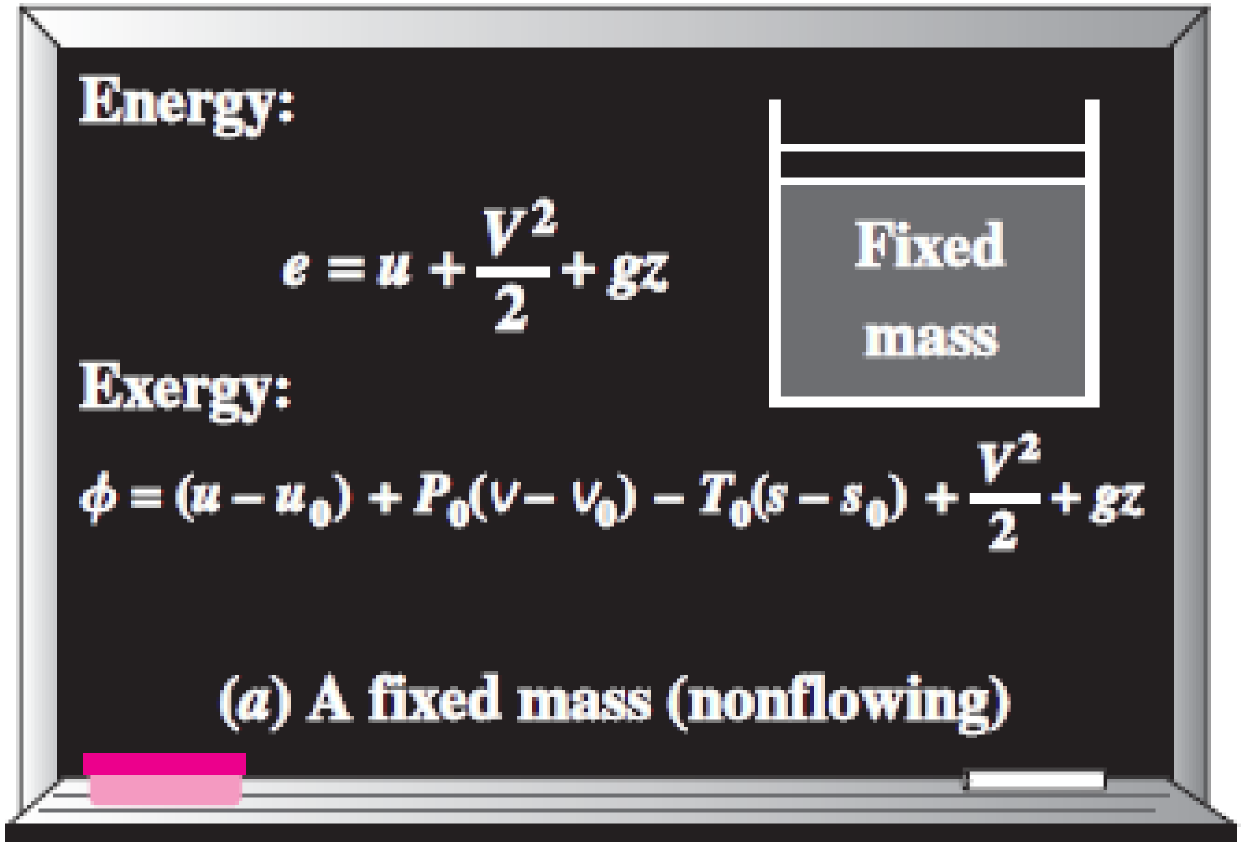
$$\psi = (h - h_o) - T_o(s - s_o) + \frac{V^2}{2} + gZ$$



$$Pv = P_0 v + w_{shaft}$$

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_o(s_2 - s_1) + \left(\frac{V_2^2 - V_1^2}{2}\right) + g(Z_2 - Z_1)$$

EXERGY CHANGE OF A SYSTEM



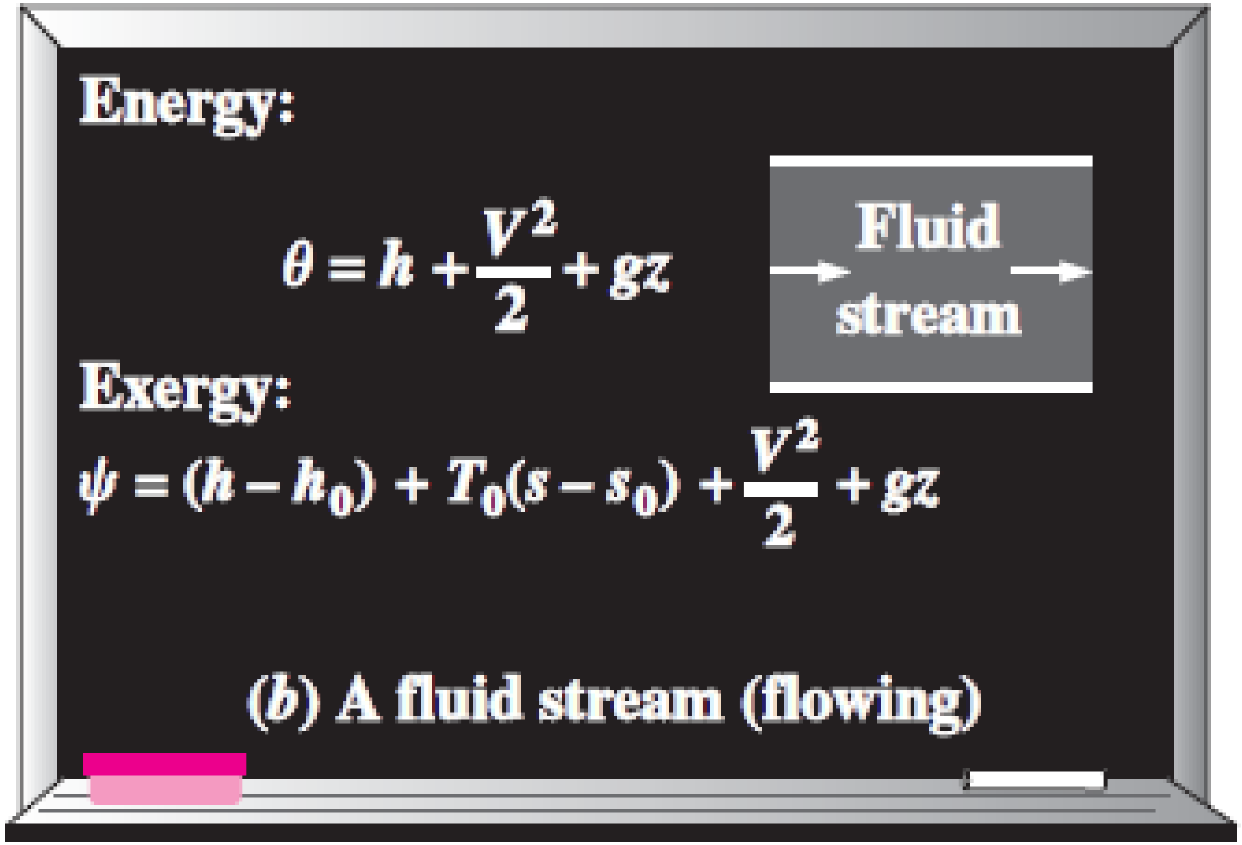
Energy:

$$e = u + \frac{V^2}{2} + gz$$

Exergy:

$$\phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

(*a*) A fixed mass (nonflowing)



Energy:

$$\theta = h + \frac{V^2}{2} + gz$$

Exergy:

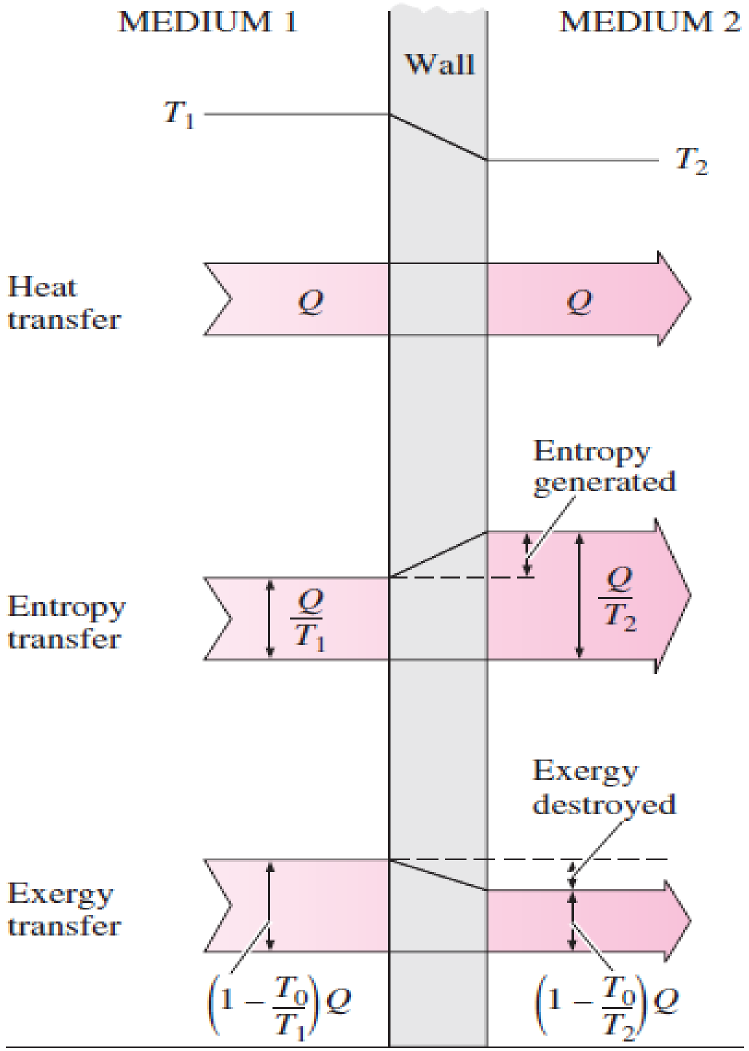
$$\psi = (h - h_0) + T_0(s - s_0) + \frac{V^2}{2} + gz$$

(*b*) A fluid stream (flowing)

EXERGY TRANSFER BY HEAT, WORK AND MASS

Exergy by Heat Transfer, Q

$$\mathbf{EX_{heat}} = \left(1 - \frac{T_o}{T}\right) Q$$



EXERGY TRANSFER BY HEAT, WORK AND MASS

Exergy Transfer by Work, W

$$\mathbf{EX_{work}} = \begin{cases} \mathbf{W - W_{surr}} & \textbf{(for boundary work)} \\ \mathbf{W} & \textbf{(for other forms of work)} \end{cases}$$

Exergy Transfer by Mass, m

$$\mathbf{EX_{mass} = m\psi}$$

THE DECREASE OF EXERGY PRINCIPLE AND EXERGY DESTRUCTION

- Irreversibilities such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, nonquasi-equilibrium compression or expansion always generate entropy, and any-thing that generates entropy always destroys exergy. *The exergy destroyed is proportional to the entropy generated.*

$$\mathbf{EX_{destroyed} = T_o S_{gen} \geq 0 \quad or \quad E\dot{X}_{destroyed} = T_o \dot{S}_{gen} \geq 0}$$

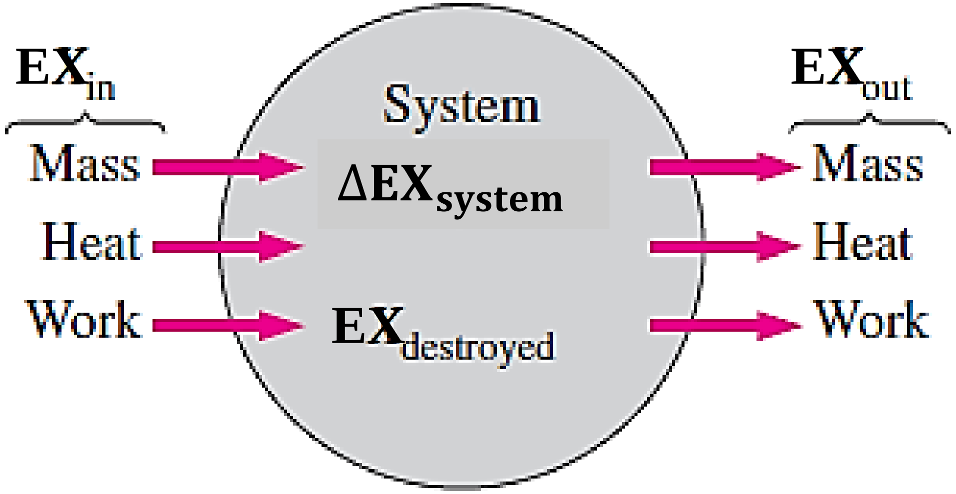
- Exergy destroyed represents the lost work potential and is also called the irreversibility or lost work.

$$\mathbf{EX_{destroyed}} \begin{cases} > 0 & \textbf{Irreversible process} \\ = 0 & \textbf{reversible process} \\ < 0 & \textbf{Impossible process} \end{cases}$$

EXERGY BALANCE: CLOSED SYSTEMS

$$\begin{pmatrix}\text{Total}\\ \text{exergy}\\ \text{entering}\end{pmatrix} - \begin{pmatrix}\text{Total}\\ \text{exergy}\\ \text{leaving}\end{pmatrix} - \begin{pmatrix}\text{Total}\\ \text{exergy}\\ \text{destroyed}\end{pmatrix} = \begin{pmatrix}\text{Change in the}\\ \text{total exergy}\\ \text{of the syatem}\end{pmatrix}$$

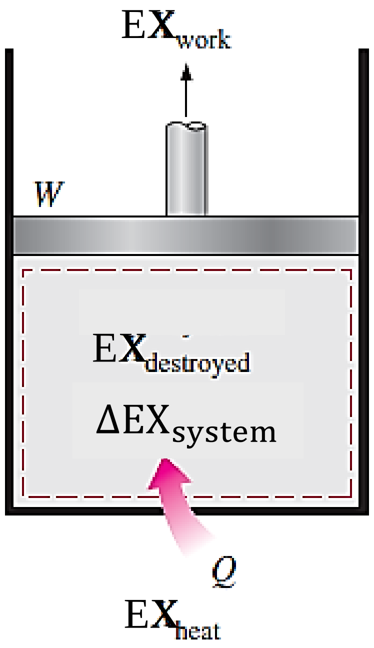
$$\mathrm{EX_{in}} - \mathrm{EX_{out}} - \mathrm{EX_{destroyed}} = \Delta \mathrm{EX_{system}} \quad \mathrm{(J)}$$



EXERGY BALANCE: CLOSED SYSTEMS

$$\pm\sum\left(1-\frac{T_o}{T_i}\right)Q_i \pm [W - p_o(V_2 - V_1)] - T_oS_{gen} = \Delta EX_{system}(J)$$

Rate form: $$\pm\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i \pm \left[\dot{W} - p_o\frac{dV_{system}}{dt}\right] - T_o\dot{S}_{gen} = \frac{dEX_{system}}{dt}(J/s)$$



EXERGY BALANCE: CONTROL VOLUMES

$$\pm\sum\left(1-\frac{T_o}{T_i}\right)Q_i \pm [W - p_o(V_2 - V_1)] + \sum_{in} m\,\psi - \sum_{out} m\,\psi - T_o S_{gen}$$
$$= \Delta EX_{CV}(J)$$

$$\pm\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i \pm \left[\dot{W} - p_o\frac{dV_{CV}}{dt}\right] + \sum_{in} \dot{m}\,\psi - \sum_{out} \dot{m}\,\psi - T_o\dot{S}_{gen} = \frac{dEX_{CV}}{dt}(J/s)$$

Exergy Balance for Steady-Flow Systems

$$\pm\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i \pm \dot{W} + \sum_{in} \dot{m}\,\psi - \sum_{out} \dot{m}\,\psi - T_o\dot{S}_{gen} = 0$$

If single stream:

$$\pm\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i \pm \dot{W} + \dot{m}(\psi_{in} - \psi_{out}) - T_o\dot{S}_{gen} = 0$$

When $\dot{EX}_{destroyed} = T_o\dot{S}_{gen} = 0 \quad \rightarrow \quad \dot{W} = \dot{W}_{rev}$

$$\dot{W}_{rev} = \pm\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i \pm \dot{m}(\psi_{in} - \psi_{out})$$

If adiabatic: $\dot{W}_{rev} = \pm\dot{m}(\psi_{in} - \psi_{out})$

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Geothermal Energy Capacity Building in Egypt (GEB)

Power generation from high-enthalpy geothermal resources: Dry-steam steam power plants

Energy and exergy analysis. Working parameters optimization

Co-funded by the
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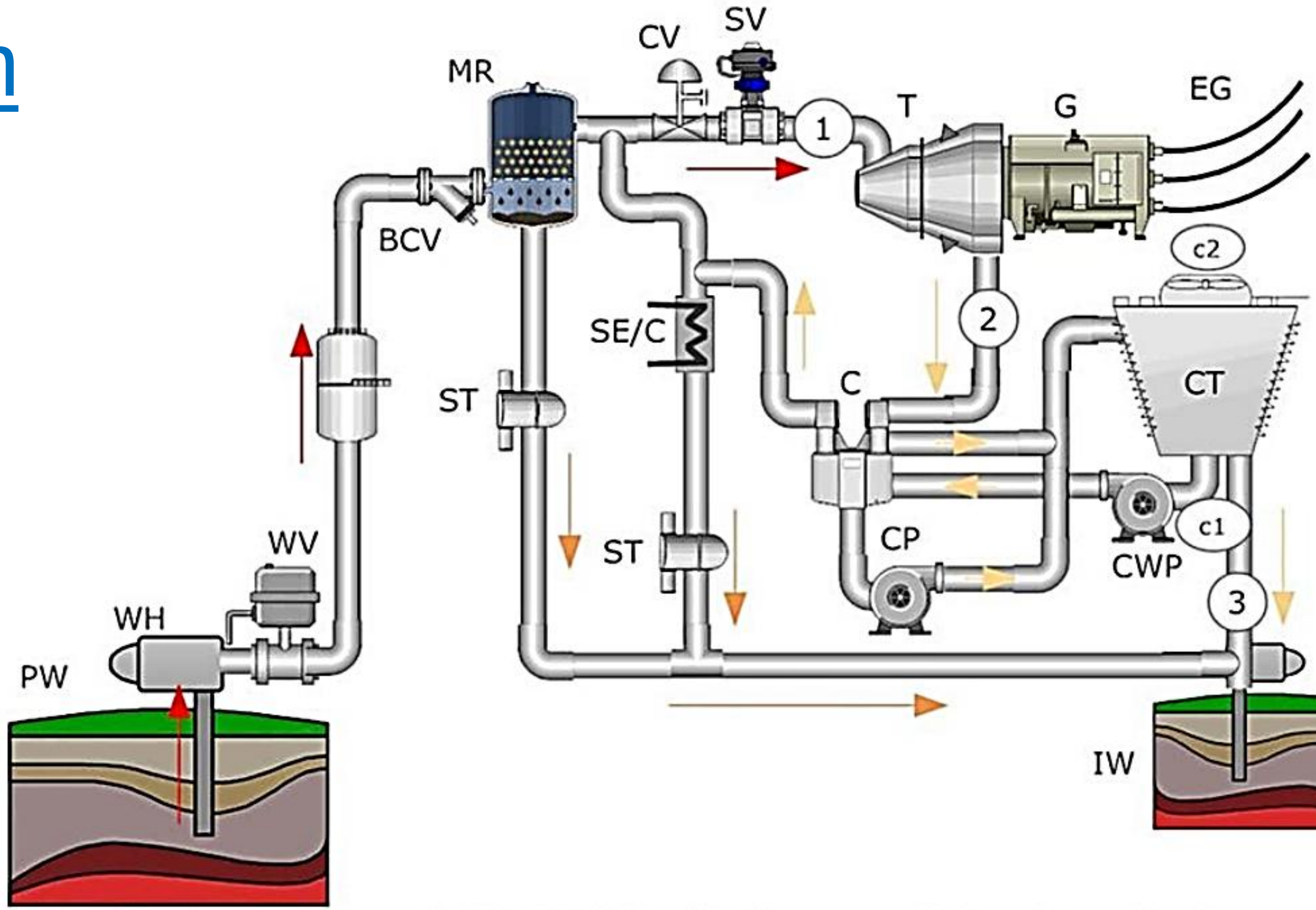
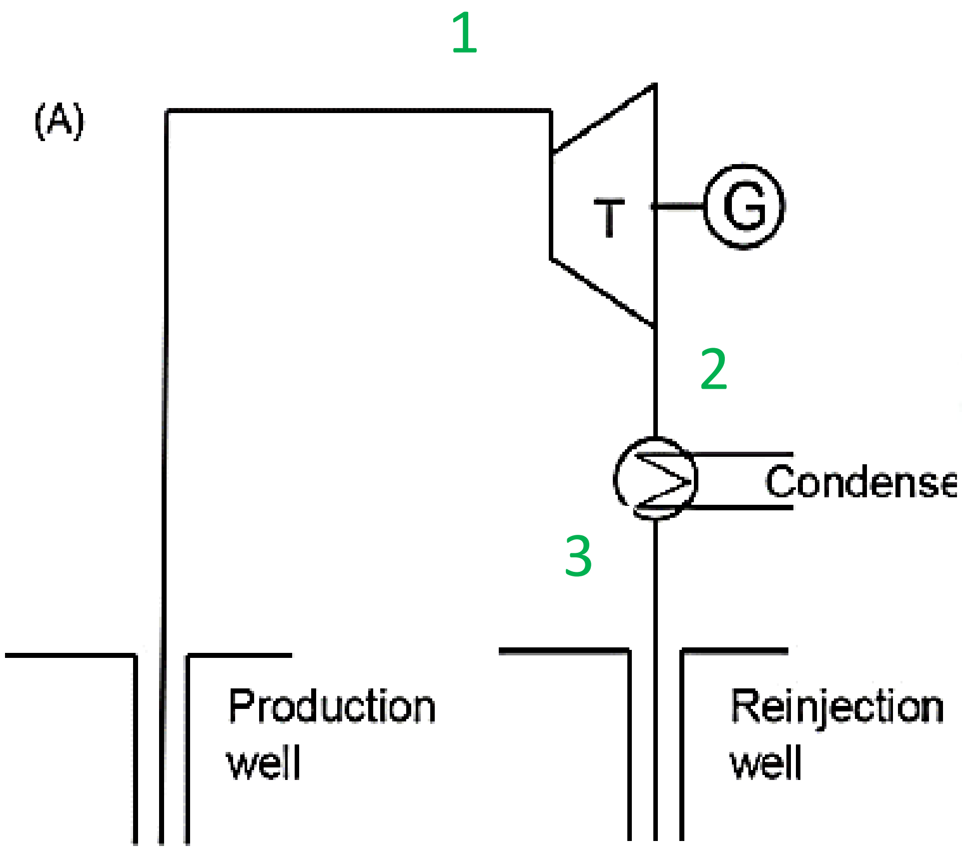
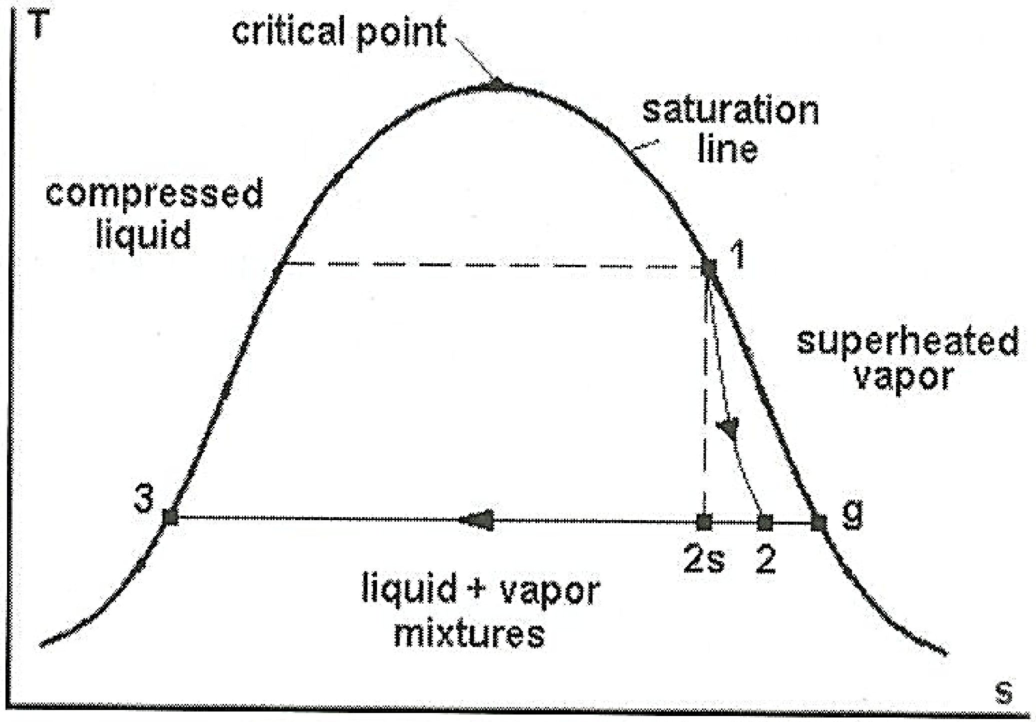
Dry steam



PW	Production well	BCV	Ball check valve	SE	Steam jet ejectors
WH	Wellhead	MR	Moisture remover	C	Condenser
WV	Well valves	ST	Steam tramps	CP	Condensate pump
IW	Injection Wells	CV	Control valve	CWP	Condensed water pump
EG	Electric grid	SV	Stop valve	T/G	Turbine- Generator

Dry steam


1
(A)
T
G
2
Condense
3
Production well
Reinjection well


T
critical point
saturation line
compressed liquid
1
superheated vapor
3
g
2s
2
liquid + vapor mixtures
s

Dry steam: Energy analysis

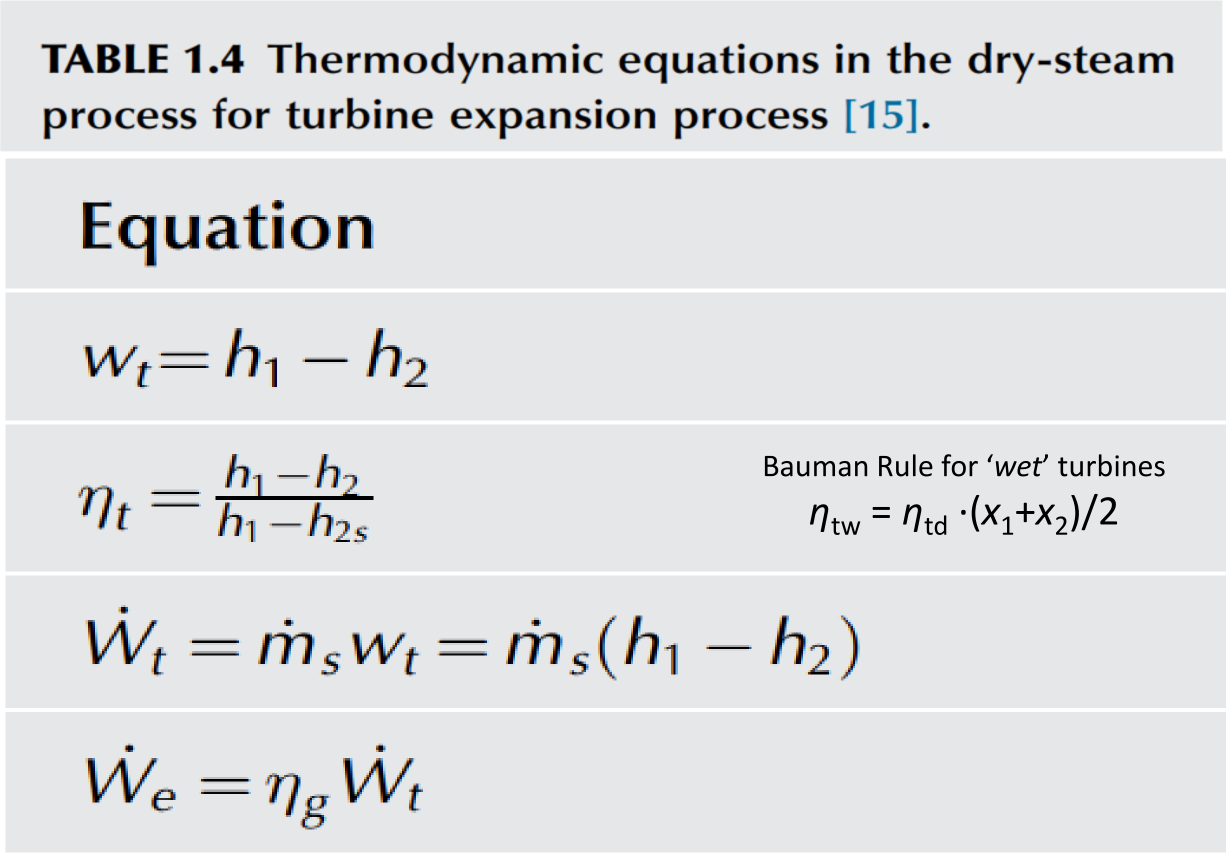


TABLE 1.4 Thermodynamic equations in the dry-steam process for turbine expansion process [15].

Equation
$w_t = h_1 - h_2$
$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}}$
$\dot{W}_t = \dot{m}_s w_t = \dot{m}_s (h_1 - h_2)$
$\dot{W}_e = \eta_g \dot{W}_t$

Bauman Rule for ‘*wet*’ turbines

$\eta_{tw} = \eta_{td} \cdot (x_1 + x_2)/2$

Dry steam: Exergy analysis

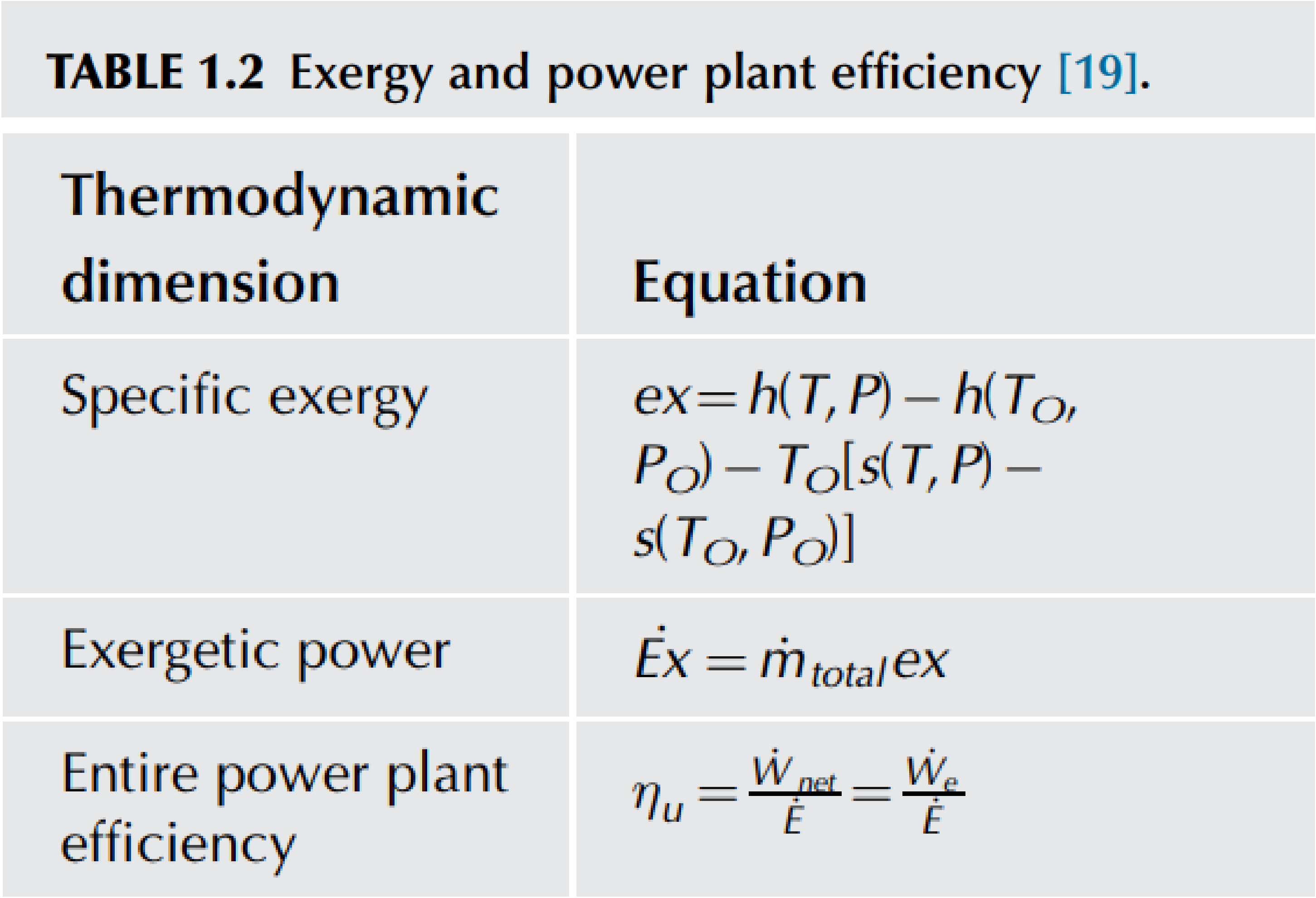


TABLE 1.2 Exergy and power plant efficiency [19].

Thermodynamic dimension	Equation
Specific exergy	$ex = h(T, P) - h(T_O, P_O) - T_O[s(T, P) - s(T_O, P_O)]$
Exergetic power	$\dot{E}x = \dot{m}_{total} ex$
Entire power plant efficiency	$\eta_u = \frac{\dot{W}_{net}}{\dot{E}} = \frac{\dot{W}_e}{\dot{E}}$

Second-Law Analysis

Turbine:

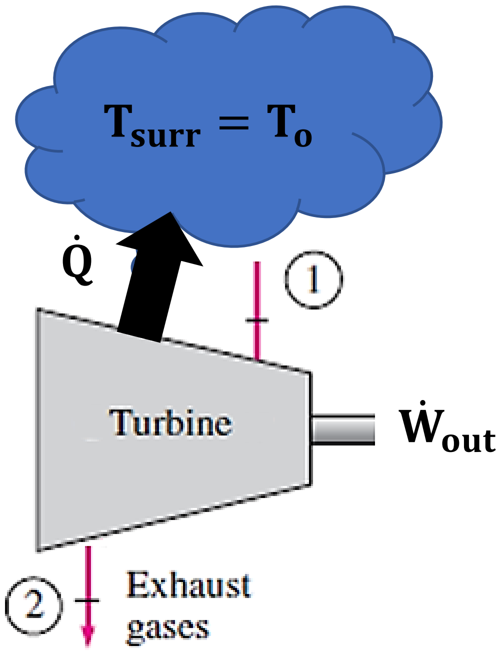
Exergy balance:

$$\dot{EX}_{in} - \dot{EX}_{out} - \dot{EX}_{loss} = \frac{dEX_{sys}}{dt} = 0\ (steady-state, steady-flow)$$

$$\dot m ex_1 - \dot W_{out} - \dot m ex_2 - \dot{EX}_{loss} = 0$$

$$\dot{EX}_{loss} = \dot{EX}_{supplied} - \dot{EX}_{recovered}$$

$$\dot{EX}_{loss} = \underbrace{\boxed{[\dot m(ex_1 - ex_2)]}}_{\dot{EX}_{supplied}} - \underbrace{\boxed{\dot W_{out}}}_{\dot{EX}_{recovered}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

The second-law efficiency:

$$\eta_{II,T} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

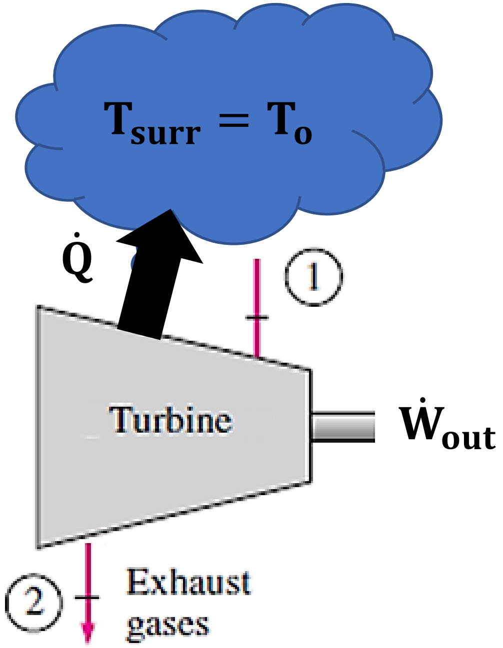
$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}(ex_1 - ex_2)}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}[(h_1 - h_2) - T_o(s_1 - s_2)]}$$

But the first law of thermodynamics requires:

$$\dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{W}_{out} + \dot{Q} - \dot{m}T_o(s_1 - s_2)} = \frac{\dot{W}_{out}}{\dot{W}_{rev}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady - state, steady - flow)}$$

$$\dot{m}s_1 - \dot{m}s_2 - \left(\frac{\dot{Q}}{T_{surr}}\right) + \dot{S}_{gen} = 0$$

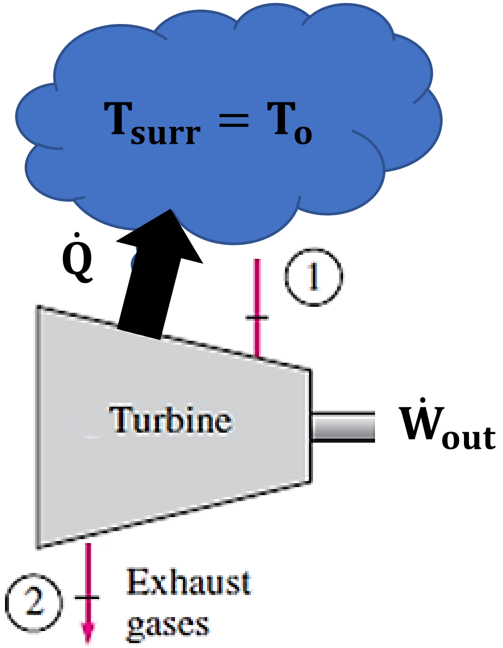
$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \left(\frac{\dot{Q}}{T_{surr}}\right) \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \left(\frac{T_o}{T_o}\right)\dot{Q} \geq 0$$

$$\text{But}: \dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \dot{m}(h_1 - h_2) - \dot{W}_{out} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = [\dot{m}(ex_1 - ex_2)] - \dot{W}_{out}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Condenser (Heat Exchanger – HX):

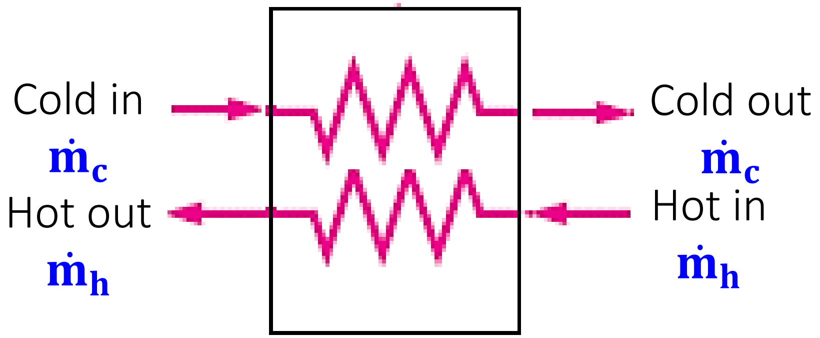
Exergy balance:

$$\dot{EX}_{in} - \dot{EX}_{out} - \dot{EX}_{loss} = \frac{dEX_{sys}}{dt} = 0\ (steady - state, steady - flow)$$

$$\dot{m}_c ex_{c,in} + \dot{m}_h ex_{h,in} - \dot{m}_c ex_{c,out} - \dot{m}_h ex_{h,out} - \dot{EX}_{loss} = 0$$

$$\dot{EX}_{loss} = \dot{EX}_{supplied} - \dot{EX}_{recovered}$$

$$\dot{EX}_{loss} = \underbrace{\boxed{\dot{m}_h\left(ex_{h,in} - ex_{h,out}\right)}}_{\dot{EX}_{supplied}} - \underbrace{\boxed{\dot{m}_c\left(ex_{c,out} - ex_{c,in}\right)}}_{\dot{EX}_{recovered}}$$



T_o: is the dead-state temperature.

In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser (Heat Exchanger – HX):

The second-law efficiency:

$$\eta_{II,HX} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

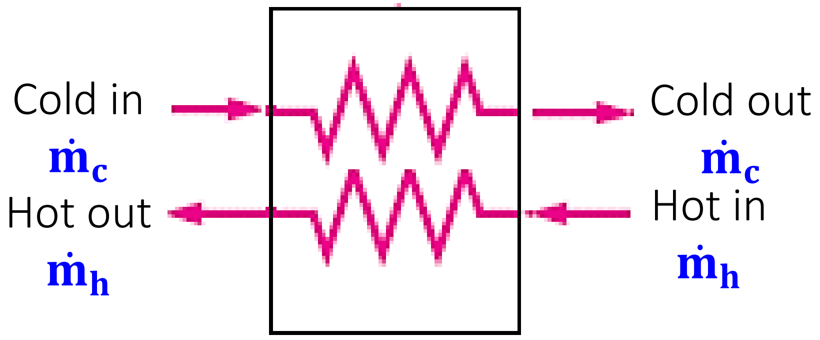
$$\eta_{II,HX} = \frac{\dot{m}_c\left(ex_{c,out} - ex_{c,in}\right)}{\dot{m}_h\left(ex_{h,in} - ex_{h,out}\right)}$$

$$\eta_{II,HX} = \frac{\dot{m}_c\left[\left(h_{c,out} - h_{c,in}\right) - T_o\left(s_{c,out} - s_{c,in}\right)\right]}{\dot{m}_h\left[\left(h_{h,in} - h_{h,out}\right) - T_o\left(s_{h,in} - s_{h,out}\right)\right]}$$

But the first law of thermodynamics requires:

$$\dot{m}_c\left(h_{c,out} - h_{c,in}\right) = \dot{m}_h\left(h_{h,in} - h_{h,out}\right)$$

$$\eta_{II,HX} = \frac{\dot{m}_h\left(h_{h,in} - h_{h,out}\right) - \dot{m}_c T_o\left(s_{c,out} - s_{c,in}\right)}{\dot{m}_h\left[\left(h_{h,in} - h_{h,out}\right) - T_o\left(s_{h,in} - s_{h,out}\right)\right]}$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser (Heat Exchanger – HX):

Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady – state, steady – flow)}$$

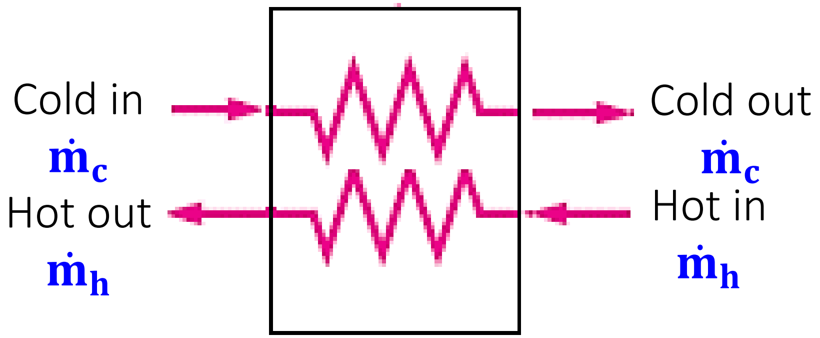
$$\dot{m}_c s_{c,in} + \dot{m}_h s_{h,in} - \dot{m}_c s_{c,out} - \dot{m}_h s_{h,out} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_c \left(s_{c,out} - s_{c,in}\right) - \dot{m}_h \left(s_{h,in} - s_{h,out}\right) \geq 0$$

$$\dot{m}_c \left(s_{c,out} - s_{c,in}\right) \geq \dot{m}_h \left(s_{h,in} - s_{h,out}\right)$$

$$\dot{EX}_{loss} = T_o \dot{S}_{gen}$$

$$\dot{EX}_{loss} = \dot{m}_c T_o \left(s_{c,out} - s_{c,in}\right) - \dot{m}_h T_o \left(s_{h,in} - s_{h,out}\right) \geq 0$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Dry steam cycle:

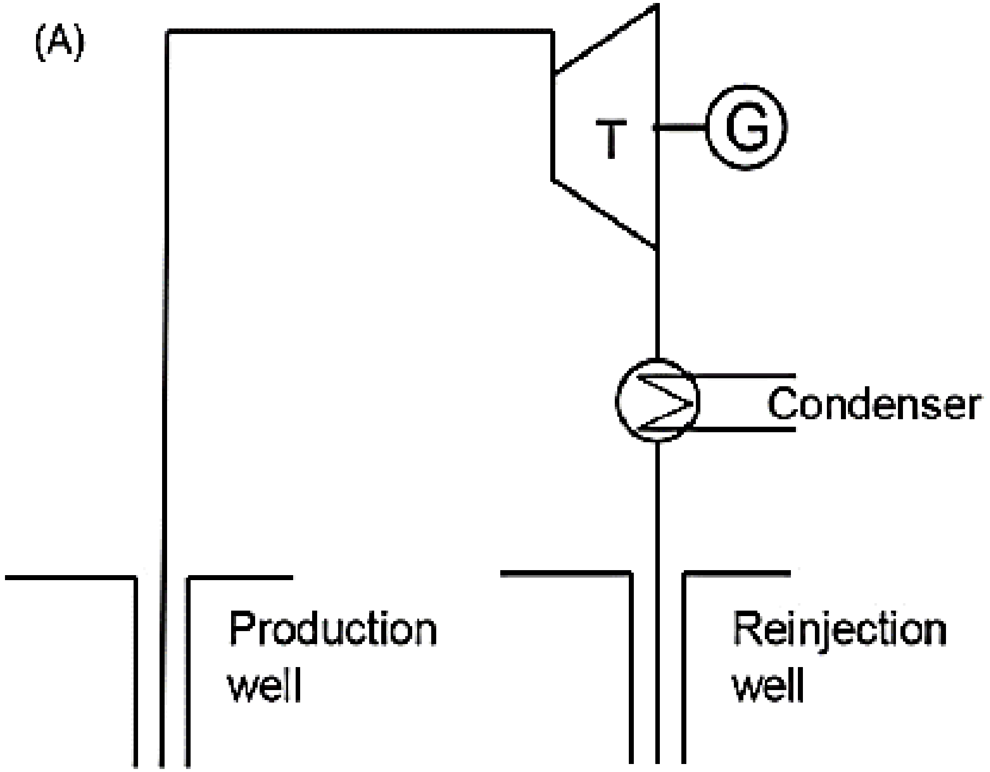
Exergy balance:

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{loss} = \frac{dX_{sys}}{dt} = 0\ (\text{steady} - \text{state}, \text{steady} - \text{flow})$$

$$\dot{m}ex_{production} - \dot{W}_{net} - \dot{m}ex_{reinjection} - \dot{X}_{loss} = 0$$

$$\dot{X}_{loss} = \dot{X}_{supplied} - \dot{X}_{recovered}$$

$$\dot{X}_{loss} = \underbrace{\boxed{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}}_{\dot{X}_{supplied}} - \underbrace{\boxed{\dot{W}_{net}}}_{\dot{X}_{recovered}}$$



Second-Law Analysis

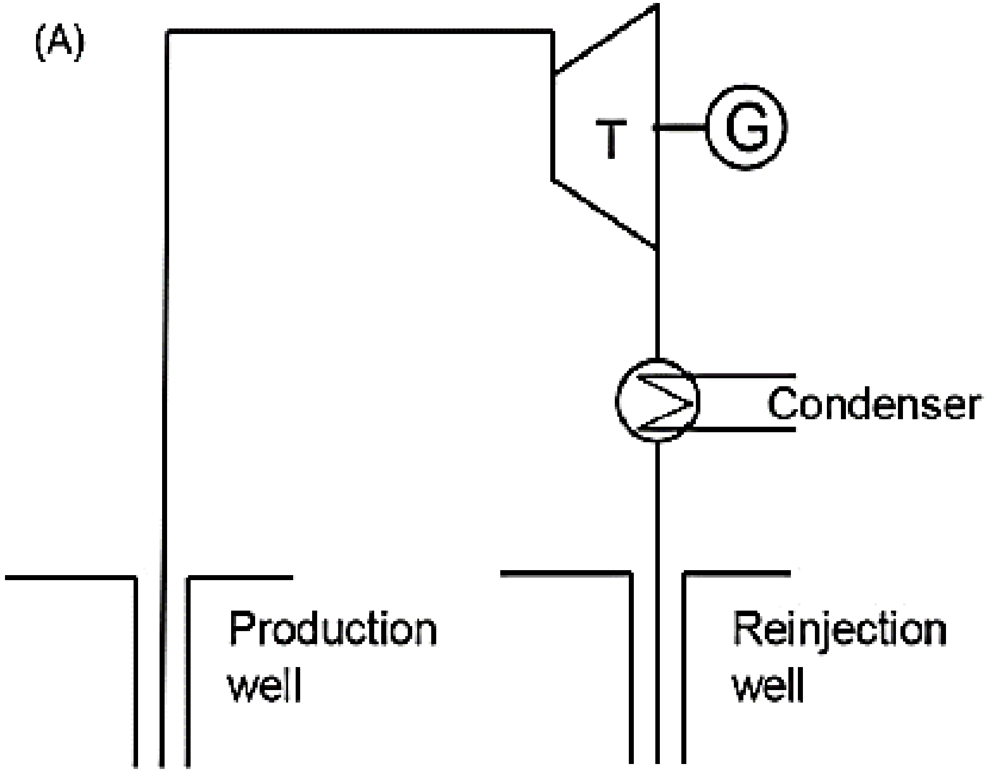
Dry steam cycle:

The second-law efficiency:

$$\eta_{II,R} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}[\Delta h - T_o \Delta s]}$$



Second-Law Analysis

Dry steam cycle:

Entropy balance:

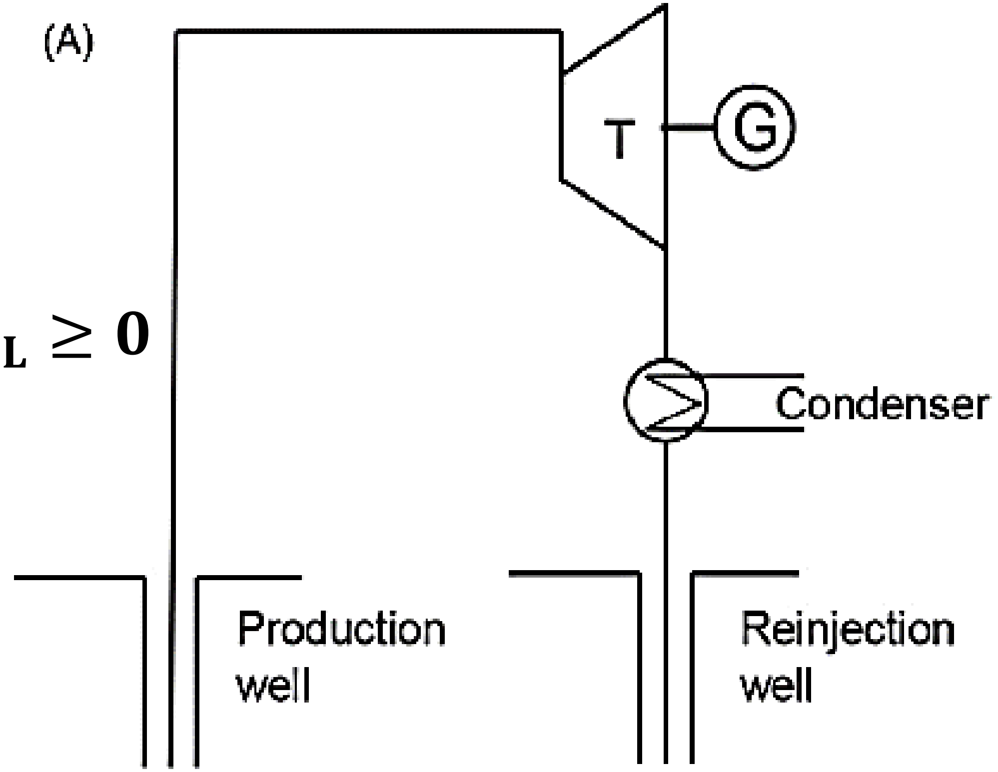
$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady} - \text{state, steady} - \text{flow)}$$

$$\dot{m}s_{production} - \dot{m}s_{reinjection} - \frac{\dot{Q}_L}{T_L} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}\left(s_{reinjection} - s_{production}\right) + \frac{\dot{Q}_L}{T_L} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o\left(s_{reinjection} - s_{production}\right) + \dot{Q}_L \geq 0$$

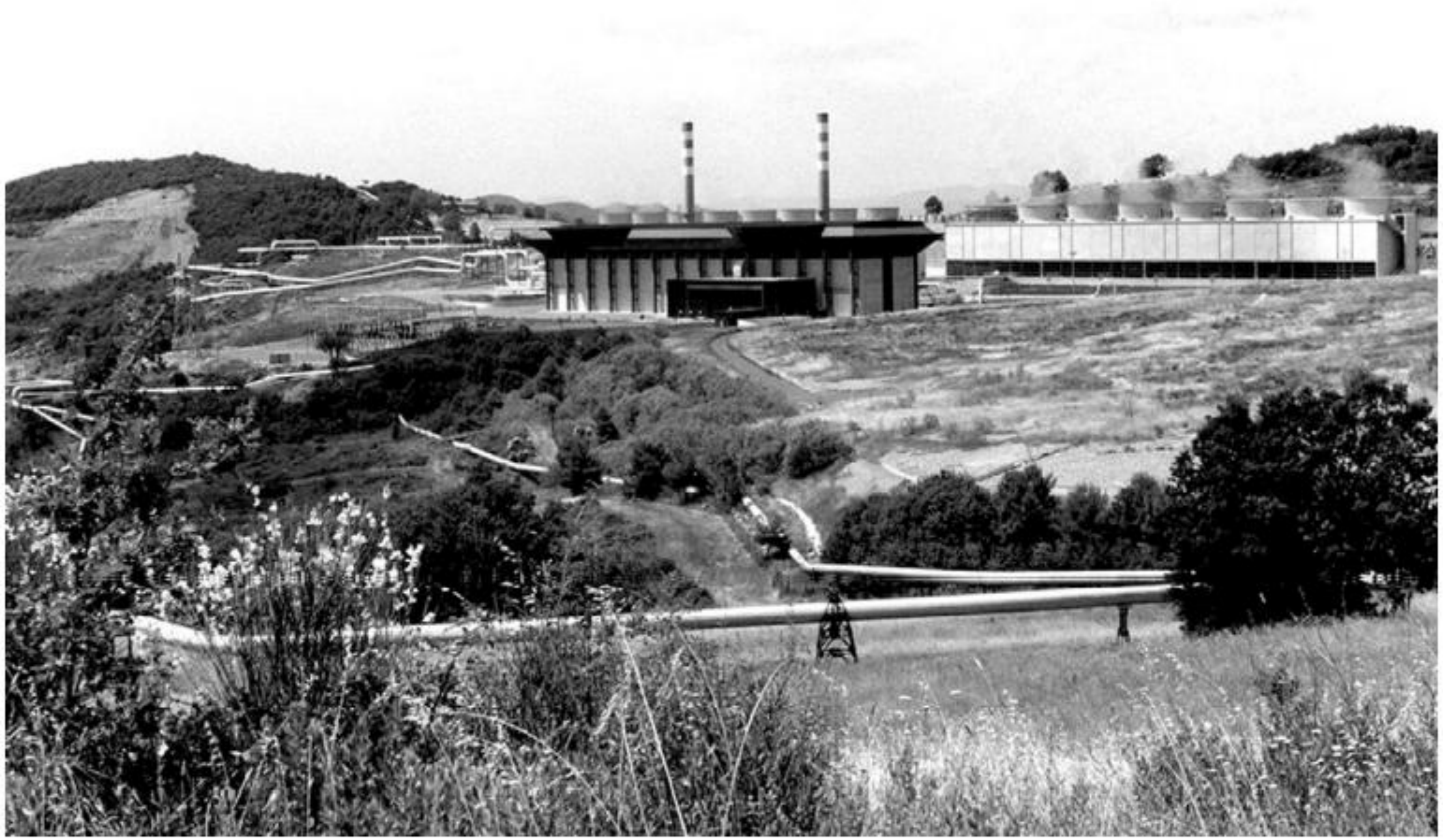
Where $T_o = T_L$



Dry steam (Larderello)

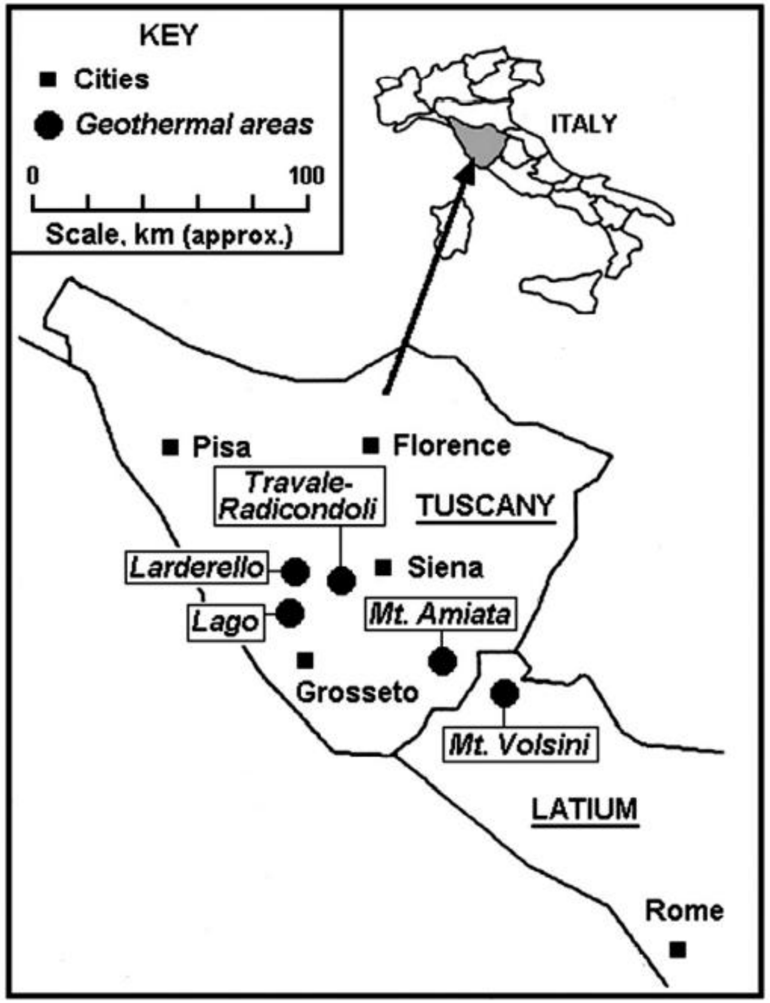
Larderello (Italy). Dry steam. 250 ºC.
Current power capacity installed: 500 MW.

Dry steam (Larderello)



Valle Secolo geothermal power plant, Larderello, Italy; photo courtesy of Enel Green Power.

Dry steam (Larderello)



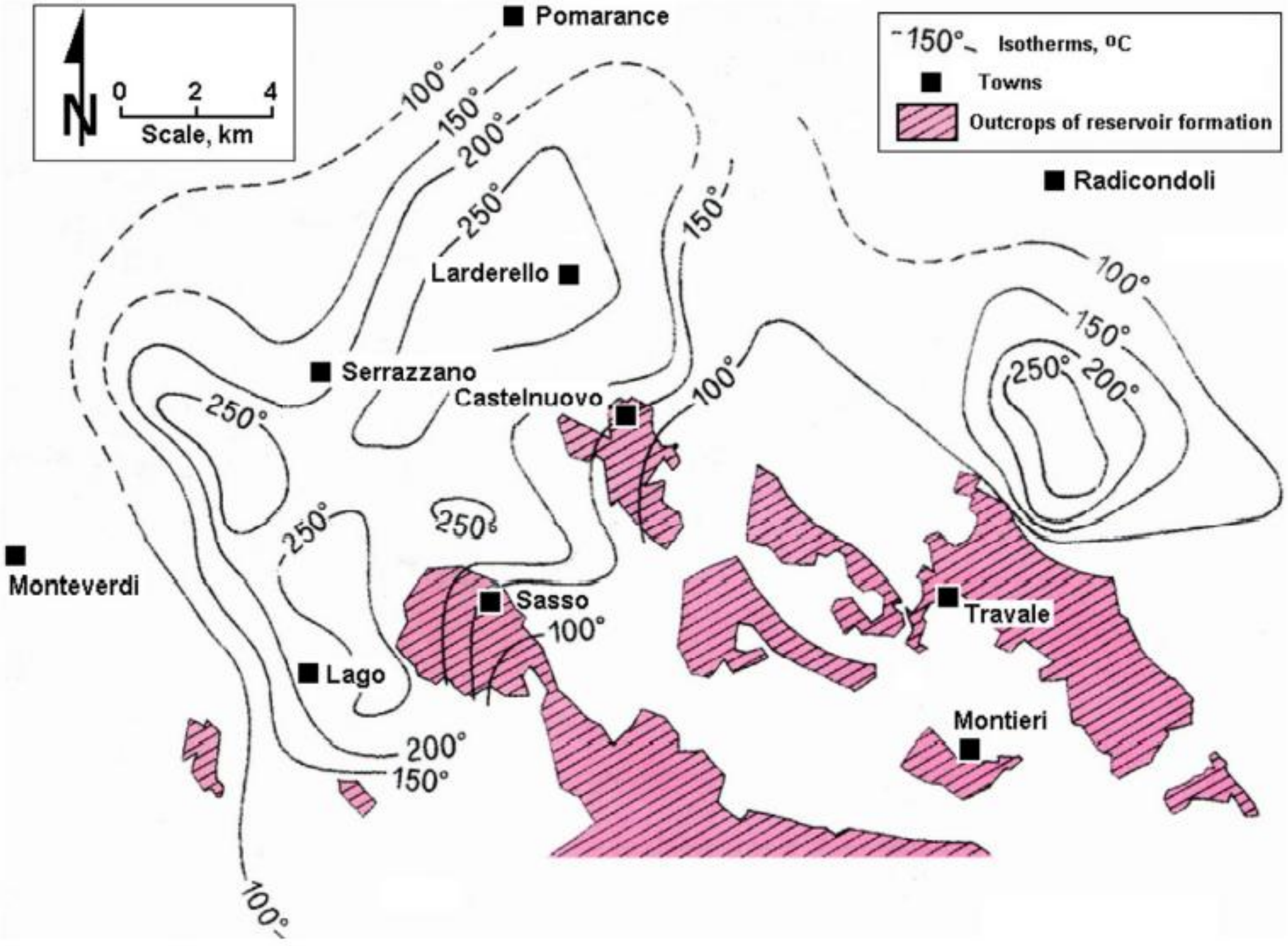


Figure 11.3 Isotherms at the top of the reservoir in greater Larderello area. *After Ref. [4] [WWW].*

Dry steam (Larderello)

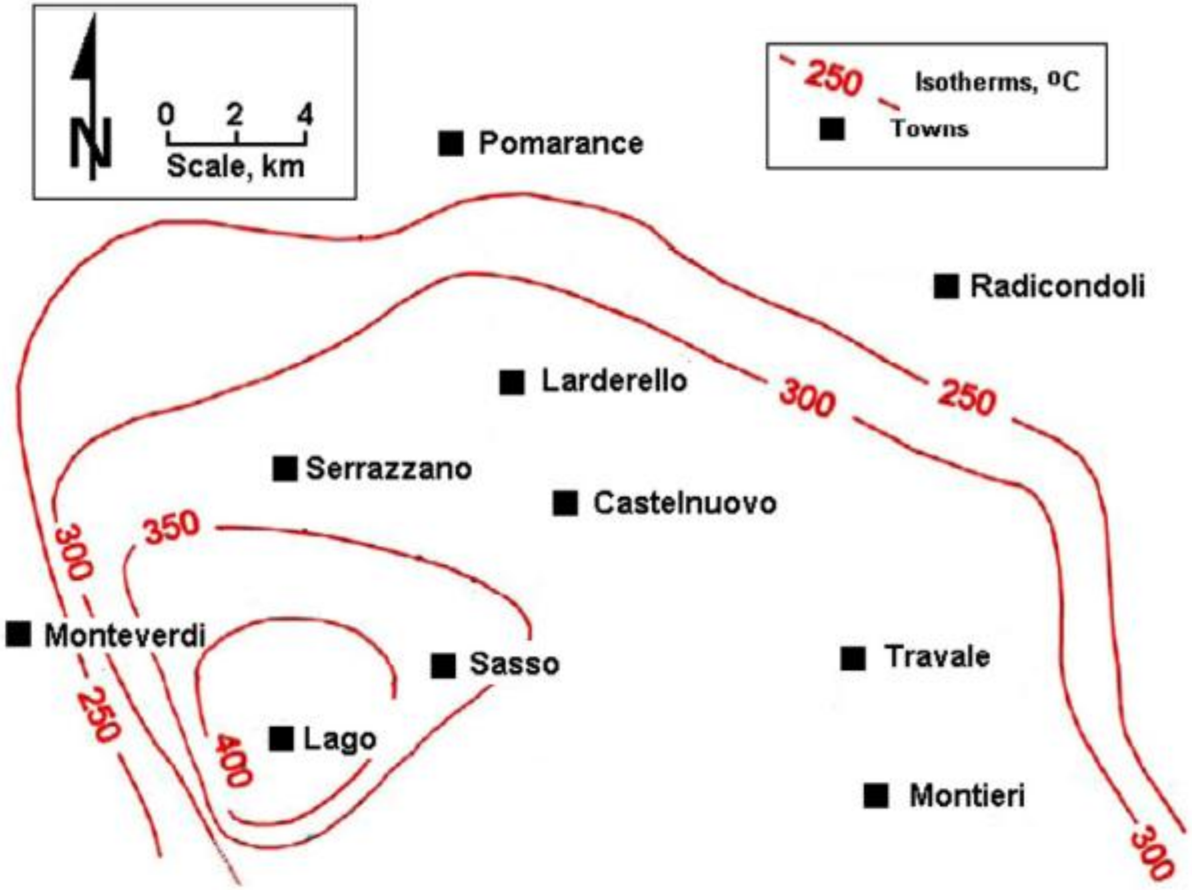


Figure 11.5 Isotherms at 3000 m b.s.l. in greater Larderello area. *Highly simplified from Ref. [6] [WWW].*

Dry steam (Larderello)

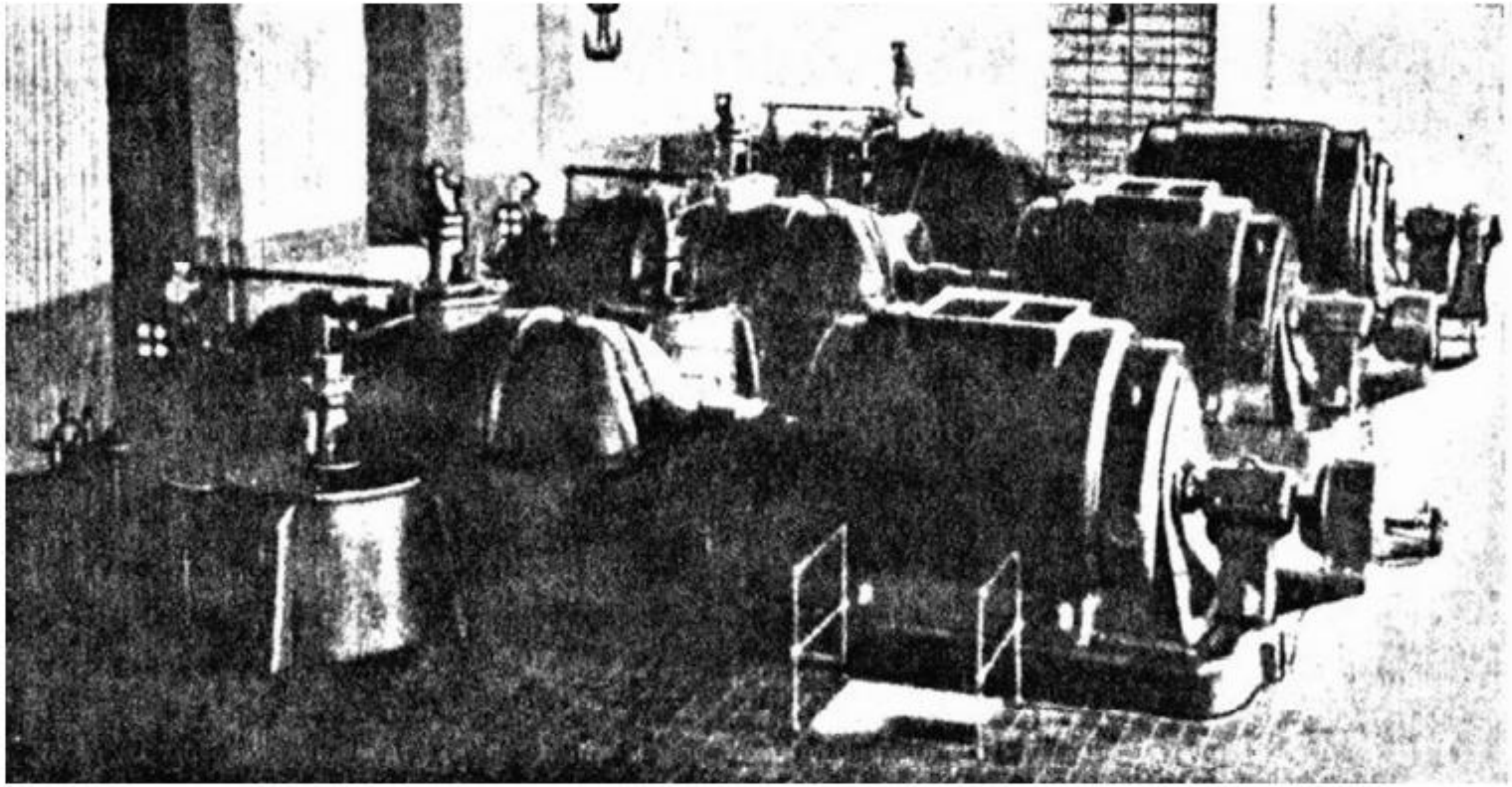


Figure 11.9 Three 2.5 MW steam turbine-generator units at Larderello, c. 1916 [11].

Dry steam (Larderello)

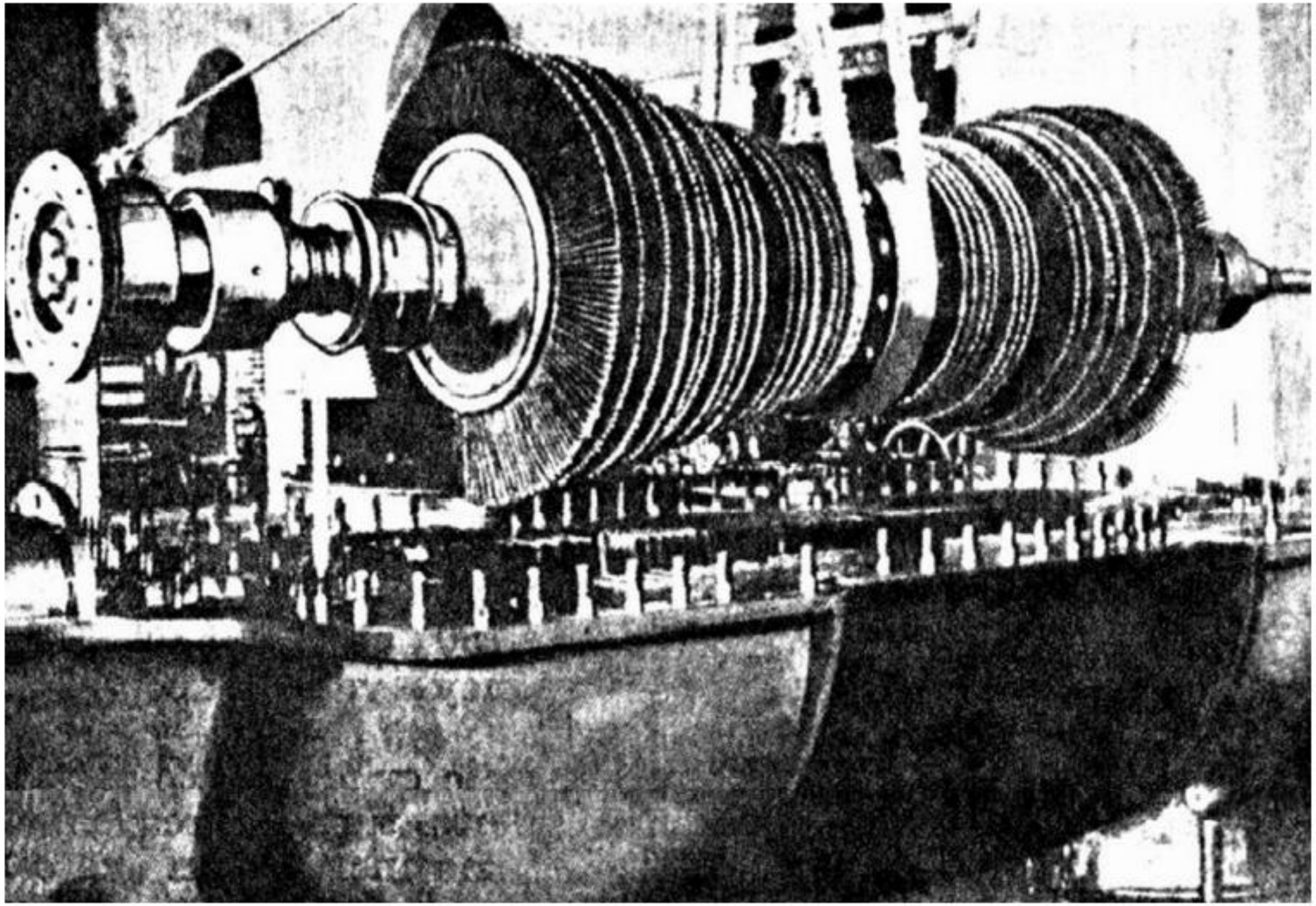


Figure 11.10 Rotor from a 2.5 MW Larderello unit, c. 1916 [11].

Dry steam (Larderello)

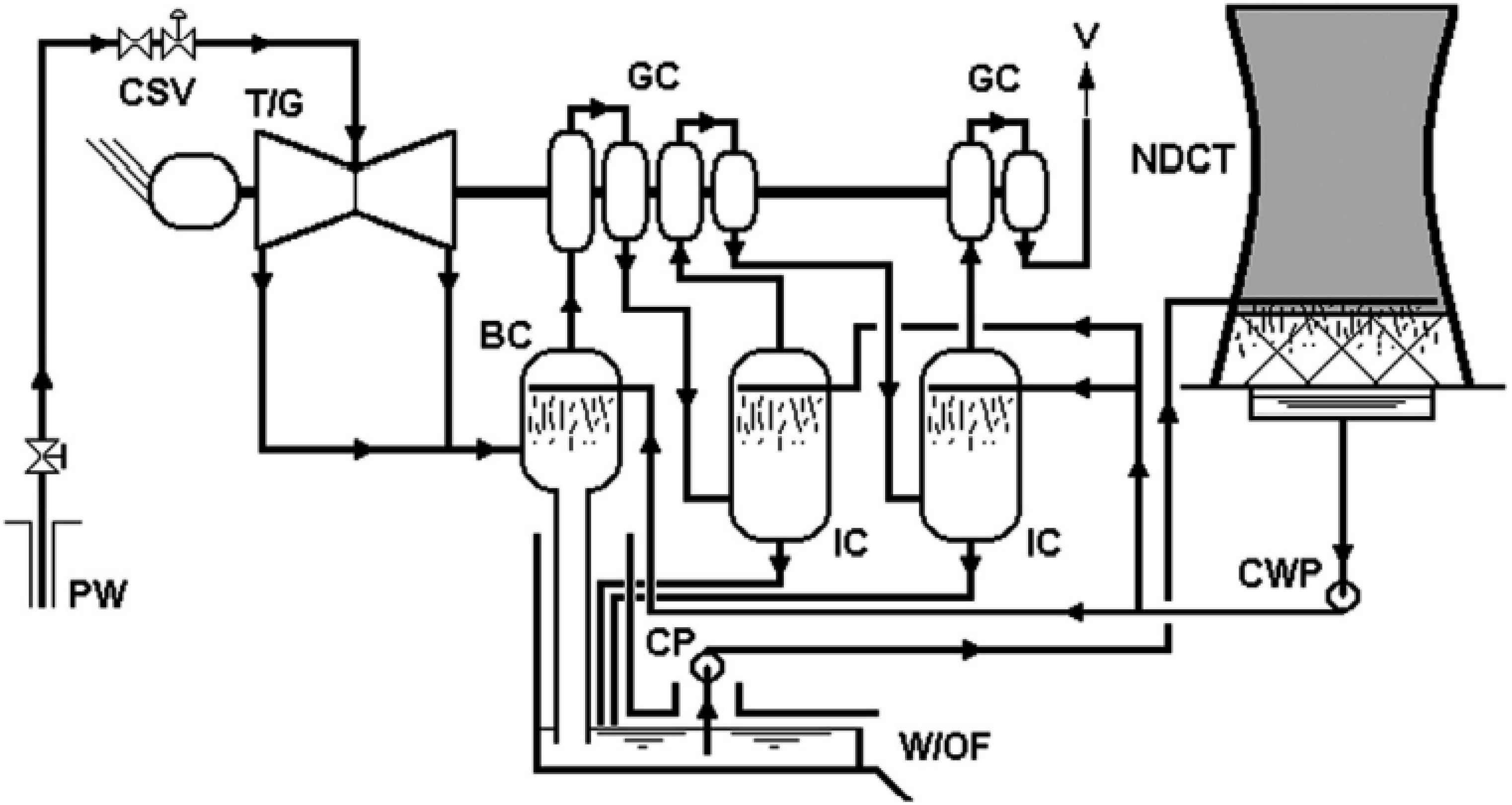


Figure 11.13 Larderello "Cycle 3" or direct-intake, condensing plant.

Dry steam (Larderello)

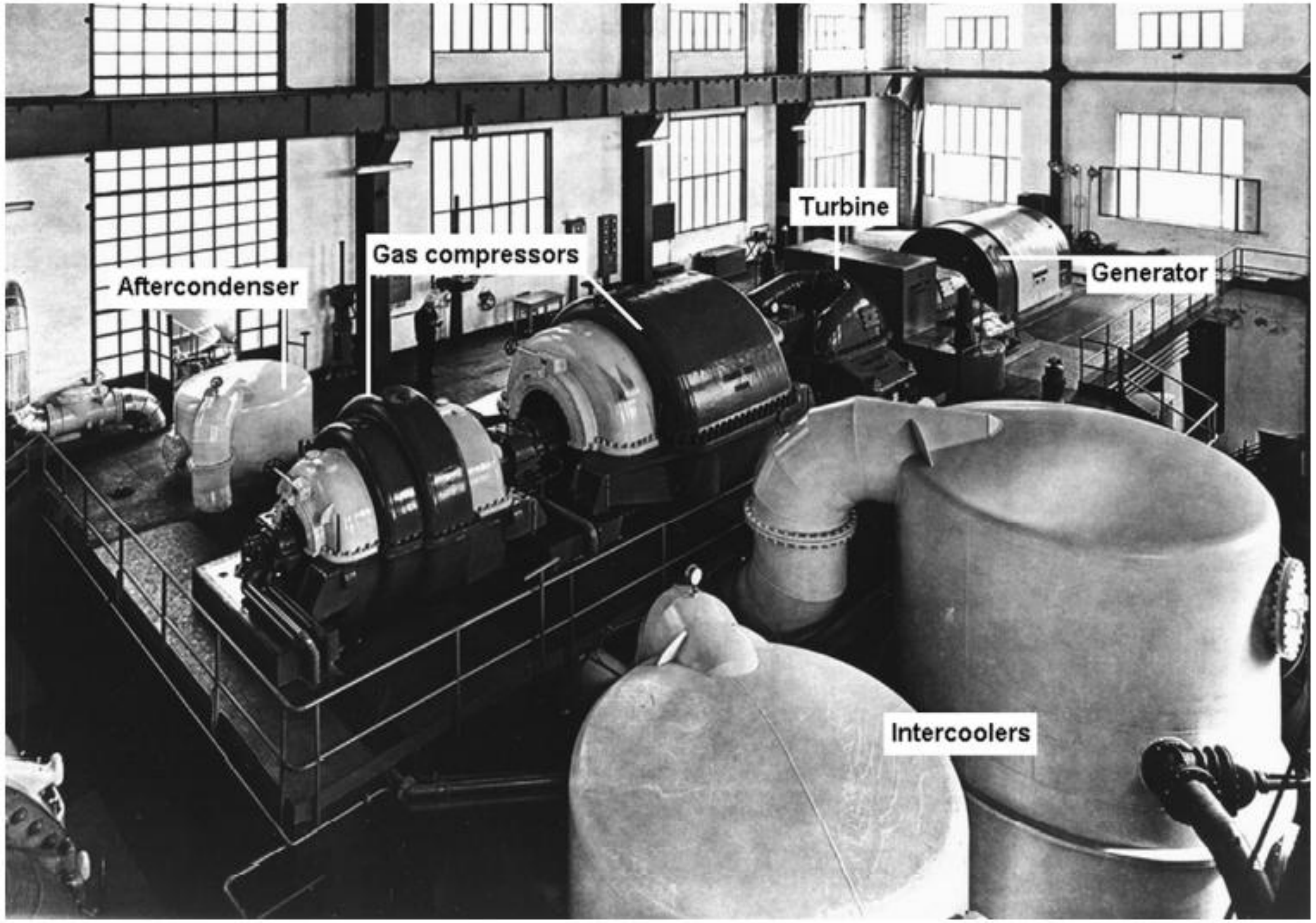


Figure 11.14 Castelnuovo turbine hall showing 26 MW turbine-generator, gas compressors, intercoolers, and aftercondenser [17, 18].

Dry steam (Larderello)

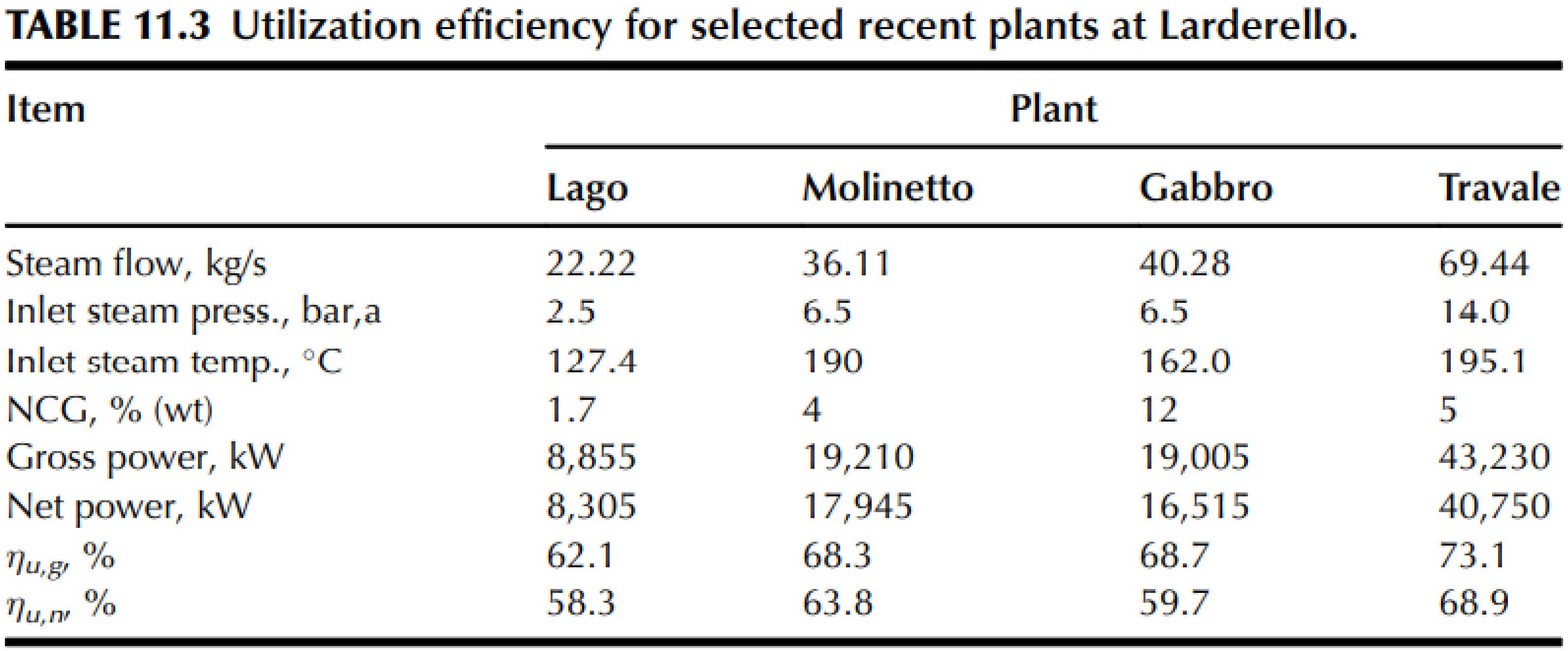


TABLE 11.3 Utilization efficiency for selected recent plants at Larderello.

Item	Plant			
	Lago	Molinetto	Gabbro	Travale
Steam flow, kg/s	22.22	36.11	40.28	69.44
Inlet steam press., bar,a	2.5	6.5	6.5	14.0
Inlet steam temp., °C	127.4	190	162.0	195.1
NCG, % (wt)	1.7	4	12	5
Gross power, kW	8,855	19,210	19,005	43,230
Net power, kW	8,305	17,945	16,515	40,750
$\eta_{u,g}$, %	62.1	68.3	68.7	73.1
$\eta_{u,n}$, %	58.3	63.8	59.7	68.9

Dry steam (Larderello)

Figure 11.17 Carboli 2 × 20 MW power units. *Photo courtesy of ENEL Green Power [WWW].*

Dry steam

The Geysers (California). Dry steam. 250ºC

20 units. Installed capacity 1400 MW

Recharge of the reservoir with treated municipal waste water.

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

Co-funded by the
Erasmus+ Programme
of the European Union

GEB

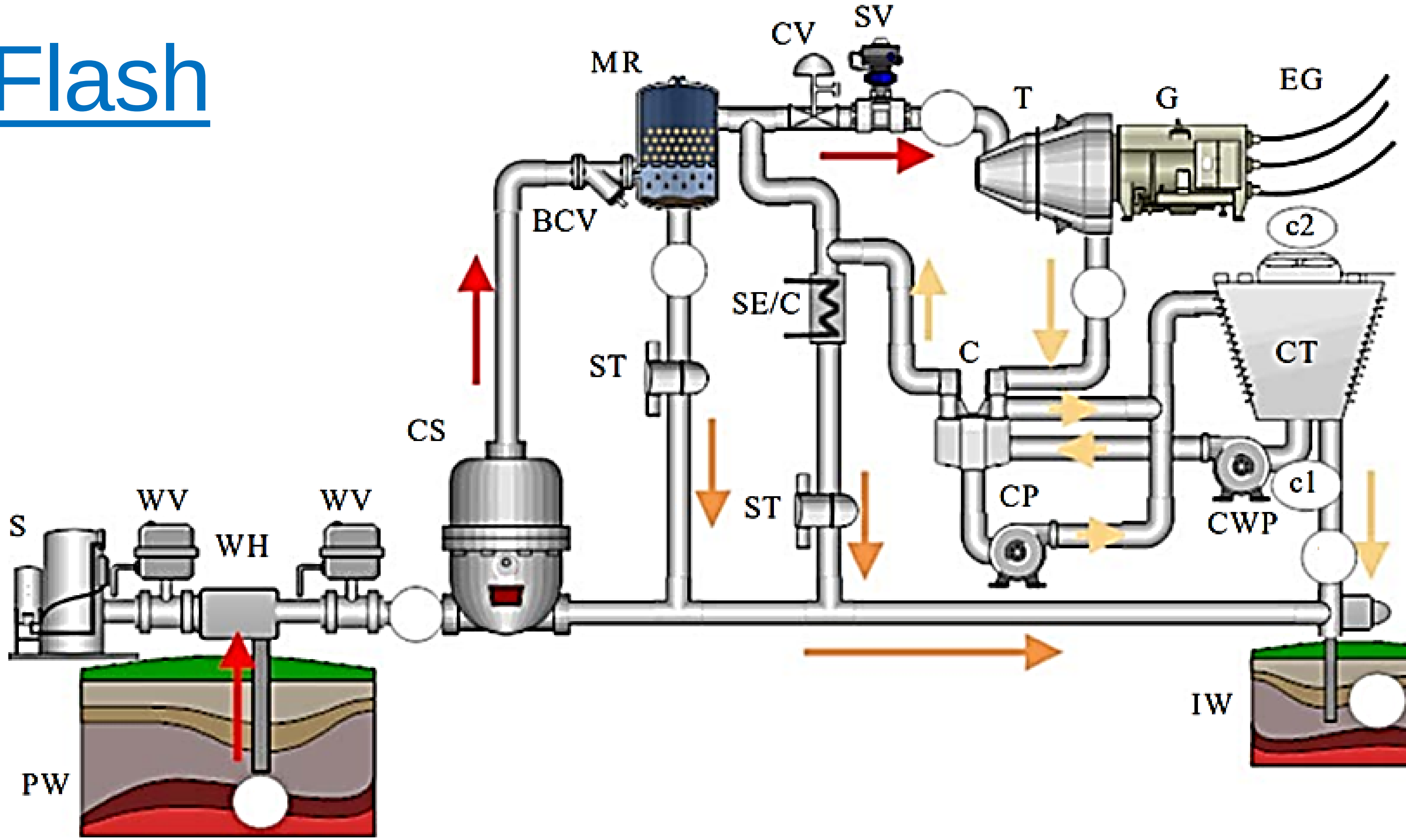
Geothermal Energy Capacity Building in Egypt (GEB)

Power generation from high-enthalpy geothermal resources: flash steam power plants

Energy and exergy analysis. Working parameters optimization

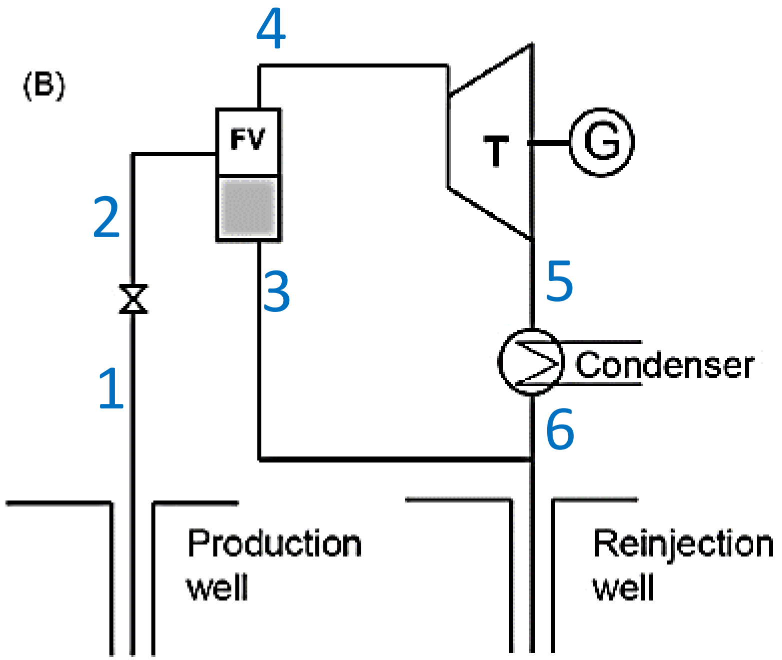
Co-funded by the
Erasmus+ Programme
of the European Union

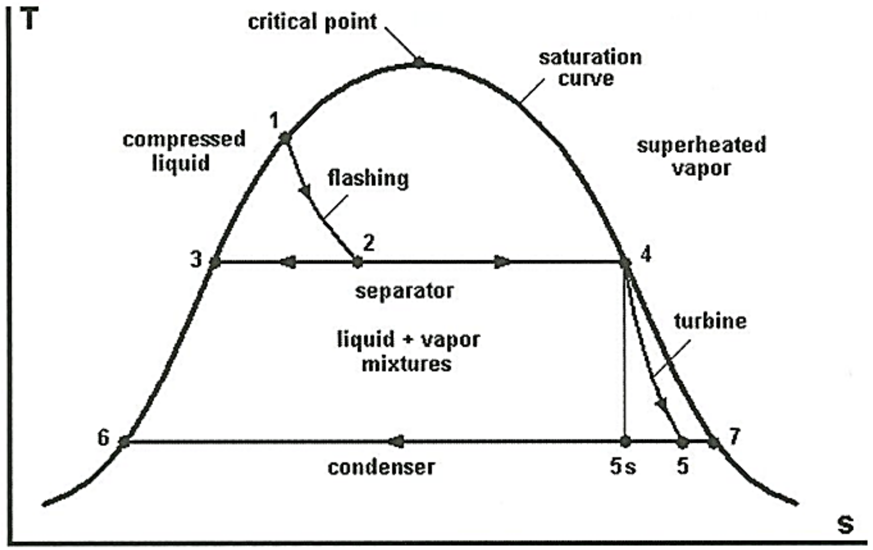
Single - Flash



PW	Production well	BCV	Ball check valve	SV	Stop valve
S	Silencer	MR	Moisture remover	SE	Steam jet ejectors
WV	Well valves	ST	Steam tramp	C	Condenser
CS	Cyclone separator	CV	Control valve	CP	Condensate pump
IW	Injection well	EG	Electric grid	CWP	Condensed water pump
G	Generator	T	Turbine	WH	Wellhead

Single - Flash





- Presure at Condenser is a function of the temperature of the cooling medium
- Pressure at the Separator (FV) is a **design parameter**

Single – Flash: Energy analysis

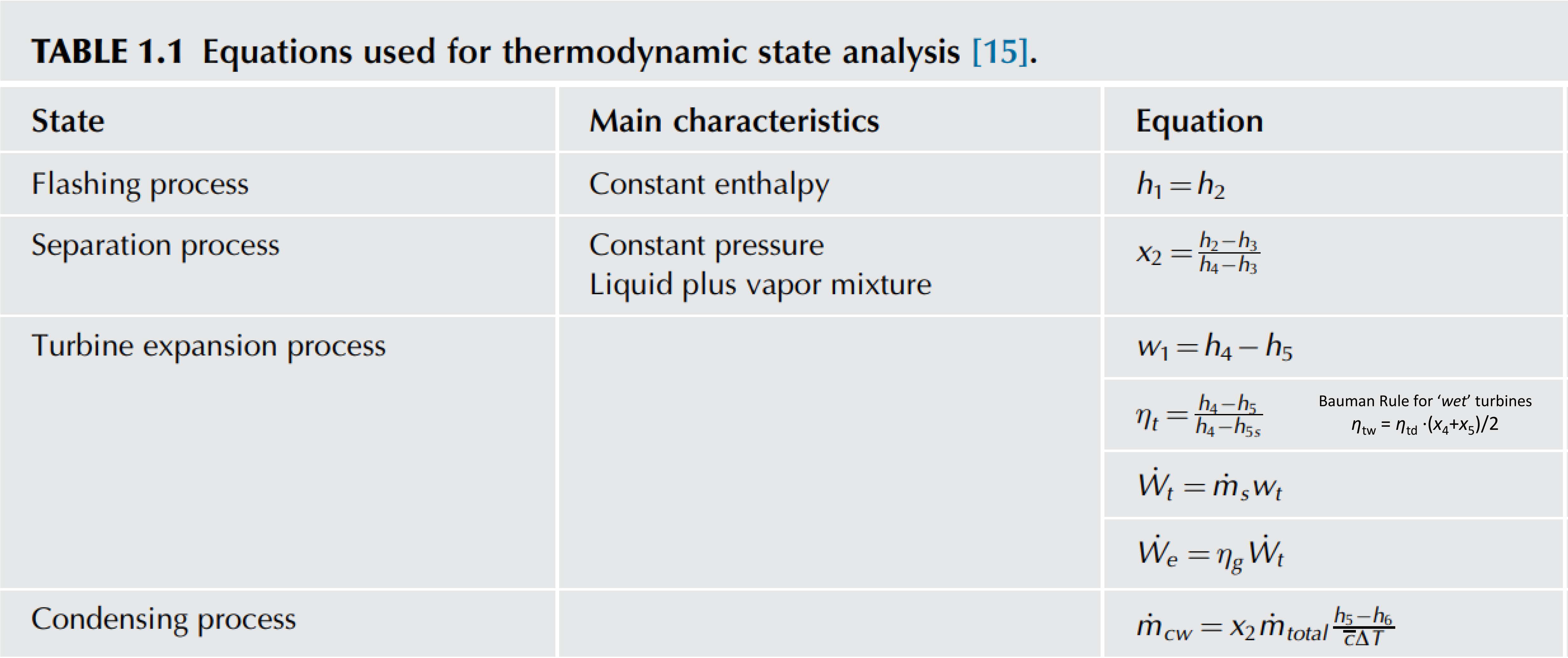


TABLE 1.1 Equations used for thermodynamic state analysis [15].

State	Main characteristics	Equation
Flashing process	Constant enthalpy	$h_1 = h_2$
Separation process	Constant pressure Liquid plus vapor mixture	$x_2 = \frac{h_2 - h_3}{h_4 - h_3}$
Turbine expansion process		$w_1 = h_4 - h_5$
		$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}}$ Bauman Rule for '*wet*' turbines $\eta_{tw} = \eta_{td} \cdot (x_4 + x_5)/2$
		$\dot{W}_t = \dot{m}_s w_t$
		$\dot{W}_e = \eta_g \dot{W}_t$
Condensing process		$\dot{m}_{cw} = x_2 \dot{m}_{total} \frac{h_5 - h_6}{\bar{c} \Delta T}$

Single – Flash: Exergy analysis

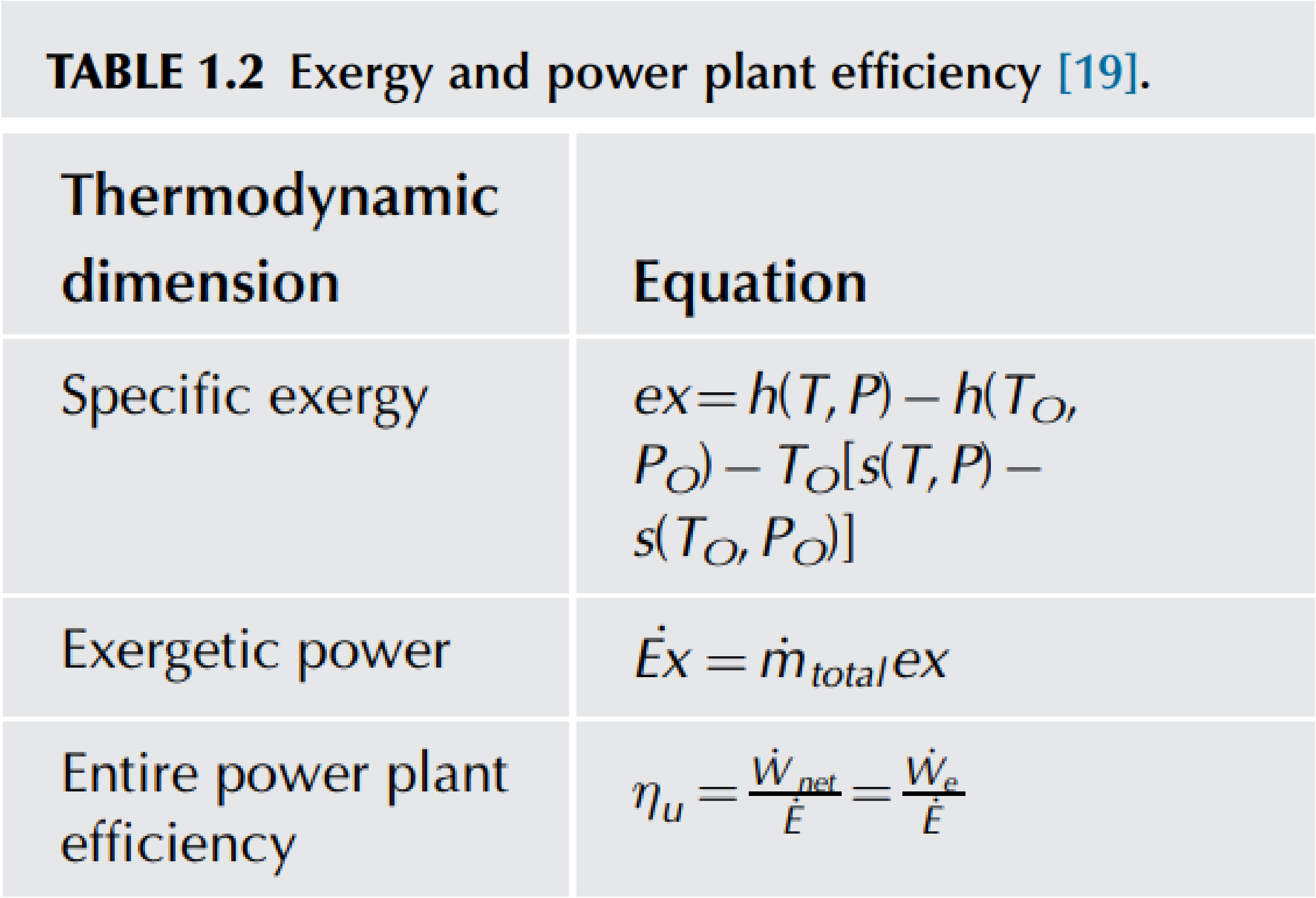


TABLE 1.2 Exergy and power plant efficiency [19].

Thermodynamic dimension	Equation
Specific exergy	$ex = h(T, P) - h(T_O, P_O) - T_O[s(T, P) - s(T_O, P_O)]$
Exergetic power	$\dot{E}x = \dot{m}_{total} ex$
Entire power plant efficiency	$\eta_u = \frac{\dot{W}_{net}}{\dot{E}} = \frac{\dot{W}_e}{\dot{E}}$

Second-Law Analysis

Turbine:

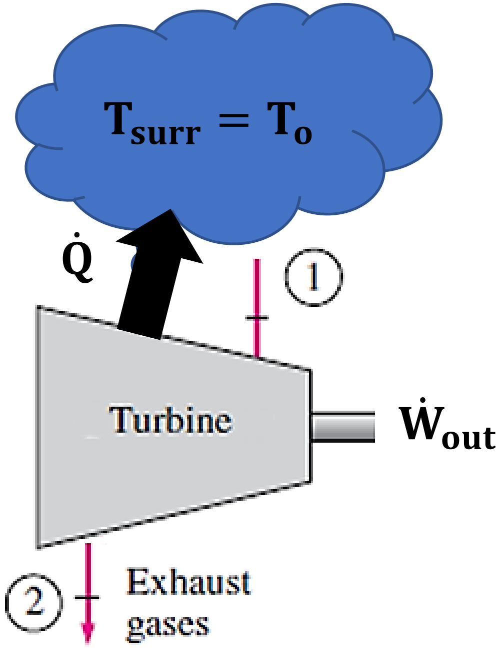
Exergy balance:

$$\dot{EX}_{in} - \dot{EX}_{out} - \dot{EX}_{loss} = \frac{dEX_{sys}}{dt} = 0\ (steady - state, steady - flow)$$

$$\dot{m}ex_1 - \dot{W}_{out} - \dot{m}ex_2 - \dot{EX}_{loss} = 0$$

$$\dot{EX}_{loss} = \dot{EX}_{supplied} - \dot{EX}_{recovered}$$

$$\dot{EX}_{loss} = \underbrace{\boxed{[\dot{m}(ex_1 - ex_2)]}}_{\dot{EX}_{supplied}} - \underbrace{\boxed{\dot{W}_{out}}}_{\dot{EX}_{recovered}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

The second-law efficiency:

$$\eta_{II,T} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

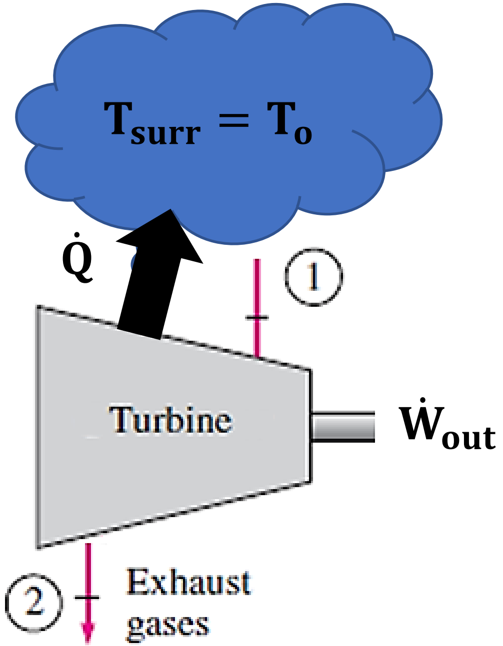
$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}(ex_1 - ex_2)}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}[(h_1 - h_2) - T_o(s_1 - s_2)]}$$

But the first law of thermodynamics requires:

$$\dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{W}_{out} + \dot{Q} - \dot{m}T_o(s_1 - s_2)} = \frac{\dot{W}_{out}}{\dot{W}_{rev}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0\ (\text{steady} - \text{state}, \text{steady} - \text{flow})$$

$$\dot{m}s_1 - \dot{m}s_2 - \left(\frac{\dot{Q}}{T_{surr}}\right) + \dot{S}_{gen} = 0$$

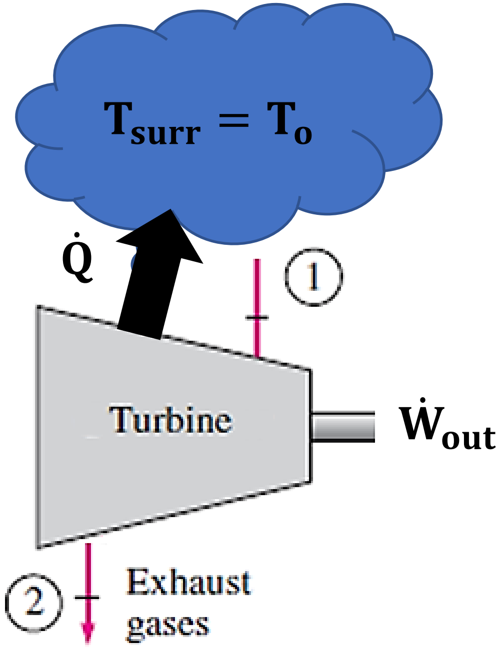
$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \left(\frac{\dot{Q}}{T_{surr}}\right) \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \left(\frac{T_o}{T_o}\right)\dot{Q} \geq 0$$

$$\text{But}: \dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \dot{m}(h_1 - h_2) - \dot{W}_{out} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = [\dot{m}(ex_1 - ex_2)] - \dot{W}_{out}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

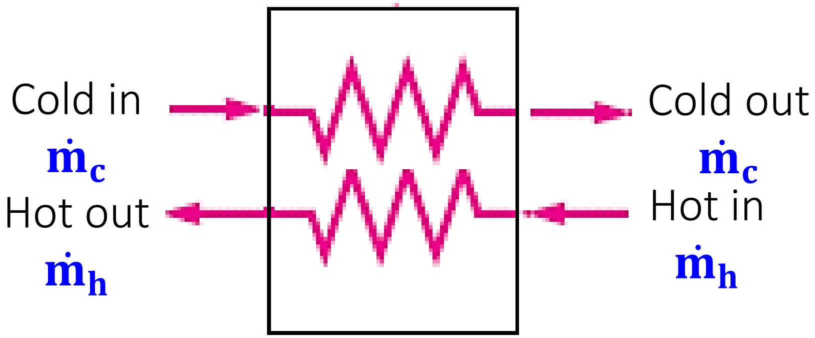
Exergy balance:

$$\dot{EX}_{in} - \dot{EX}_{out} - \dot{EX}_{loss} = \frac{dEX_{sys}}{dt} = 0 \; (steady - state, steady - flow)$$

$$\dot{m}_c ex_{c,in} + \dot{m}_h ex_{h,in} - \dot{m}_c ex_{c,out} - \dot{m}_h ex_{h,out} - \dot{EX}_{loss} = 0$$

$$\dot{EX}_{loss} = \dot{EX}_{supplied} - \dot{EX}_{recovered}$$

$$\dot{EX}_{loss} = \underbrace{\boxed{\dot{m}_h\left(ex_{h,in} - ex_{h,out}\right)}}_{\dot{EX}_{supplied}} - \underbrace{\boxed{\dot{m}_c\left(ex_{c,out} - ex_{c,in}\right)}}_{\dot{EX}_{recovered}}$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

The second-law efficiency:

$$\eta_{II,HX} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

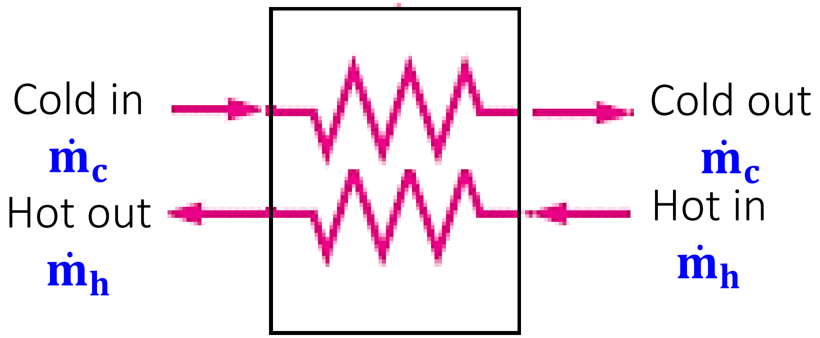
$$\eta_{II,HX} = \frac{\dot{m}_c\left(ex_{c,out} - ex_{c,in}\right)}{\dot{m}_h\left(ex_{h,in} - ex_{h,out}\right)}$$

$$\eta_{II,HX} = \frac{\dot{m}_c\left[\left(h_{c,out} - h_{c,in}\right) - T_o\left(s_{c,out} - s_{c,in}\right)\right]}{\dot{m}_h\left[\left(h_{h,in} - h_{h,out}\right) - T_o\left(s_{h,in} - s_{h,out}\right)\right]}$$

But the first law of thermodynamics requires:

$$\dot{m}_c\left(h_{c,out} - h_{c,in}\right) = \dot{m}_h\left(h_{h,in} - h_{h,out}\right)$$

$$\eta_{II,HX} = \frac{\dot{m}_h\left(h_{h,in} - h_{h,out}\right) - \dot{m}_c T_o\left(s_{c,out} - s_{c,in}\right)}{\dot{m}_h\left[\left(h_{h,in} - h_{h,out}\right) - T_o\left(s_{h,in} - s_{h,out}\right)\right]}$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady − state, steady − flow)}$$

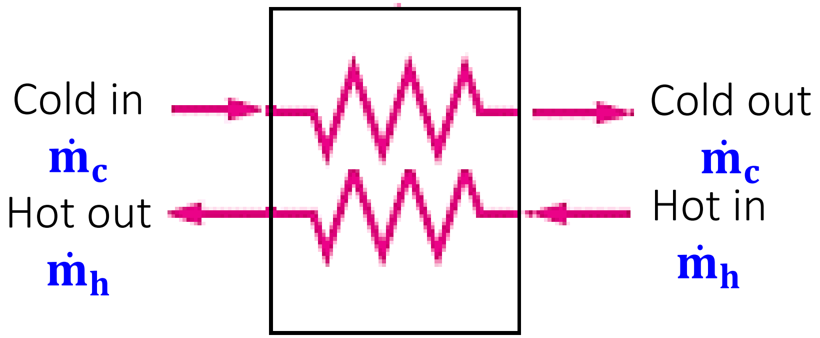
$$\dot{m}_c s_{c,in} + \dot{m}_h s_{h,in} - \dot{m}_c s_{c,out} - \dot{m}_h s_{h,out} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_c\left(s_{c,out} - s_{c,in}\right) - \dot{m}_h\left(s_{h,in} - s_{h,out}\right) \geq 0$$

$$\dot{m}_c\left(s_{c,out} - s_{c,in}\right) \geq \dot{m}_h\left(s_{h,in} - s_{h,out}\right)$$

$$\dot{EX}_{loss} = T_o \dot{S}_{gen}$$

$$\dot{EX}_{loss} = \dot{m}_c T_o\left(s_{c,out} - s_{c,in}\right) - \dot{m}_h T_o\left(s_{h,in} - s_{h,out}\right) \geq 0$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Single - Flash cycle:

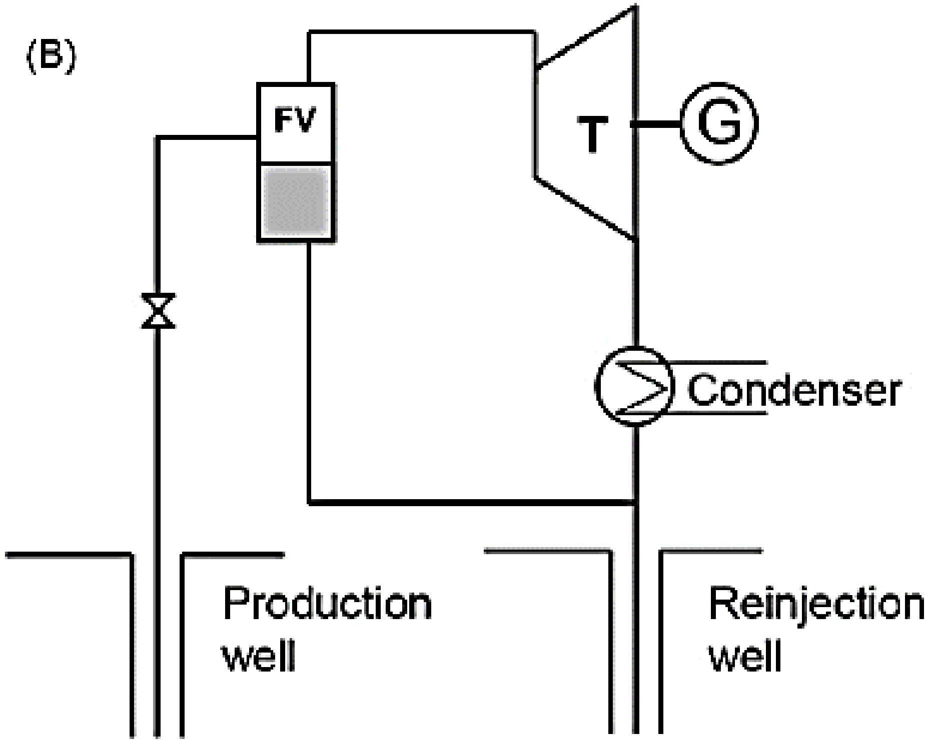
Exergy balance:

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{loss} = \frac{dX_{sys}}{dt} = 0 \; (steady - state, steady - flow)$$

$$\dot{m}ex_{production} - \dot{W}_{net} - \dot{m}ex_{reinjection} - \dot{X}_{loss} = 0$$

$$\dot{X}_{loss} = \dot{X}_{supplied} - \dot{X}_{recovered}$$

$$\dot{X}_{loss} = \underbrace{\boxed{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}}_{\dot{X}_{supplied}} - \underbrace{\boxed{\dot{W}_{net}}}_{\dot{X}_{recovered}}$$



Second-Law Analysis

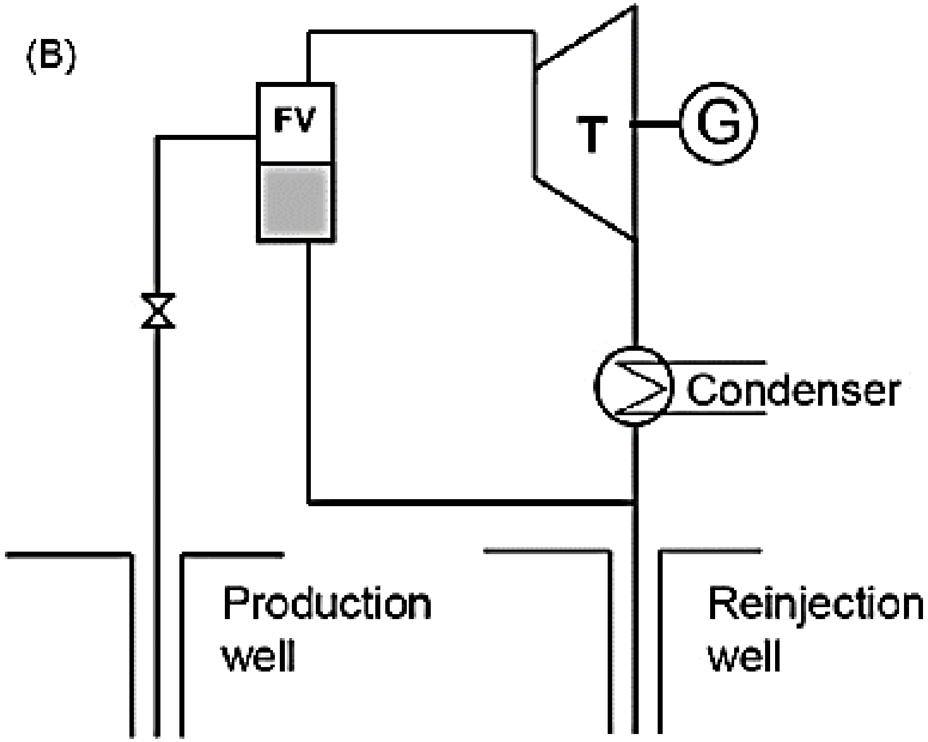
Single - Flash cycle:

The second-law efficiency:

$$\eta_{II,R} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}[\Delta h - T_o \Delta s]}$$



Second-Law Analysis

Single - Flash cycle:

Entropy balance:

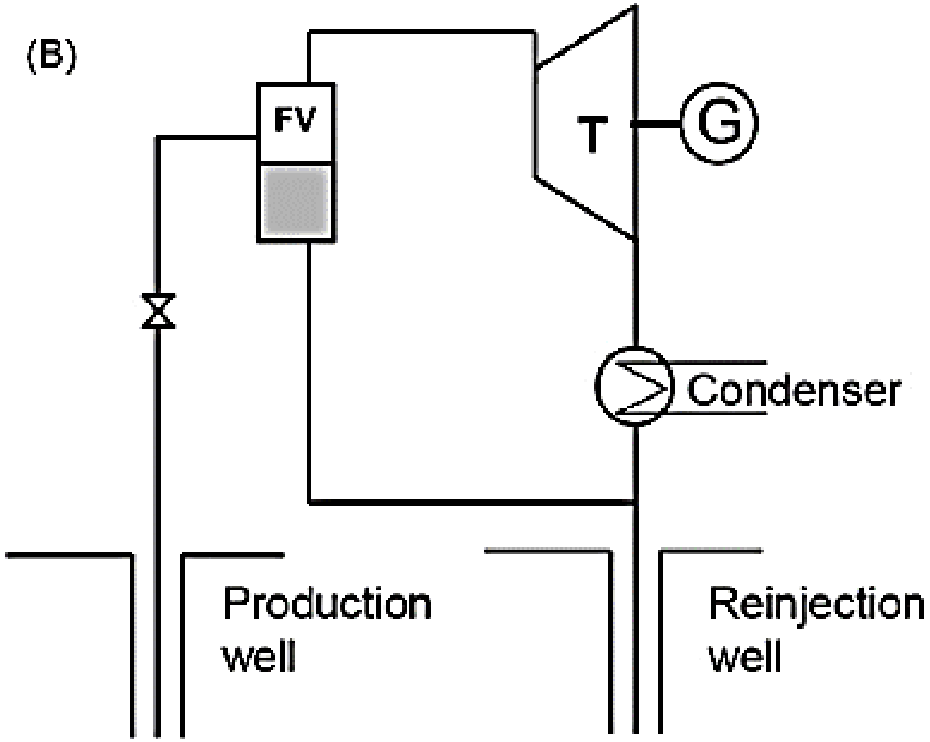
$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady − state, steady − flow)}$$

$$\dot{m}s_{production} - \dot{m}s_{reinjection} - \frac{\dot{Q}_L}{T_L} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}\left(s_{reinjection} - s_{production}\right) + \frac{\dot{Q}_L}{T_L} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o\left(s_{reinjection} - s_{production}\right) + \dot{Q}_L$$

Where $T_o = T_L$



Problem 1

A **single-flash** geothermal power plant operates from a reservoir that can provide 100 kg/s of saturated liquid at 230ºC.

Estimate:

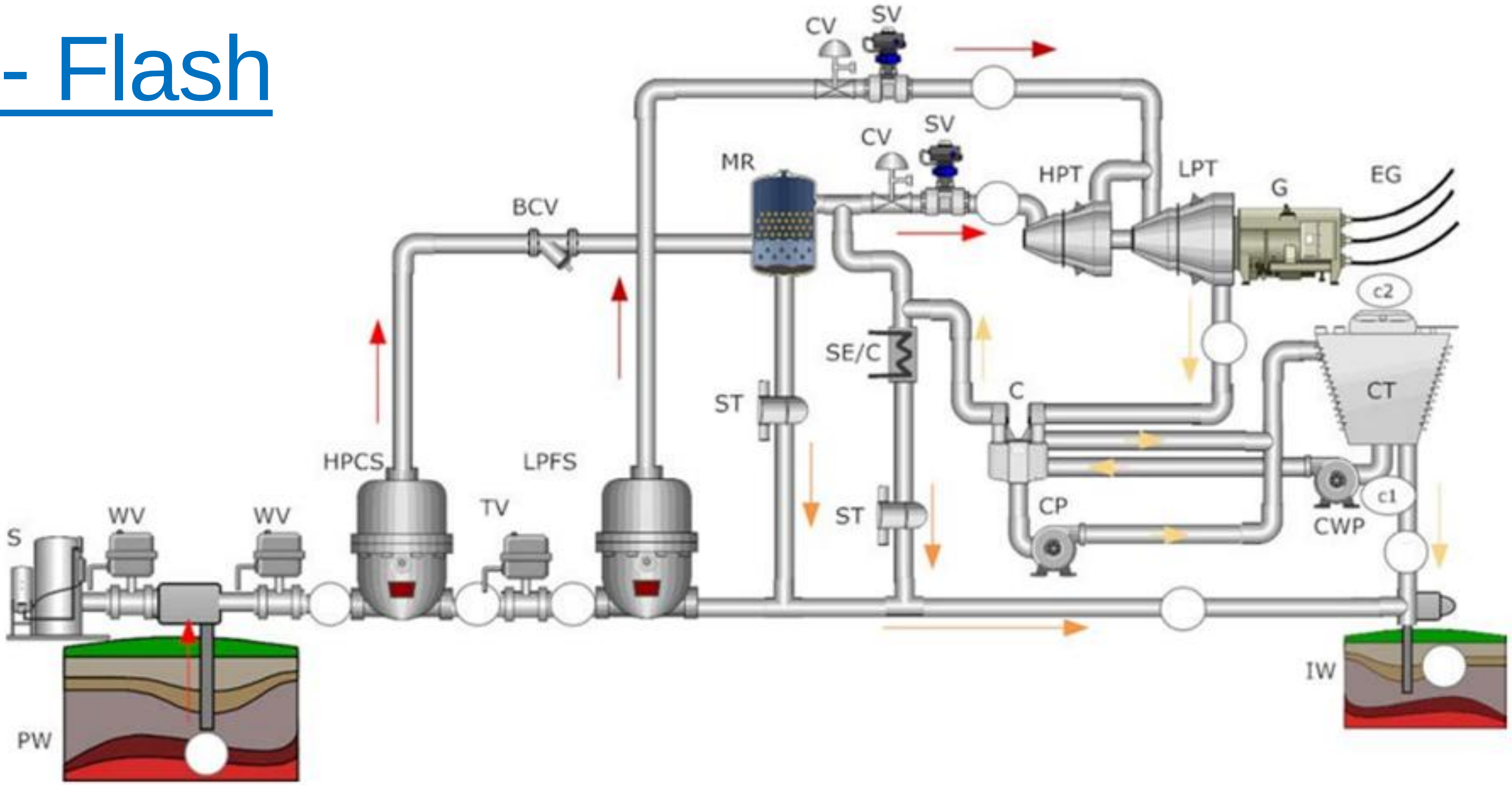
- **Optimal pressure** (optimal temperature) at the flash vessel to obtain maximum power
- **Net power** for this optimal pressure and **exergy efficiency**
- **Net annual energy production** (utilization factor: 0.90)

Assumptions.

- The condenser temperature is 50 ºC
- Neglect auxiliary energy consumptions
- Turbine mechanical efficiency: 100 %. Electric generator efficiency: 100 %
- Turbine isentropic efficiency: 0.85
- Environmental conditions: 15 ºC, 1 bar

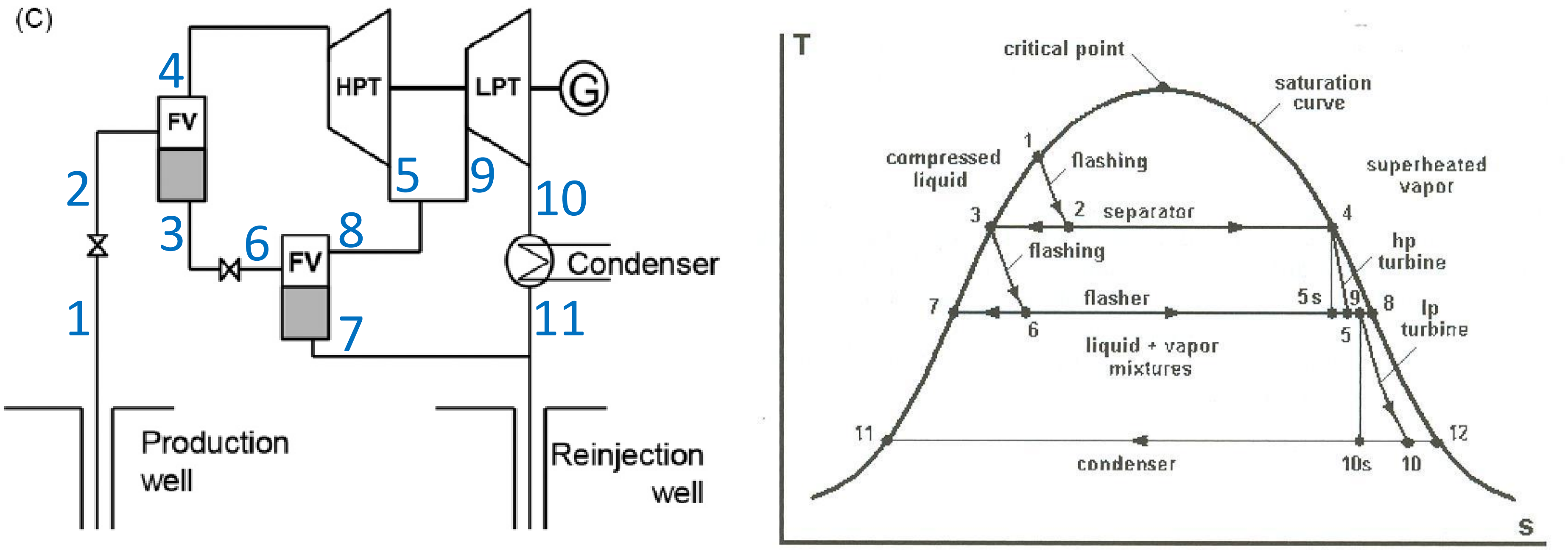
Use Solver Add-in in Excel, with Refprop linked for water thermodynamic properties.

Double - Flash



PW	Production well	SV	Stop valve	MR	Moisture remover
S	Silencer	SE	Steam jet ejectors	ST	Steam tramp
WV	Well valves	C	Condenser	CV	Control valve
BCV	Ball check valve	CP	Condensate pump	TV	Throttle valve
IW	Injection well	HPT	High-pressure turbine	G	Generator
CWP	Condensed water pump	LPT	Low-pressure turbine	EG	Electric grid
HPCS	High-pressure cyclone separator	LPFS	Low-pressure flash separator	WH	Wellhead

Double - Flash



- Presure at Condenser is a function of the temperature of the cooling medium
- Pressure at the Separators (FV) is a **design parameter**

Double – Flash: Energy analysis

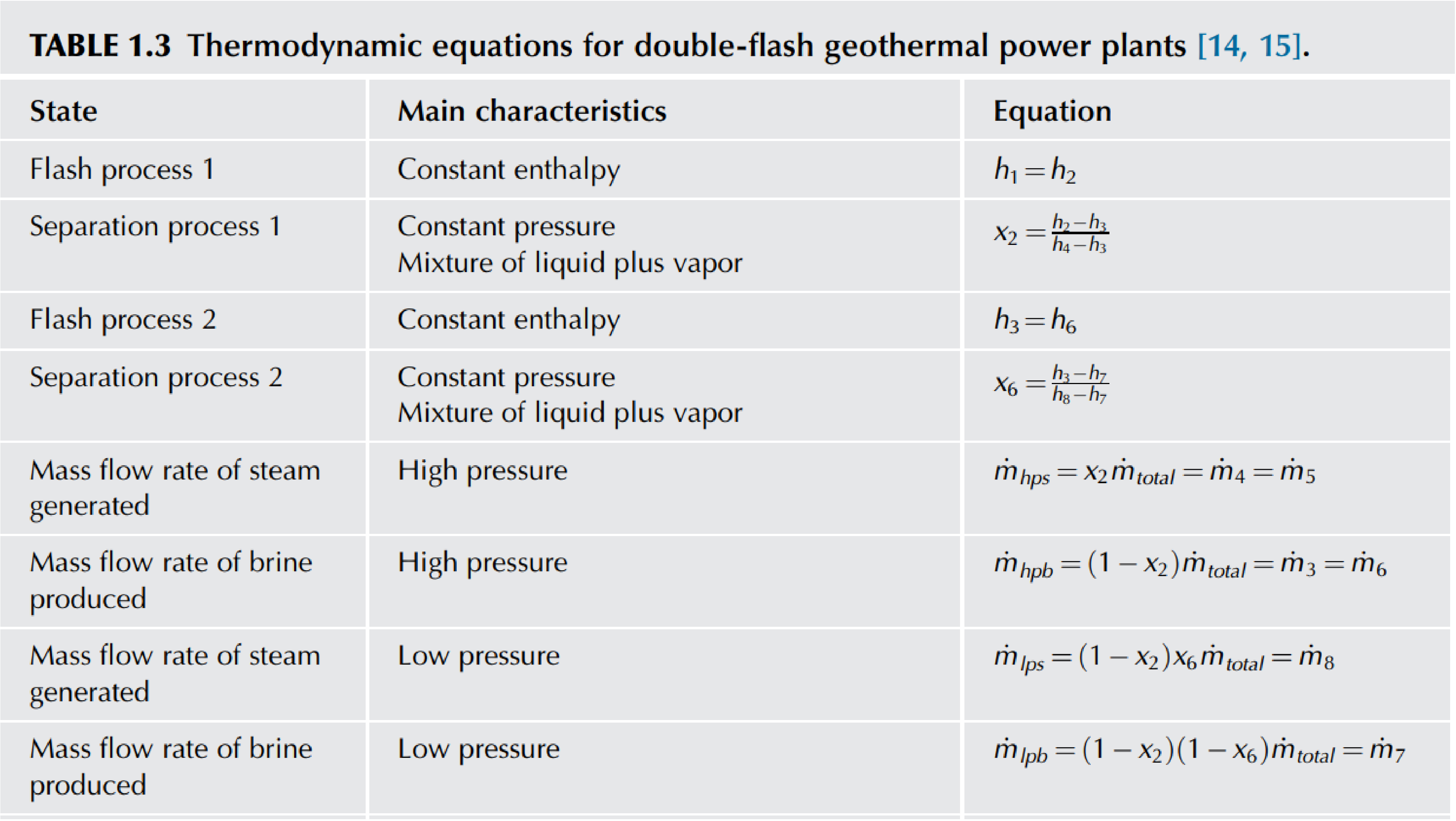


TABLE 1.3 Thermodynamic equations for double-flash geothermal power plants [14, 15].

State	Main characteristics	Equation
Flash process 1	Constant enthalpy	$h_1 = h_2$
Separation process 1	Constant pressure Mixture of liquid plus vapor	$x_2 = \frac{h_2 - h_3}{h_4 - h_3}$
Flash process 2	Constant enthalpy	$h_3 = h_6$
Separation process 2	Constant pressure Mixture of liquid plus vapor	$x_6 = \frac{h_3 - h_7}{h_8 - h_7}$
Mass flow rate of steam generated	High pressure	$\dot{m}_{hps} = x_2 \dot{m}_{total} = \dot{m}_4 = \dot{m}_5$
Mass flow rate of brine produced	High pressure	$\dot{m}_{hpb} = (1 - x_2) \dot{m}_{total} = \dot{m}_3 = \dot{m}_6$
Mass flow rate of steam generated	Low pressure	$\dot{m}_{lps} = (1 - x_2) x_6 \dot{m}_{total} = \dot{m}_8$
Mass flow rate of brine produced	Low pressure	$\dot{m}_{lpb} = (1 - x_2)(1 - x_6) \dot{m}_{total} = \dot{m}_7$

Double – Flash: Energy analysis

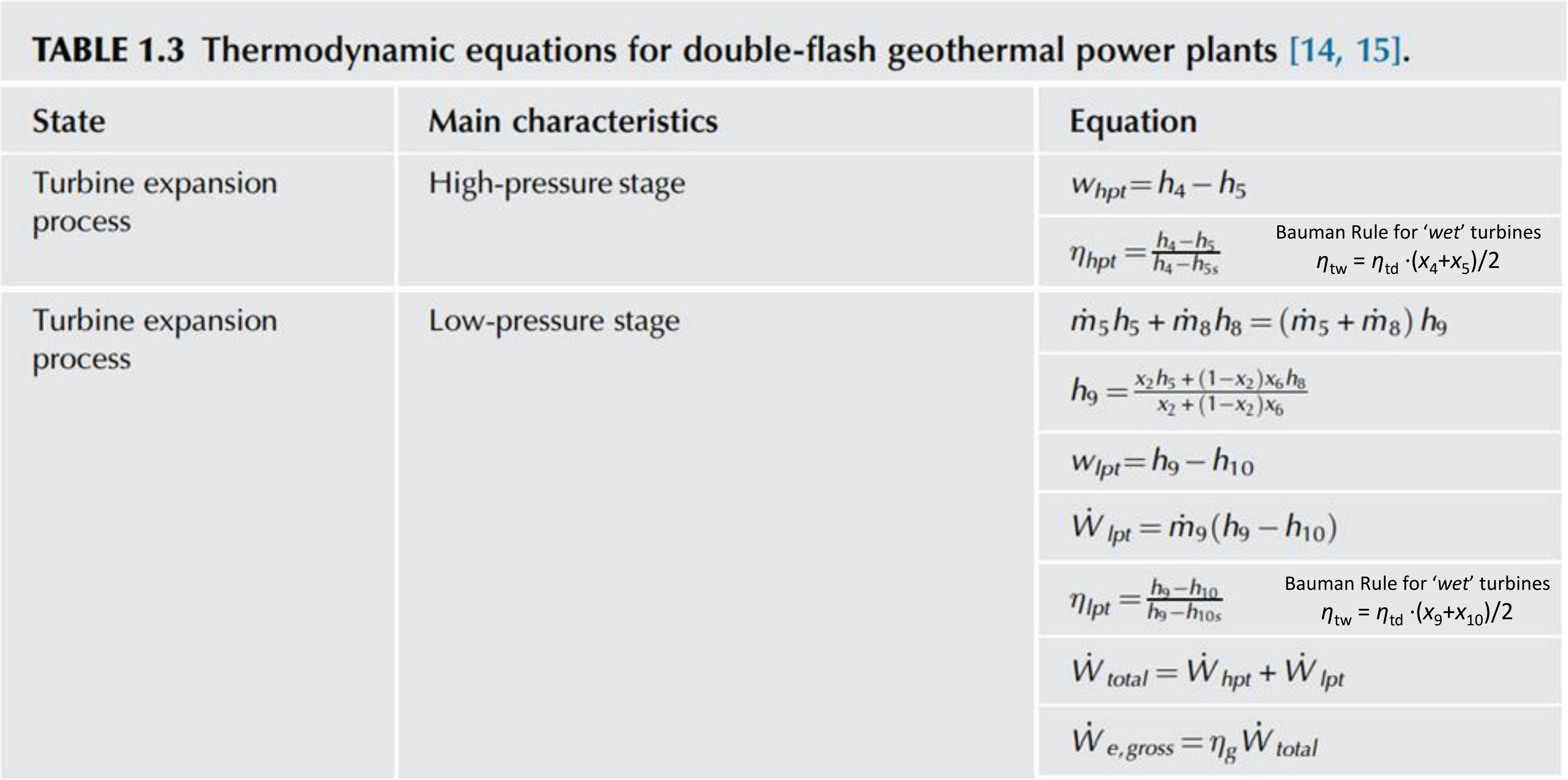


TABLE 1.3 Thermodynamic equations for double-flash geothermal power plants [14, 15].

State	Main characteristics	Equation
Turbine expansion process	High-pressure stage	$w_{hpt} = h_4 - h_5$
		$\eta_{hpt} = \frac{h_4 - h_5}{h_4 - h_{5s}}$ Bauman Rule for '*wet*' turbines $\eta_{tw} = \eta_{td} \cdot (x_4 + x_5)/2$
Turbine expansion process	Low-pressure stage	$\dot{m}_5 h_5 + \dot{m}_8 h_8 = (\dot{m}_5 + \dot{m}_8)\, h_9$
		$h_9 = \frac{x_2 h_5 + (1 - x_2) x_6 h_8}{x_2 + (1 - x_2) x_6}$
		$w_{lpt} = h_9 - h_{10}$
		$\dot{W}_{lpt} = \dot{m}_9 (h_9 - h_{10})$
		$\eta_{lpt} = \frac{h_9 - h_{10}}{h_9 - h_{10s}}$ Bauman Rule for '*wet*' turbines $\eta_{tw} = \eta_{td} \cdot (x_9 + x_{10})/2$
		$\dot{W}_{total} = \dot{W}_{hpt} + \dot{W}_{lpt}$
		$\dot{W}_{e,gross} = \eta_g \dot{W}_{total}$

Double – Flash: Exergy analysis

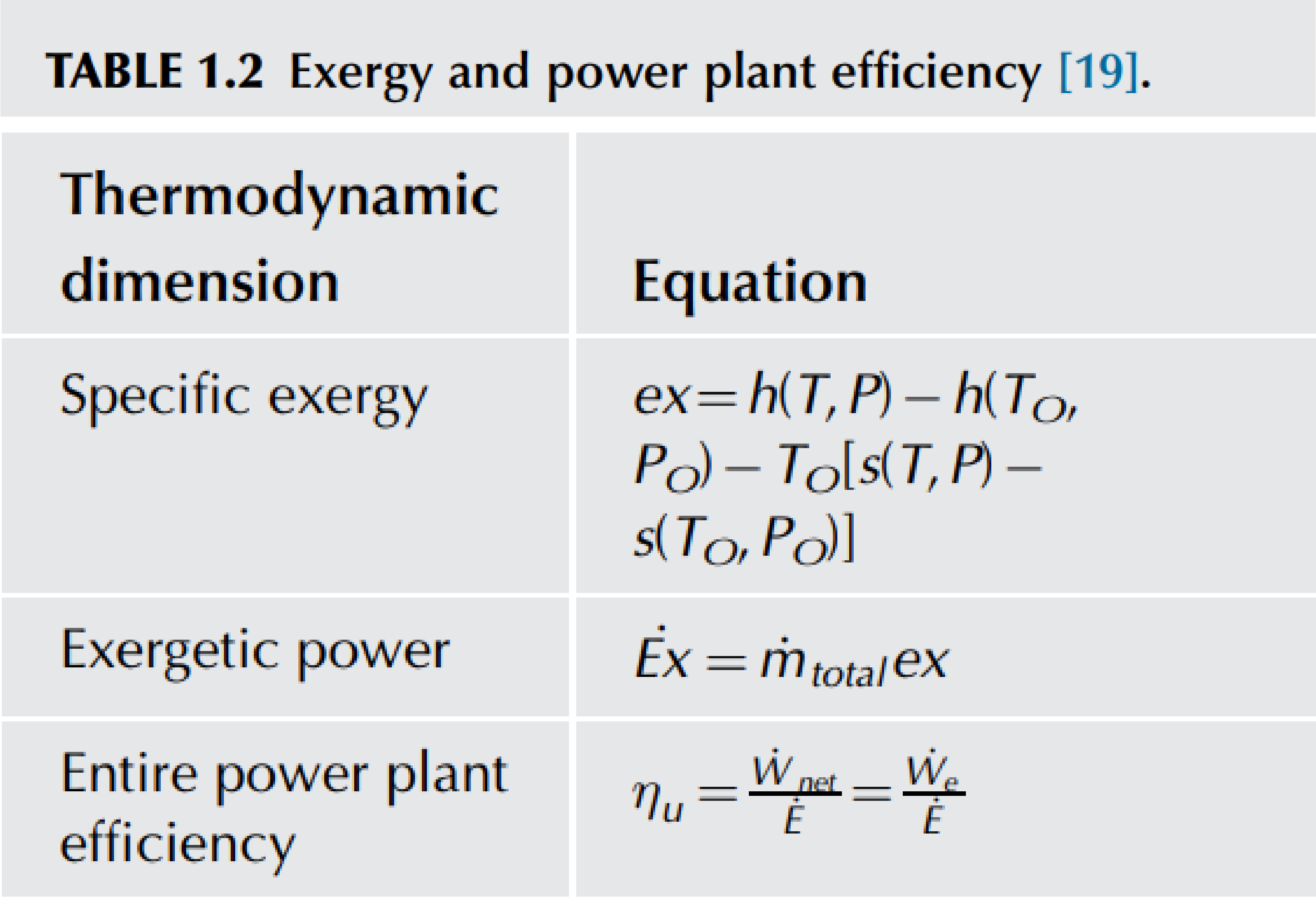


TABLE 1.2 Exergy and power plant efficiency [19].

Thermodynamic dimension	Equation
Specific exergy	$ex = h(T, P) - h(T_O, P_O) - T_O[s(T, P) - s(T_O, P_O)]$
Exergetic power	$\dot{E}x = \dot{m}_{total} ex$
Entire power plant efficiency	$\eta_u = \frac{\dot{W}_{net}}{\dot{E}} = \frac{\dot{W}_e}{\dot{E}}$

Problem 2

A **double-flash** geothermal power plant operates from a reservoir that can provide 100 kg/s of saturated liquid at 230ºC.

Estimate:
- **Optimal pressures** (optimal temperatures) at the **two flash vessels** to obtain maximum power
- **Net power** for this optimal pressure and **exergy efficiency**
- **Net annual energy production** (utilization factor: 0.90)

Assumptions.
- The condenser temperature is 50 ºC
- Neglect auxiliary energy consumptions
- Turbine mechanical efficiency: 100 %. Electric generator efficiency: 100 %
- Turbine isentropic efficiency: 0.85

Use Solver Add-in in Excel, with Refprop linked for water thermodynamic properties.

Double - Flash

Krafla I y II (Iceland)
Double - Flash. 30 MW each.

Environmental Impact (Dry steam and Flash)

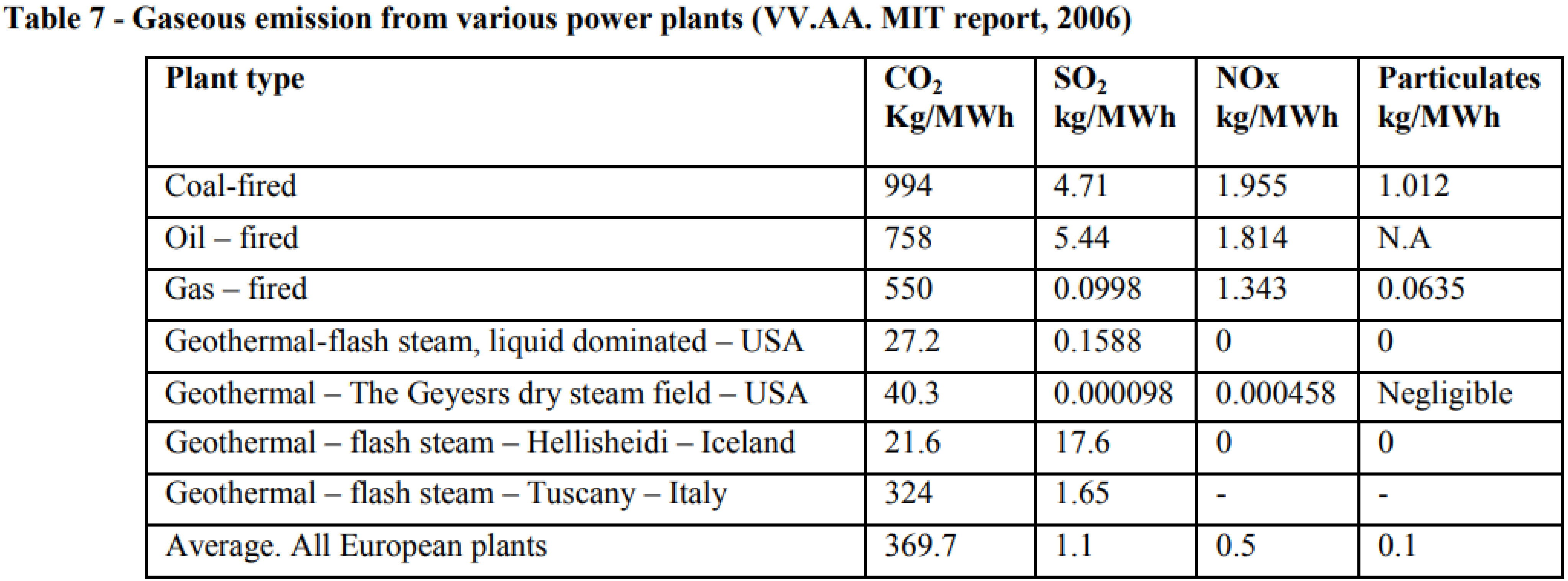


Table 7 - Gaseous emission from various power plants (VV.AA. MIT report, 2006)

Plant type	**CO_2 Kg/MWh**	**SO_2 kg/MWh**	**NOx kg/MWh**	**Particulates kg/MWh**
Coal-fired	994	4.71	1.955	1.012
Oil – fired	758	5.44	1.814	N.A
Gas – fired	550	0.0998	1.343	0.0635
Geothermal-flash steam, liquid dominated – USA	27.2	0.1588	0	0
Geothermal – The Geyesrs dry steam field – USA	40.3	0.000098	0.000458	Negligible
Geothermal – flash steam – Hellisheidi – Iceland	21.6	17.6	0	0
Geothermal – flash steam – Tuscany – Italy	324	1.65	-	-
Average. All European plants	369.7	1.1	0.5	0.1

https://www.geothermal-energy.org/pdf/IGAstandard/WGC/2015/37008.pdf

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

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Geothermal Energy Capacity Building in Egypt (GEB)

Advanced geothermal energy conversion systems
Hybrid geothermal power systems

Co-funded by the
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of the European Union

Hybrid Fossil – Geothermal Systems

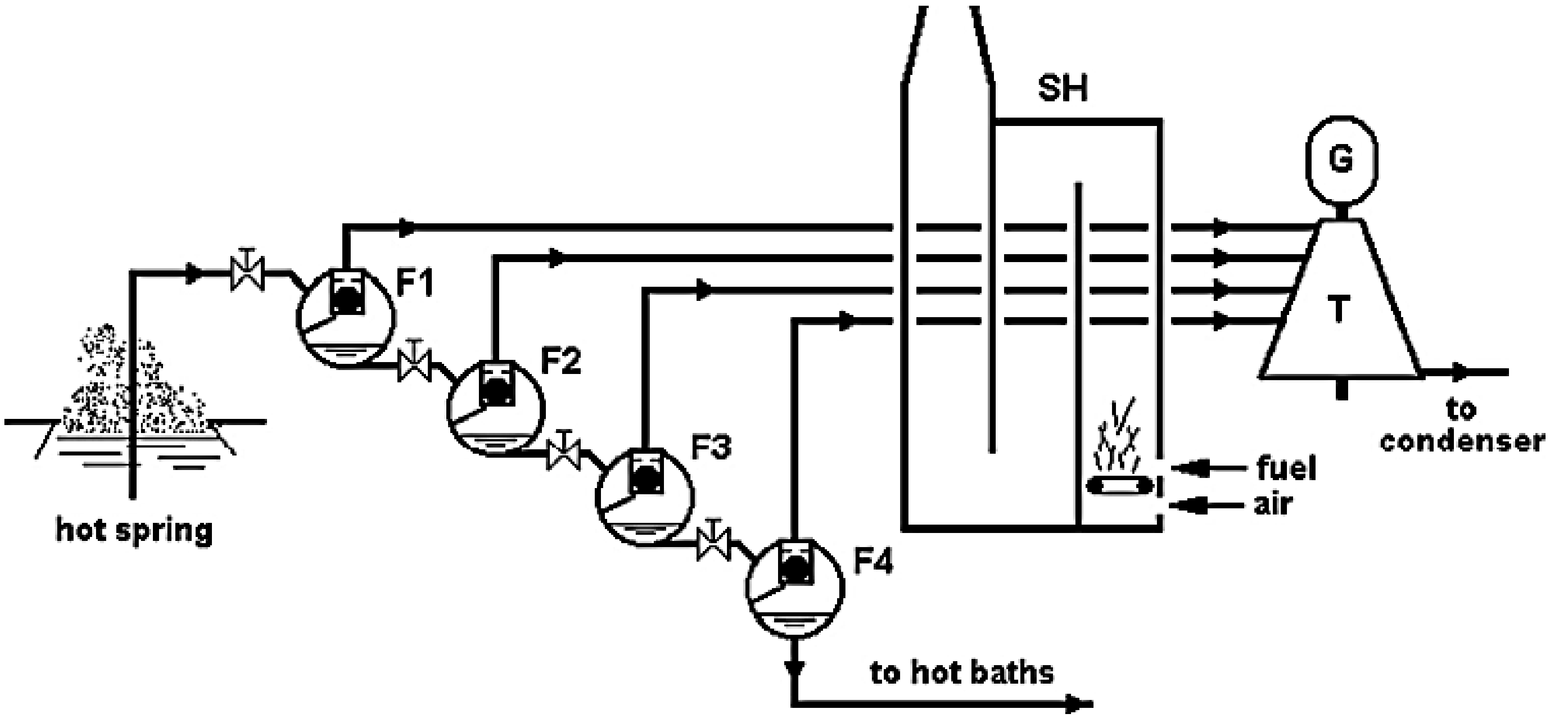

Figure 9.19 Caufourier's four-stage flash plant with fossil superheating. *After Ref. [19]*.

Solar-Augmented Flash Plants

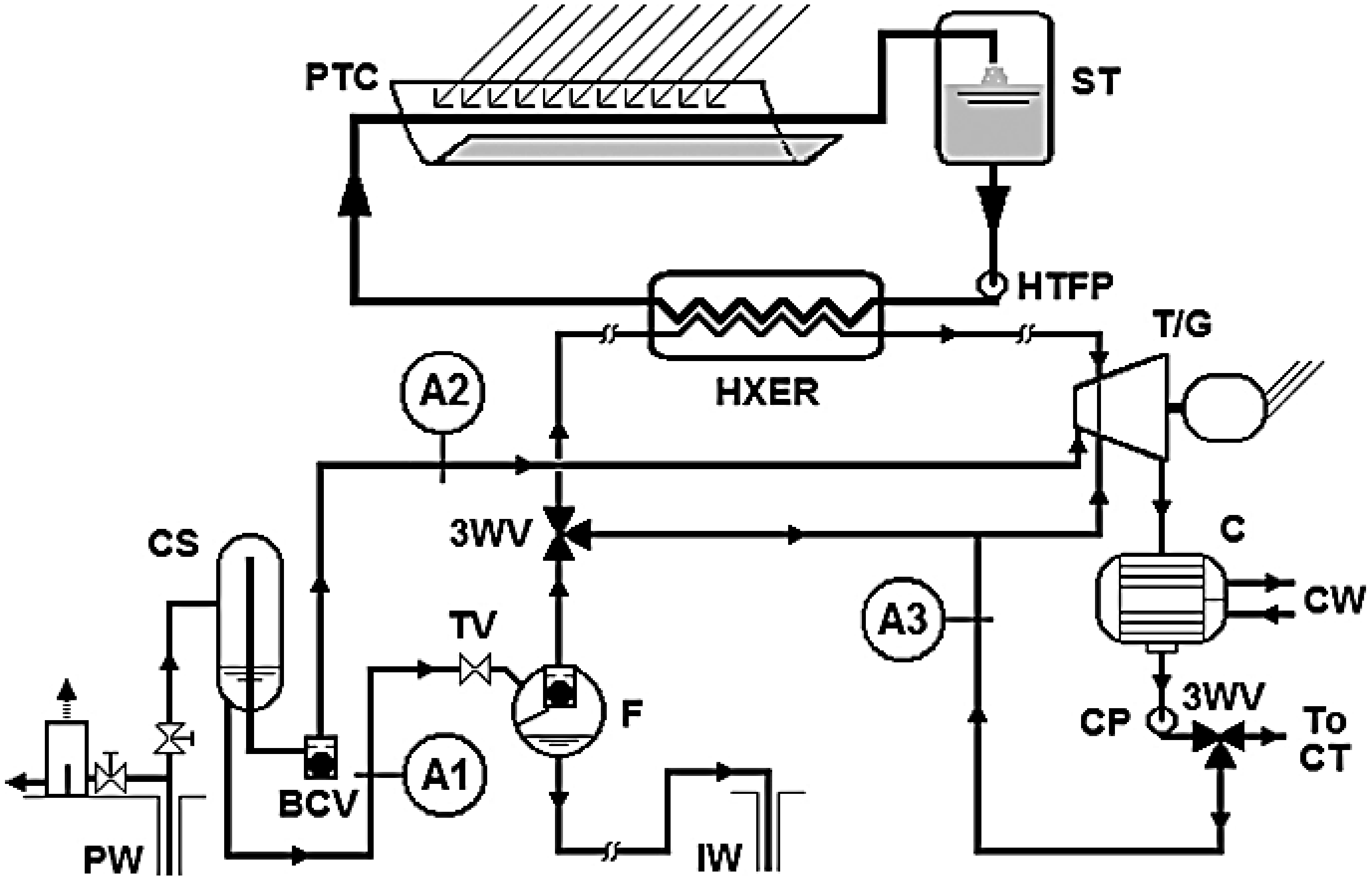

Figure 9.27 Solar-geothermal double-flash plant.

Geopressured Systems

Bigger depths
Very high pressures (up to 1000 bar)
Temperature (150ºC – 200ºC)
High salinity
Natural gas reservoirs

Use

Water pressure (hydraulic turbine)
Water termal energy (district heating)
Natural gas (combined cycle)

Hybrid Plant using a Geopressured System

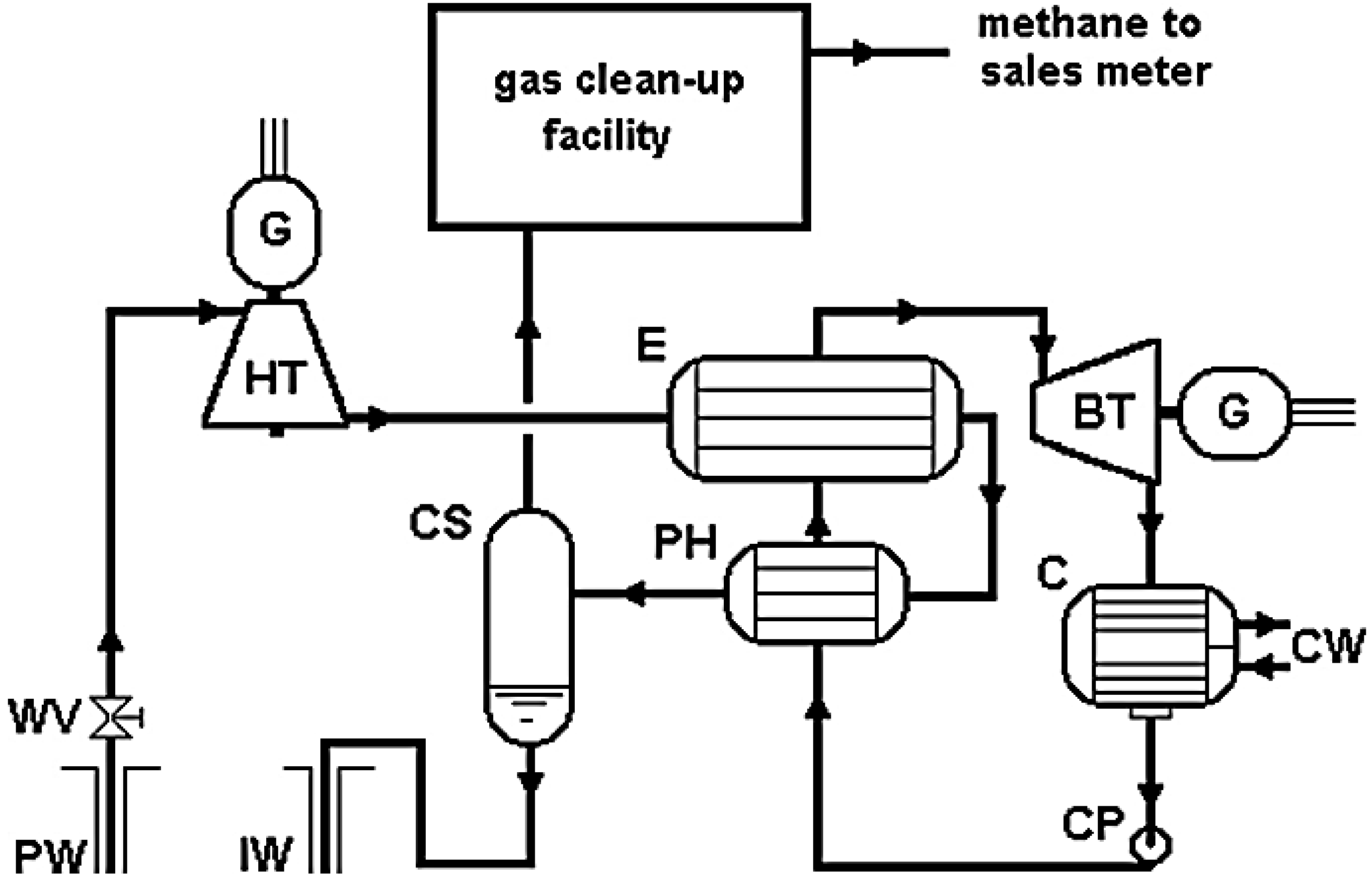

Figure 9.20 Conceptual hybrid plant utilizing a geopressured geothermal resource.

Combined Heat and Power Plants

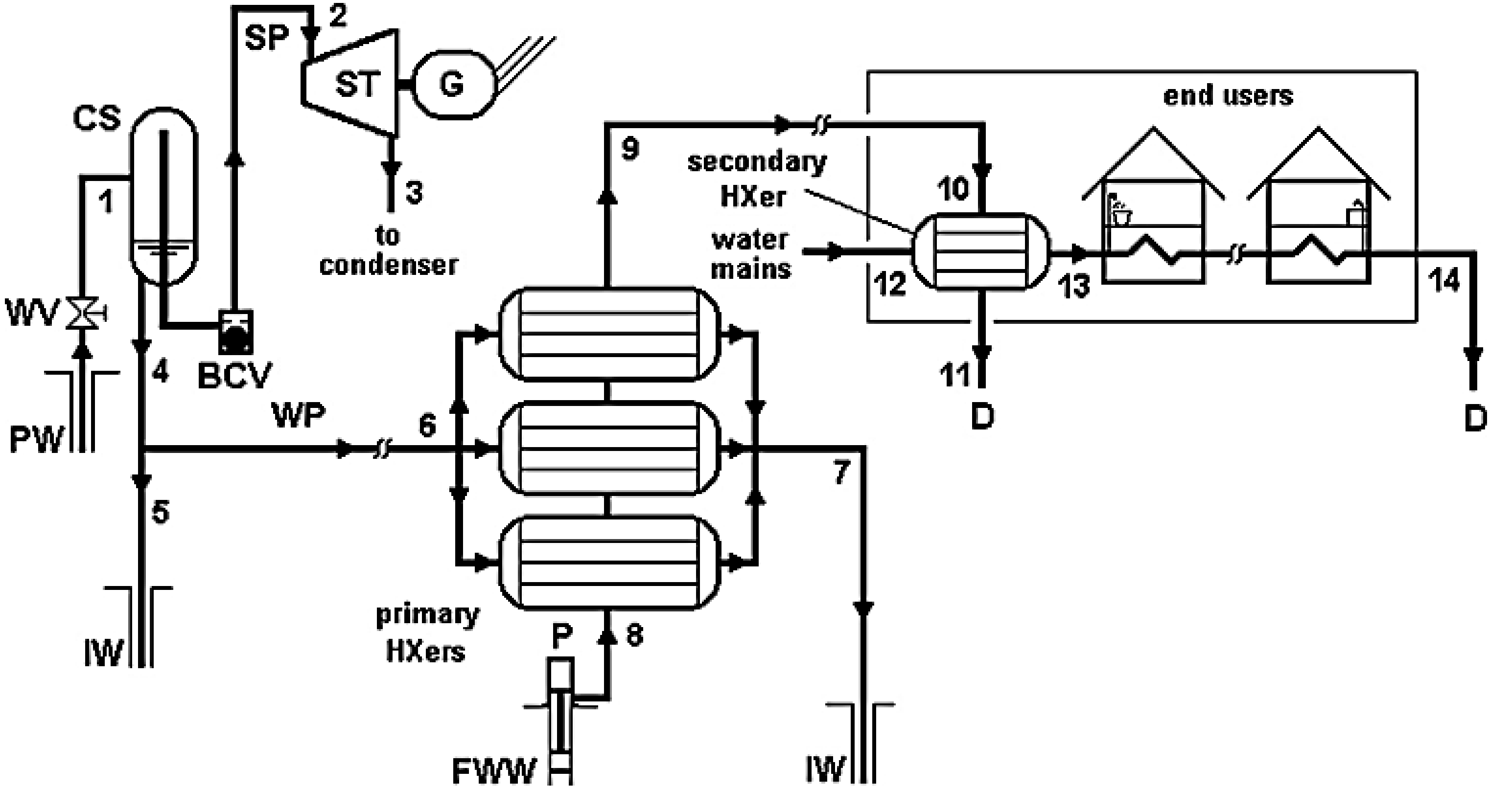

Figure 9.21 Combined geothermal heat and power plant. *After Ref. [30].*

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

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GEB

Geothermal Energy Capacity Building in Egypt (GEB)

Power generation from low-enthalpy geothermal resources: Binary cycle power plants

Energy and exergy analysis

Co-funded by the
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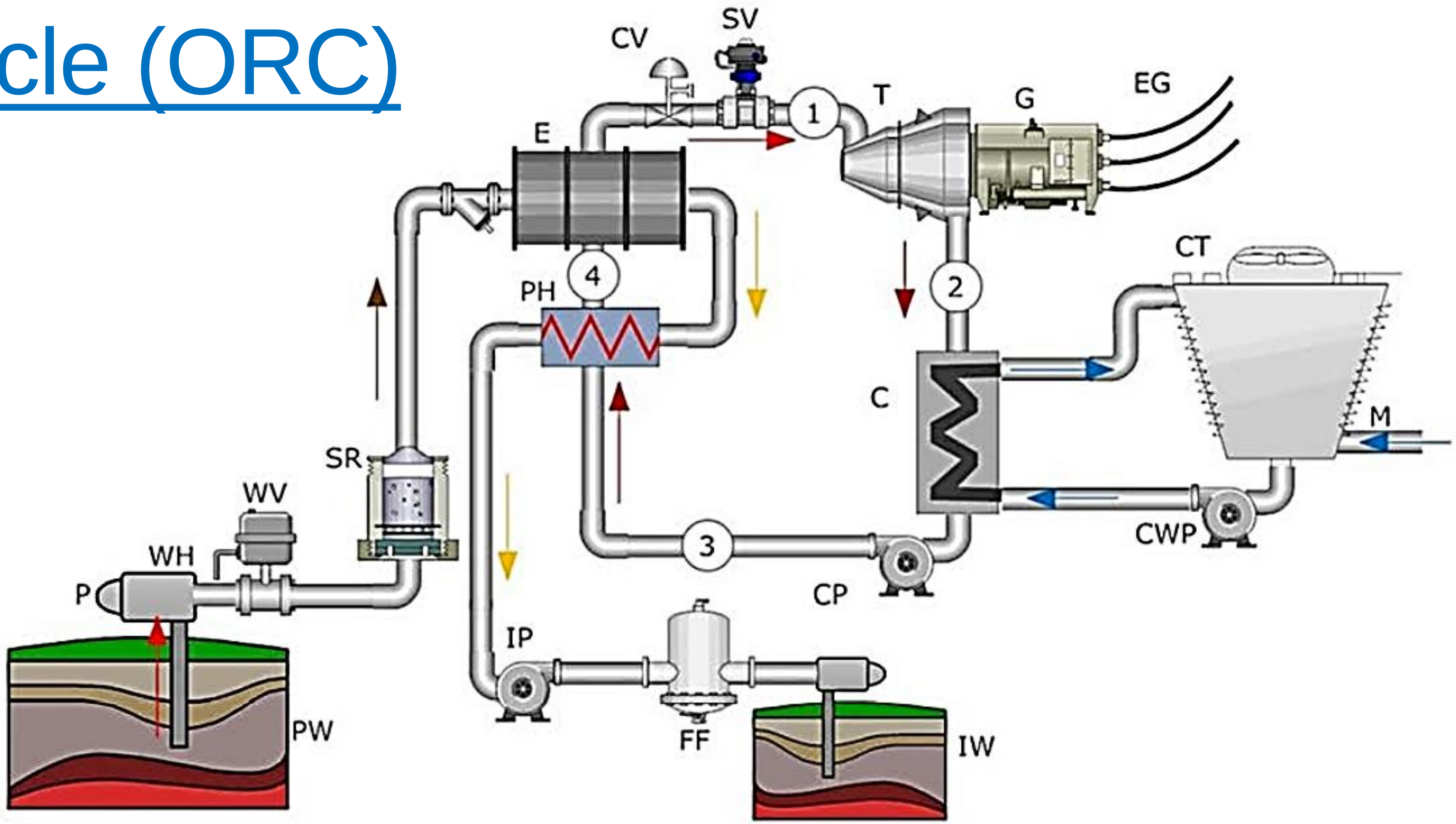
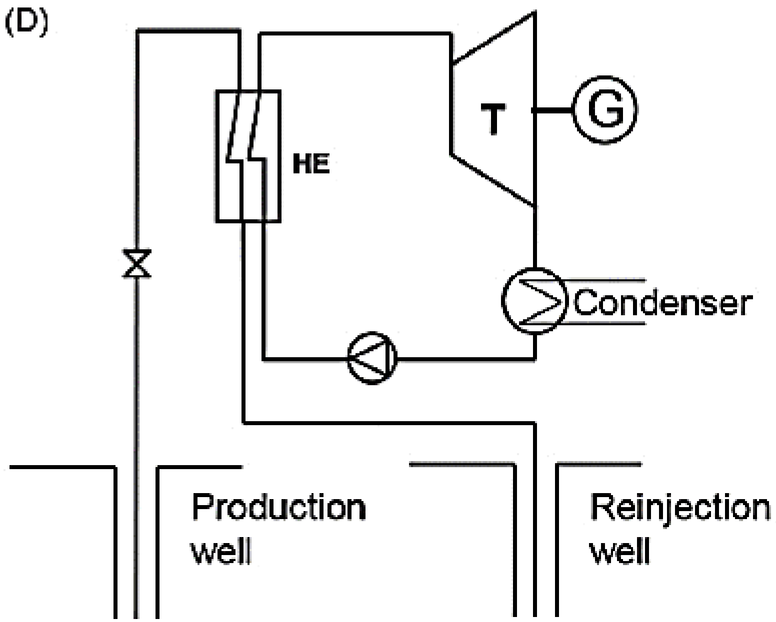
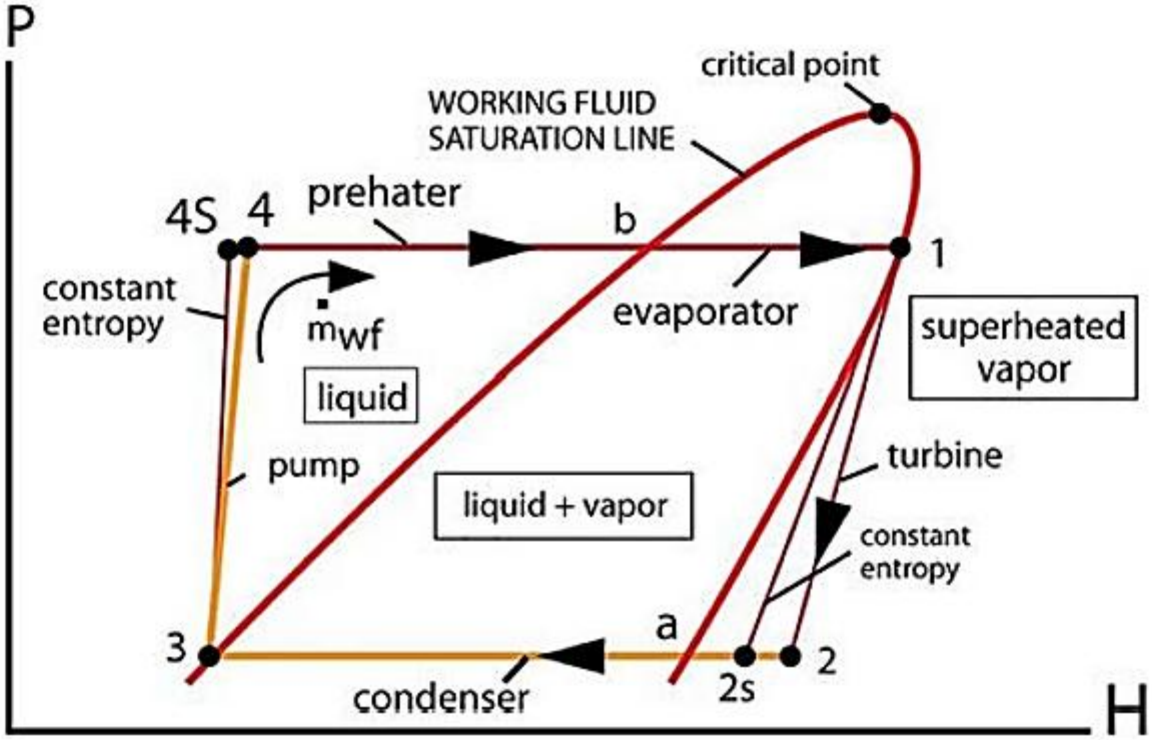
Binary Cycle (ORC)



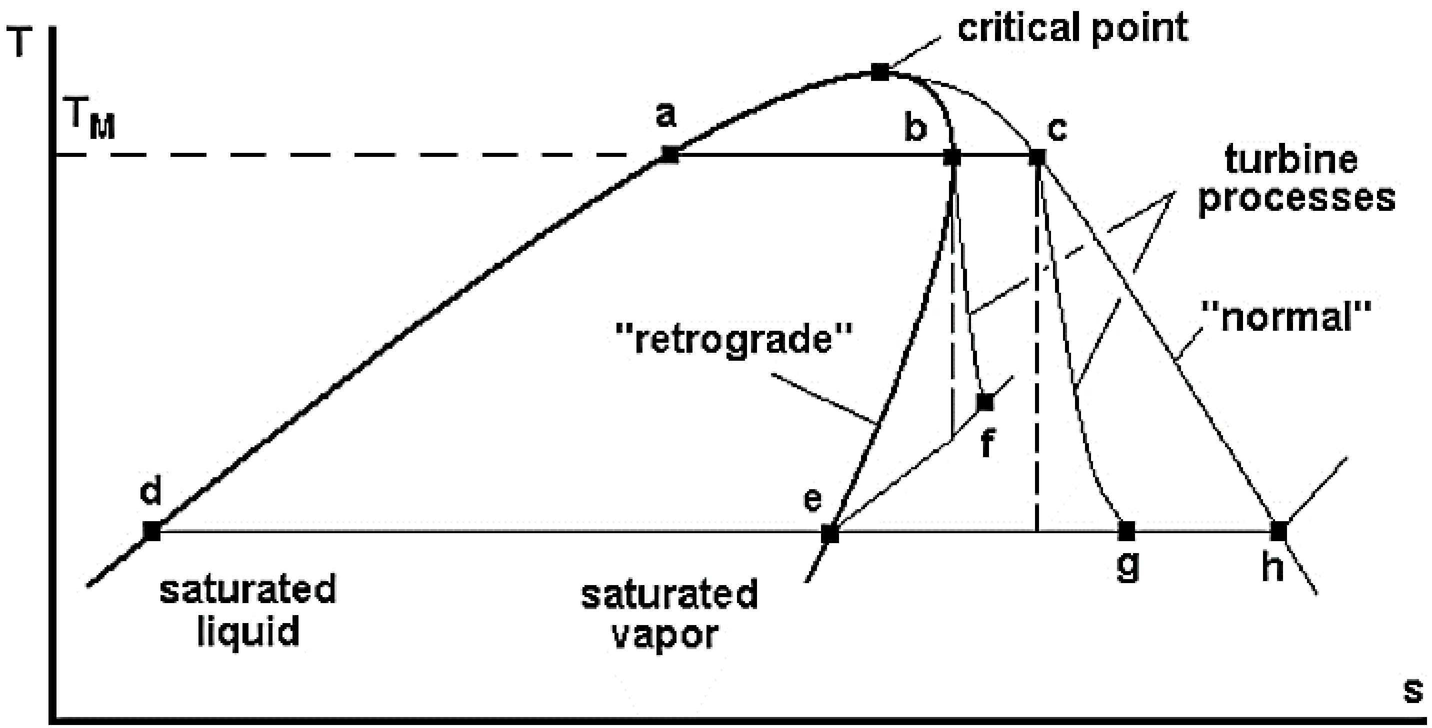
PW	Production well	E	Evaporator	CV	Control valve
P	Pump	PH	Preheater	SV	Stop valve
WH	Wellhead	IP	Injection pump	T	Turbine
WV	Well valve	FF	Final filter	G	Generator
SR	Sand remover	IW	Injection well	EG	Electric grid
C	Condenser	CP	Condensate pump	CWP	Cooling water pump
M	Make-up water	CT	Cooling tower		

Binary Cycle (ORC)





Binary Cycle (ORC)



Ts diagram contrasting *normal* and *retrograde* saturated vapor curves

Binary Cycle (ORC)

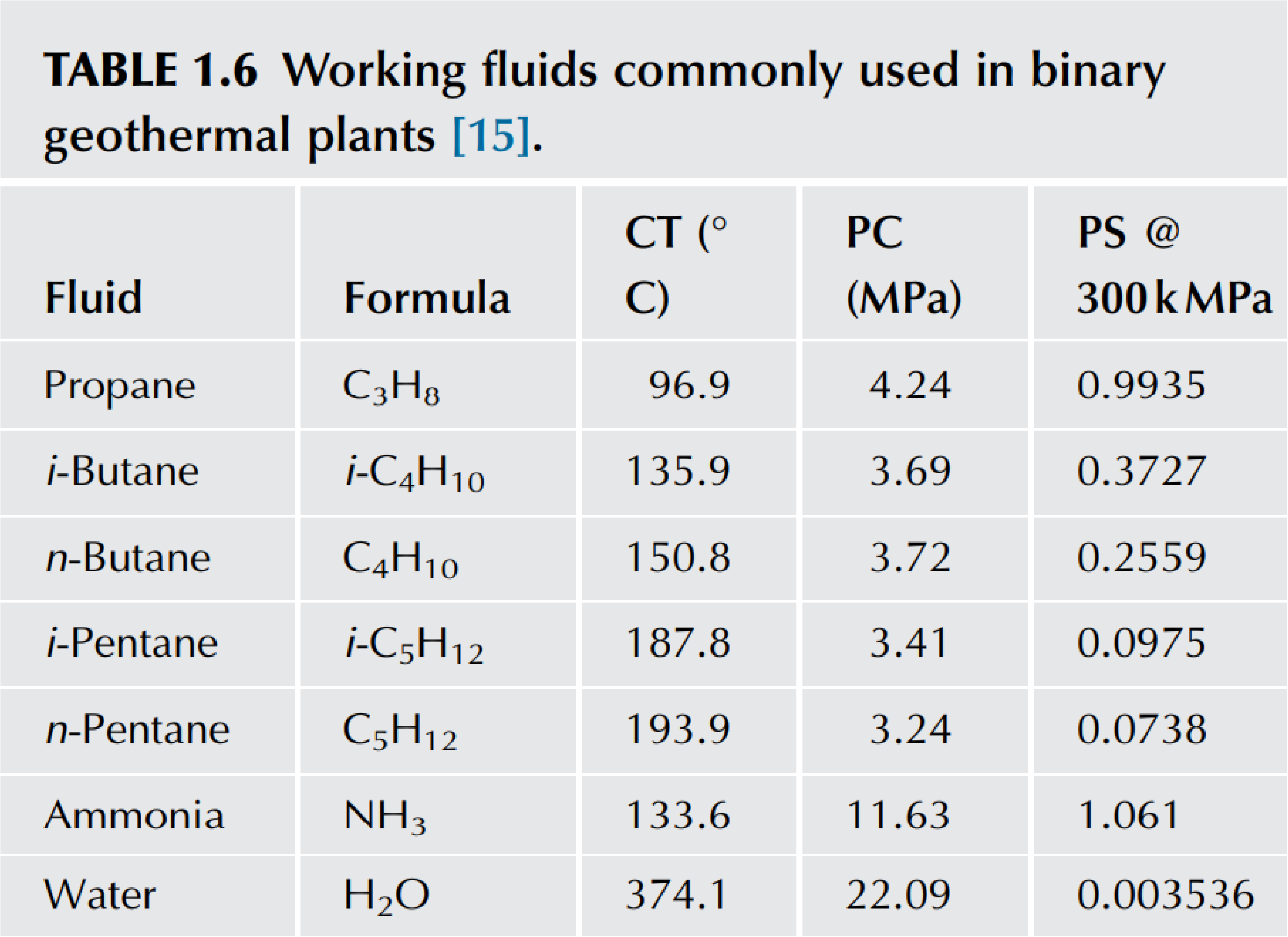
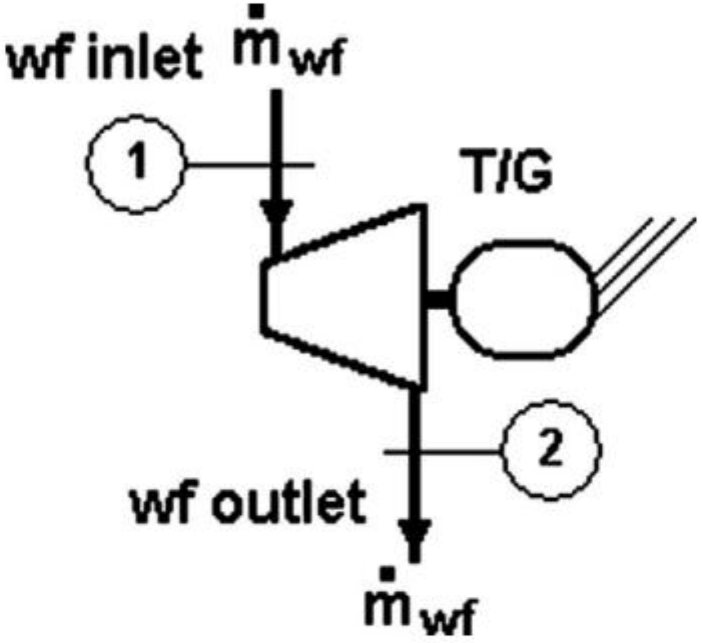


TABLE 1.6 Working fluids commonly used in binary geothermal plants [15].

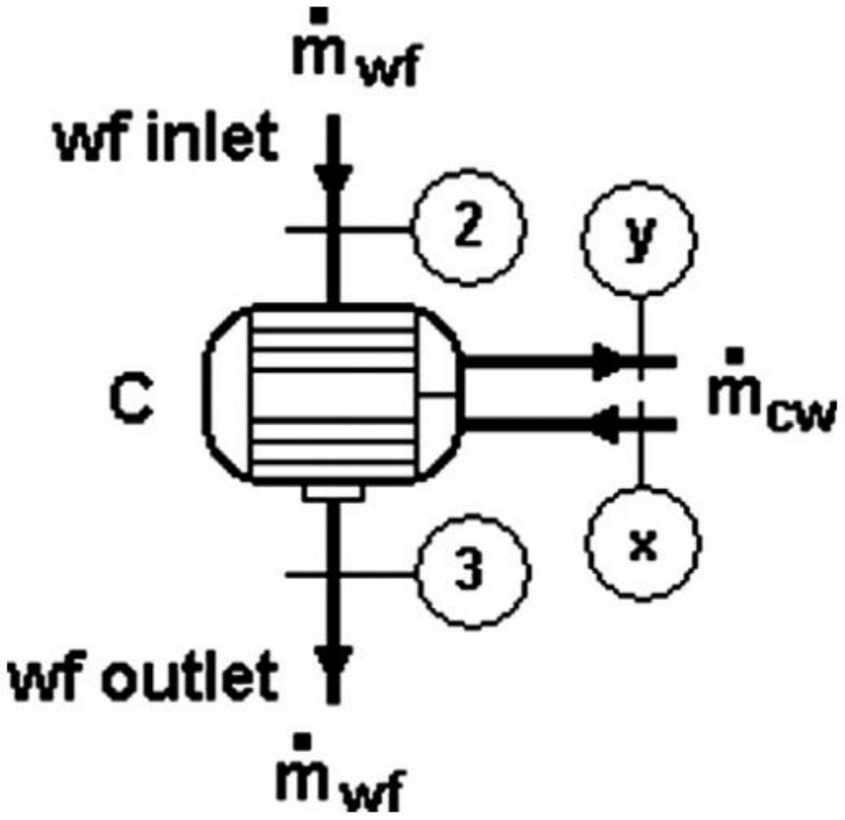
Fluid	**Formula**	**CT (° C)**	**PC (MPa)**	**PS @ 300 k MPa**
Propane	C_3H_8	96.9	4.24	0.9935
i-Butane	*i*-C_4H_{10}	135.9	3.69	0.3727
n-Butane	C_4H_{10}	150.8	3.72	0.2559
i-Pentane	*i*-C_5H_{12}	187.8	3.41	0.0975
n-Pentane	C_5H_{12}	193.9	3.24	0.0738
Ammonia	NH_3	133.6	11.63	1.061
Water	H_2O	374.1	22.09	0.003536

Binary Cycle (ORC): Energy análisis: Turbine



$$\dot{W}_t = \dot{m}_{wf}(h_1 - h_2) = \dot{m}_{wf}\eta_t(h_1 - h_{2s})$$

Binary Cycle (ORC): Energy análisis: Condenser

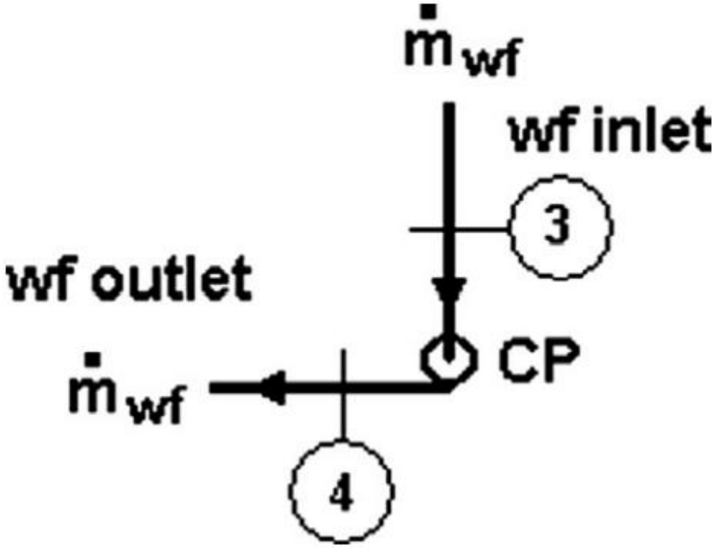


$$\dot{Q}_c = \dot{m}_{wf}(h_2 - h_3)$$

$$\dot{m}_{cw}(h_y - h_x) = \dot{m}_{wf}(h_2 - h_3)$$

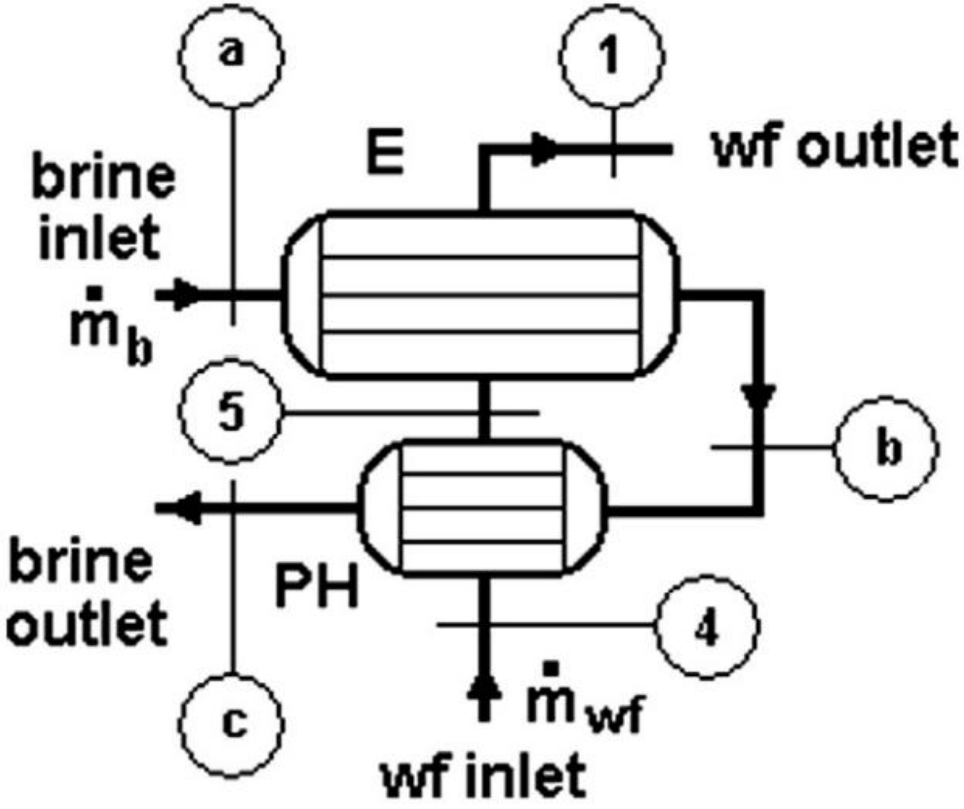
$$\dot{m}_{cw}\overline{c}(T_y - T_x) = \dot{m}_{wf}(h_2 - h_3)$$

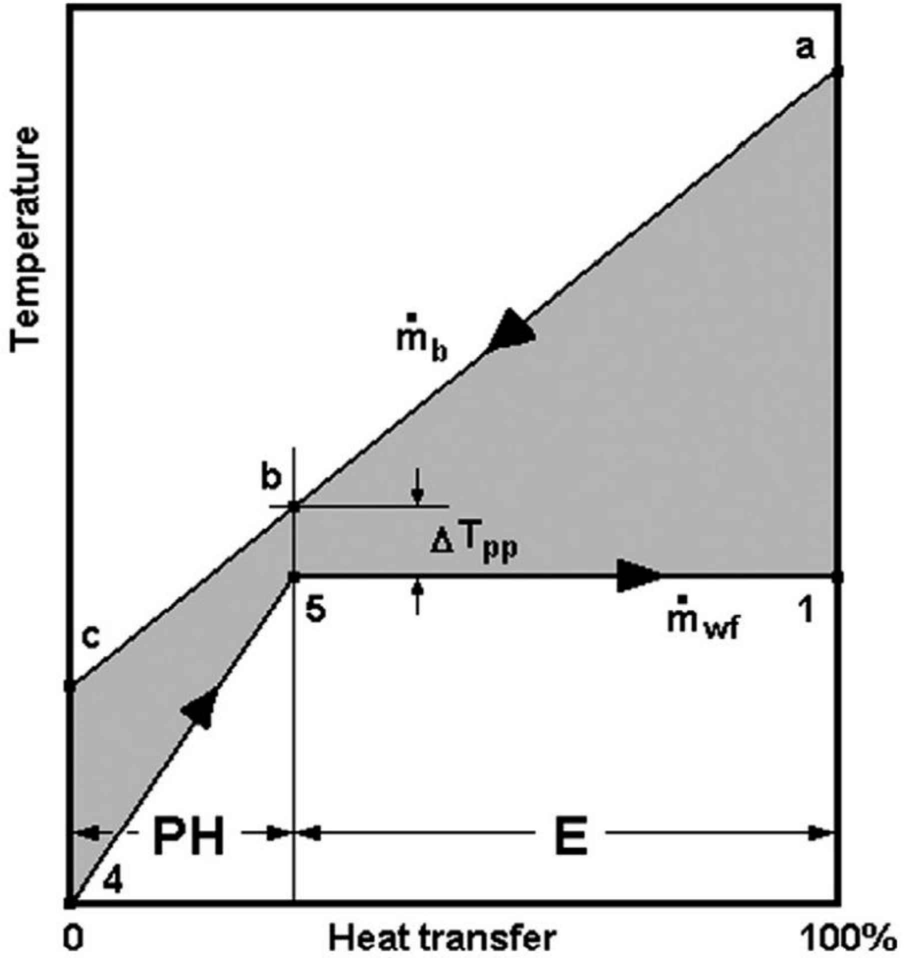
Binary Cycle (ORC): Energy análisis: Feed pump



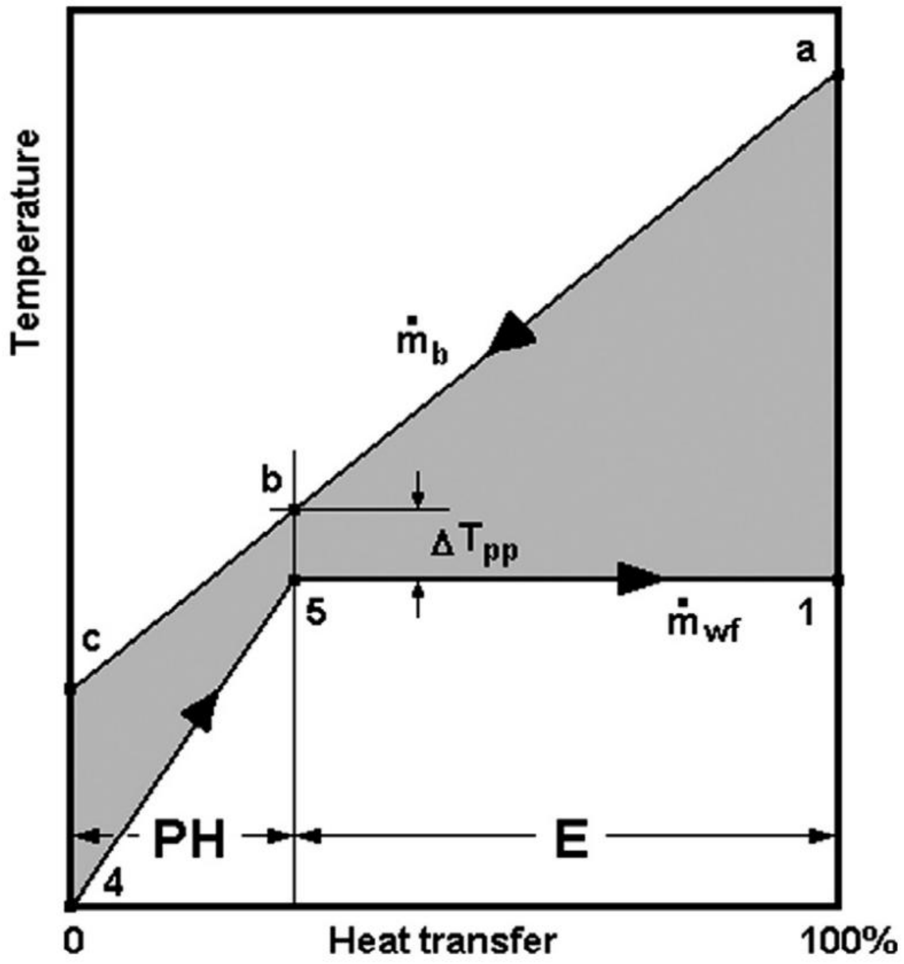
$$\dot{W}_p = \dot{m}_{wf}(h_4 - h_3) = \dot{m}_{wf}(h_{4s} - h_3)/\eta_p$$

Binary Cycle (ORC): Energy análisis: Heat Exchanger





Binary Cycle (ORC): Energy análisis: Heat Exchanger

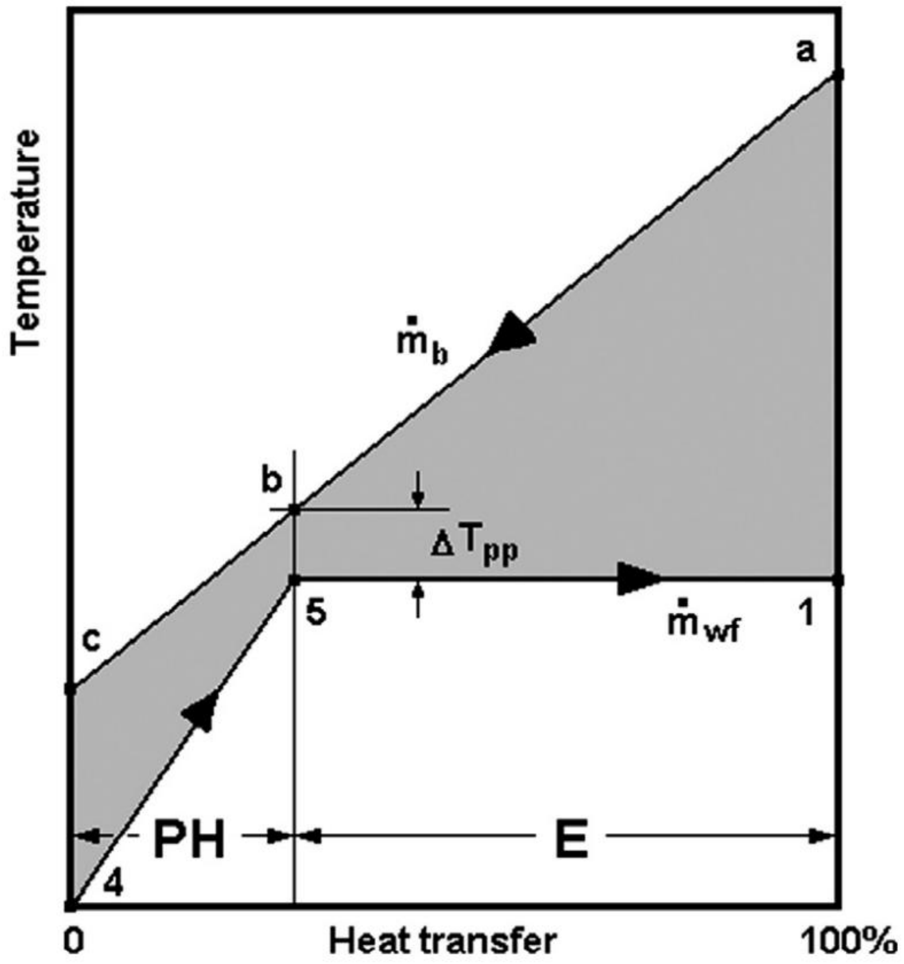


$$\dot{m}_b(h_a - h_c) = \dot{m}_{wf}(h_1 - h_4)$$

$$\dot{m}_b\bar{c}_b(T_a - T_c) = \dot{m}_{wf}(h_1 - h_4)$$

$$\dot{m}_b = \dot{m}_{wf}\frac{h_1 - h_4}{\bar{c}_b(T_a - T_c)}$$

Binary Cycle (ORC): Energy análisis: Heat Exchanger

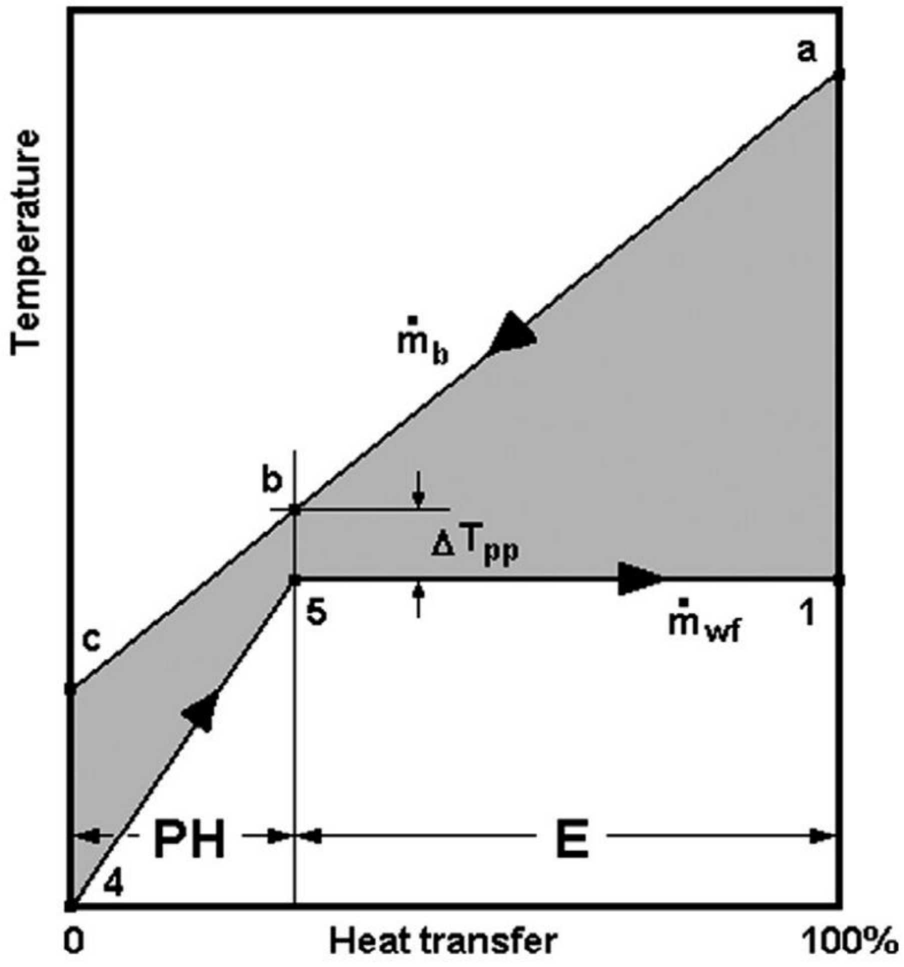


$$\text{Preheater:} \quad \dot{m}_b \bar{c}_b (T_b - T_c) = \dot{m}_{wf} (h_5 - h_4)$$

$$\text{Evaporator:} \quad \dot{m}_b \bar{c}_b (T_a - T_b) = \dot{m}_{wf} (h_1 - h_5)$$

- The brine inlet temperature T_a is known
- The pinch-point temperature difference, ΔT_{pp}, is generally known from manufacturer's specifications
- T_b to be found from the known value for T_5 (while it is theoretically possible for the pinch-point to occur at the cold end of the preheater, this practically never happens)

Binary Cycle (ORC): Heat Exchanger Surface Area



$$\dot{Q}_{PH} = \overline{U}A_{PH}\ LMTD|_{PH}$$

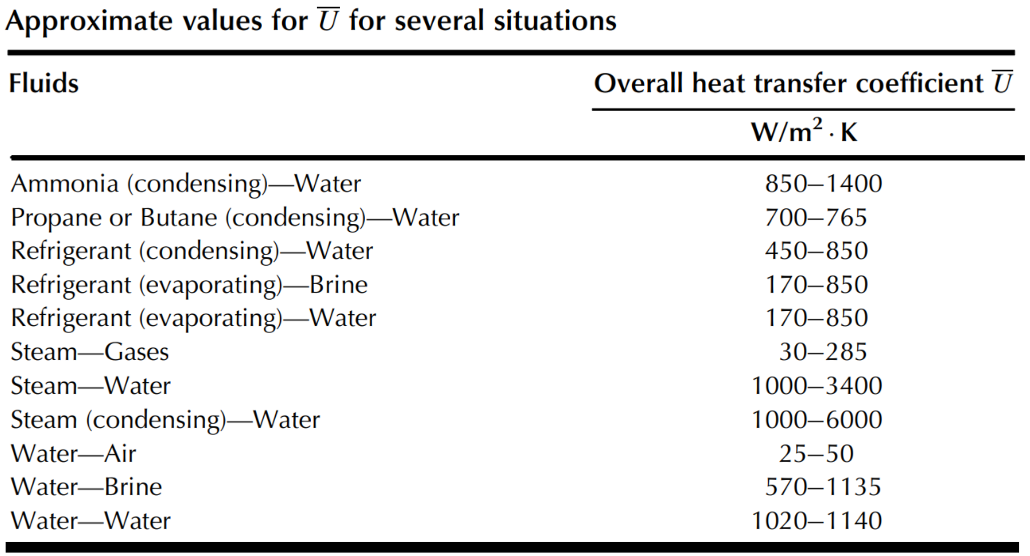
$$LMTD|_{PH} = \frac{(T_b - T_5) - (T_c - T_4)}{\ln\left[\dfrac{T_b - T_5}{T_c - T_4}\right]}$$

$$\dot{Q}_{PH} = \dot{m}_b \overline{c}_b (T_b - T_c) = \dot{m}_{wf}(h_5 - h_4)$$

$$\dot{Q}_E = \overline{U}A_E\ LMTD|_E$$

$$LMTD|_E = \frac{(T_a - T_1) - (T_b - T_5)}{\ln\left[\dfrac{T_a - T_1}{T_b - T_5}\right]}$$

$$\dot{Q}_E = \dot{m}_b \overline{c}_b (T_a - T_b) = \dot{m}_{wf}(h_1 - h_5)$$


Approximate values for $\overline{U}$ for several situations

Fluids	Overall heat transfer coefficient $\overline{U}$ $W/m^2 \cdot K$
Ammonia (condensing)—Water	850–1400
Propane or Butane (condensing)—Water	700–765
Refrigerant (condensing)—Water	450–850
Refrigerant (evaporating)—Brine	170–850
Refrigerant (evaporating)—Water	170–850
Steam—Gases	30–285
Steam—Water	1000–3400
Steam (condensing)—Water	1000–6000
Water—Air	25–50
Water—Brine	570–1135
Water—Water	1020–1140

Binary Cycle (ORC): Energy analysis

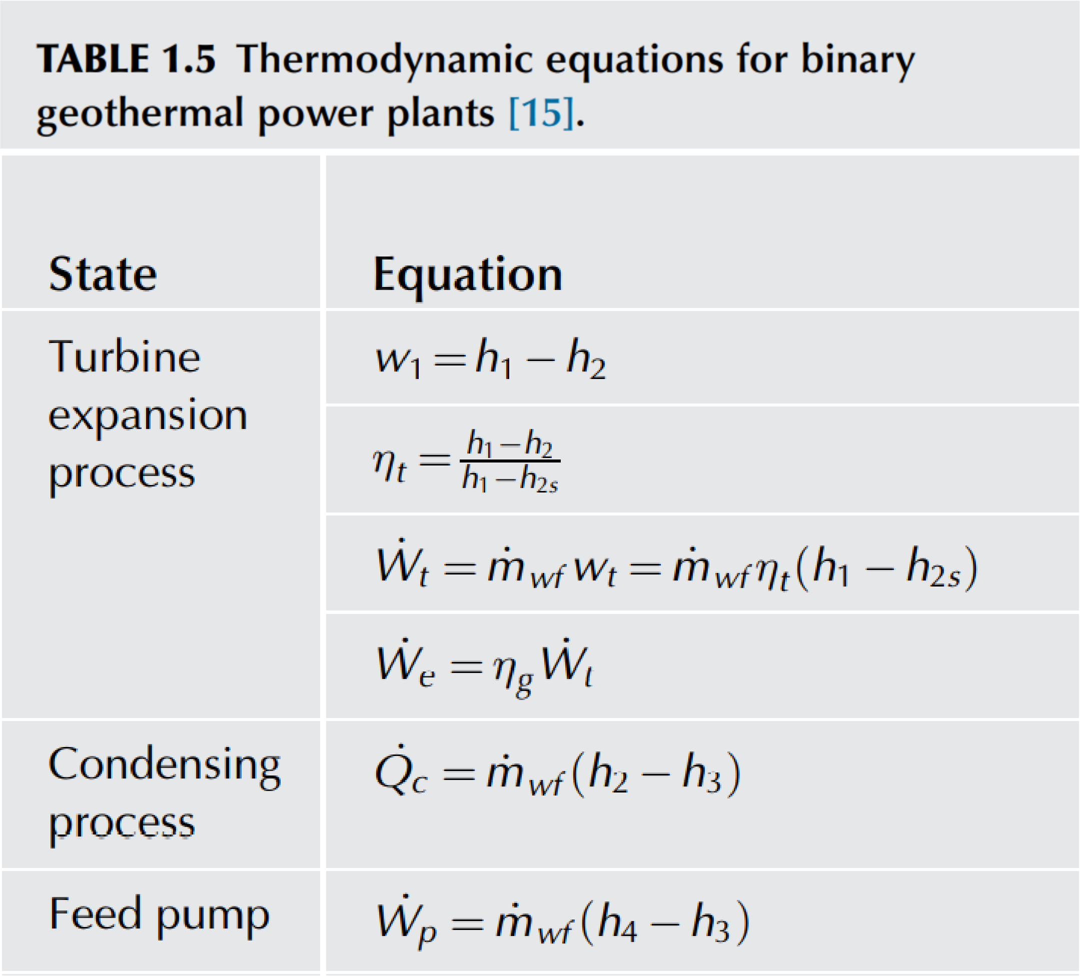


TABLE 1.5 Thermodynamic equations for binary geothermal power plants [15].

State	Equation
Turbine expansion process	$w_1 = h_1 - h_2$
	$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}}$
	$\dot{W}_t = \dot{m}_{wf} w_t = \dot{m}_{wf} \eta_t (h_1 - h_{2s})$
	$\dot{W}_e = \eta_g \dot{W}_t$
Condensing process	$\dot{Q}_c = \dot{m}_{wf} (h_2 - h_3)$
Feed pump	$\dot{W}_p = \dot{m}_{wf} (h_4 - h_3)$

Binary Cycle (ORC): Energy analysis

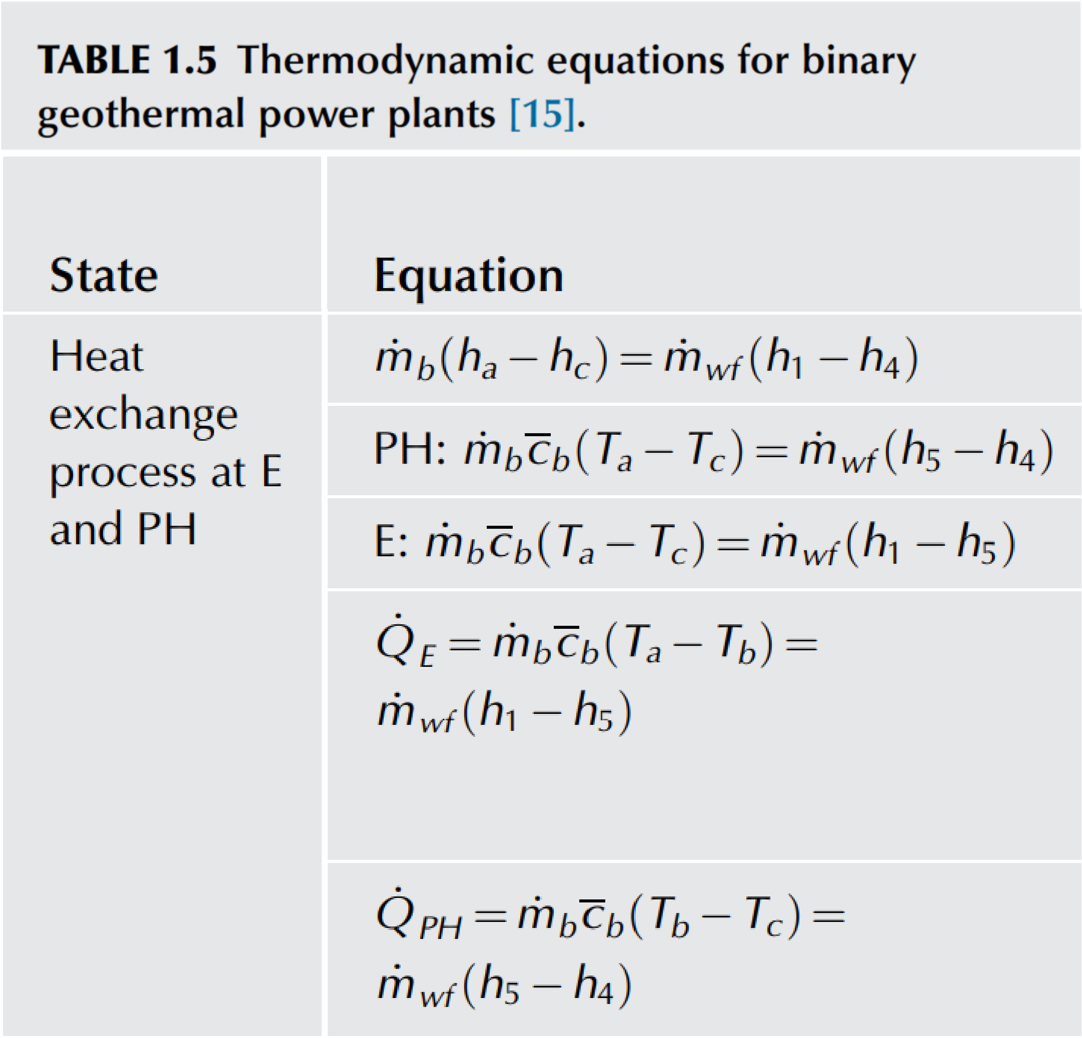


TABLE 1.5 Thermodynamic equations for binary geothermal power plants [15].

State	Equation
Heat exchange process at E and PH	$\dot{m}_b(h_a - h_c) = \dot{m}_{wf}(h_1 - h_4)$
	PH: $\dot{m}_b\overline{c}_b(T_a - T_c) = \dot{m}_{wf}(h_5 - h_4)$
	E: $\dot{m}_b\overline{c}_b(T_a - T_c) = \dot{m}_{wf}(h_1 - h_5)$
	$\dot{Q}_E = \dot{m}_b\overline{c}_b(T_a - T_b) = \dot{m}_{wf}(h_1 - h_5)$
	$\dot{Q}_{PH} = \dot{m}_b\overline{c}_b(T_b - T_c) = \dot{m}_{wf}(h_5 - h_4)$

Binary Cycle (ORC): Energy analysis

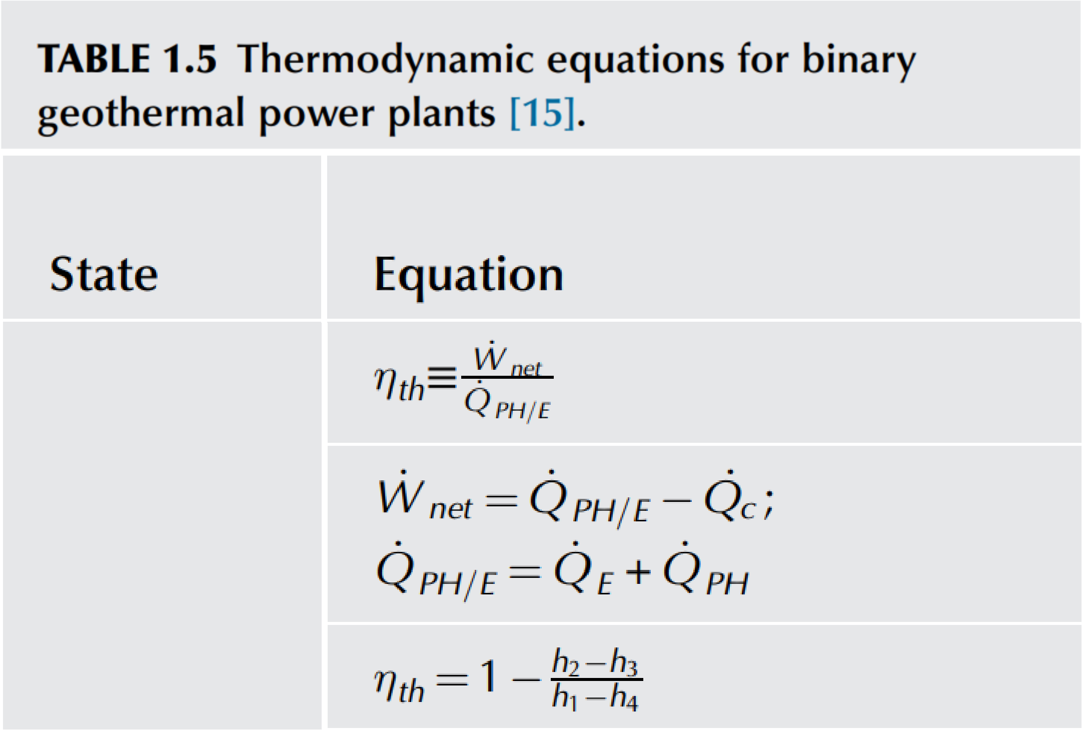


TABLE 1.5 Thermodynamic equations for binary geothermal power plants [15].

State	Equation
	$\eta_{th} \equiv \frac{\dot{W}_{net}}{\dot{Q}_{PH/E}}$
	$\dot{W}_{net} = \dot{Q}_{PH/E} - \dot{Q}_{C};$ $\dot{Q}_{PH/E} = \dot{Q}_{E} + \dot{Q}_{PH}$
	$\eta_{th} = 1 - \frac{h_2 - h_3}{h_1 - h_4}$

Binary Cycle (ORC): Exergy analysis

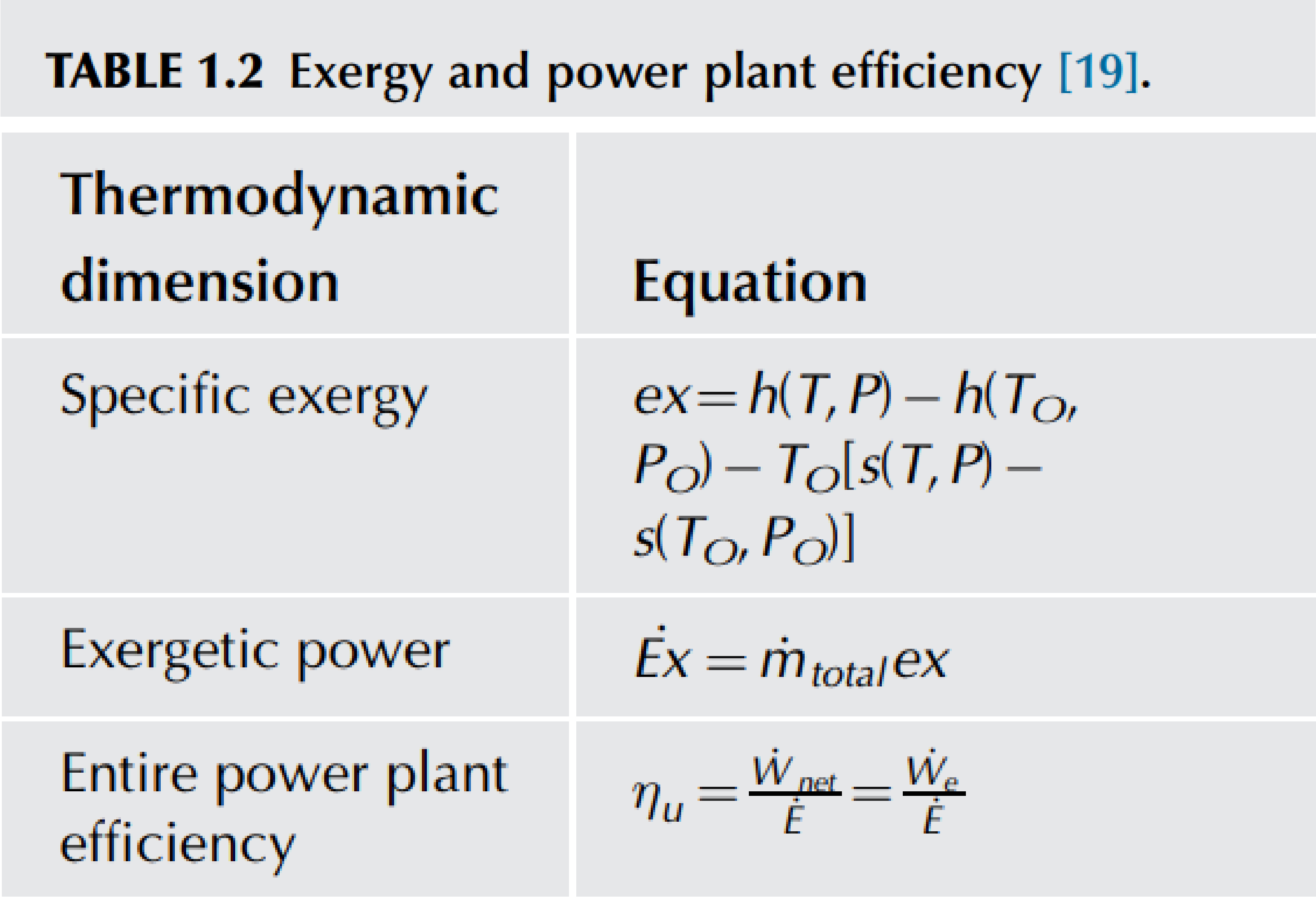


TABLE 1.2 Exergy and power plant efficiency [19].

Thermodynamic dimension	Equation
Specific exergy	$ex = h(T,P) - h(T_O, P_O) - T_O[s(T,P) - s(T_O,P_O)]$
Exergetic power	$\dot{Ex} = \dot{m}_{total}ex$
Entire power plant efficiency	$\eta_u = \frac{\dot{W}_{net}}{\dot{E}} = \frac{\dot{W}_e}{\dot{E}}$

Second-Law Analysis

Turbine:

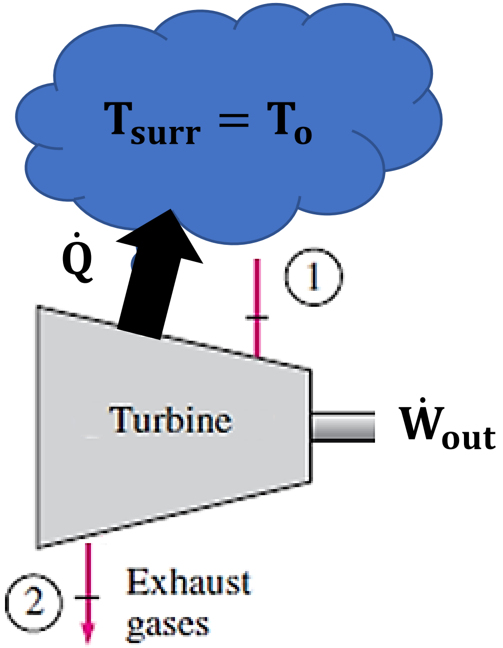
Exergy balance:

$$E\dot{X}_{in} - E\dot{X}_{out} - E\dot{X}_{loss} = \frac{dEX_{sys}}{dt} = 0\ (\text{steady} - \text{state}, \text{steady} - \text{flow})$$

$$\dot{m}ex_1 - \dot{W}_{out} - \dot{m}ex_2 - E\dot{X}_{loss} = 0$$

$$E\dot{X}_{loss} = E\dot{X}_{supplied} - E\dot{X}_{recovered}$$

$$E\dot{X}_{loss} = \underbrace{\boxed{[\dot{m}(ex_1 - ex_2)]}}_{E\dot{X}_{supplied}} - \underbrace{\boxed{\dot{W}_{out}}}_{E\dot{X}_{recovered}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

The second-law efficiency:

$$\eta_{II,T} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

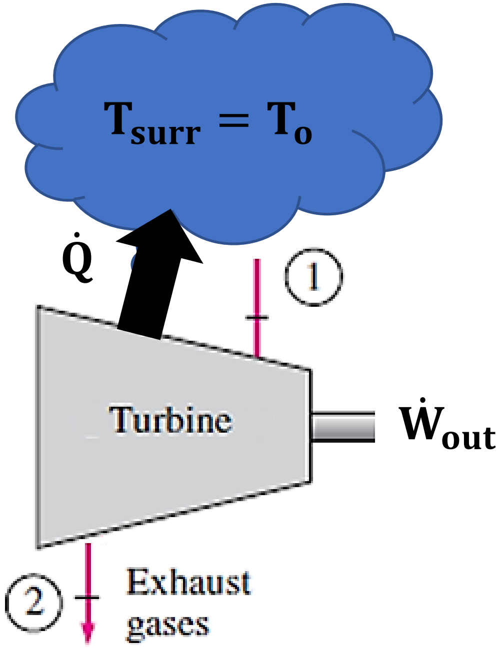
$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}(ex_1 - ex_2)}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{m}[(h_1 - h_2) - T_o(s_1 - s_2)]}$$

But the first law of thermodynamics requires:

$$\dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$

$$\eta_{II,T} = \frac{\dot{W}_{out}}{\dot{W}_{out} + \dot{Q} - \dot{m}T_o(s_1 - s_2)} = \frac{\dot{W}_{out}}{\dot{W}_{rev}}$$



T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

Second-Law Analysis

Turbine:

Entropy balance:

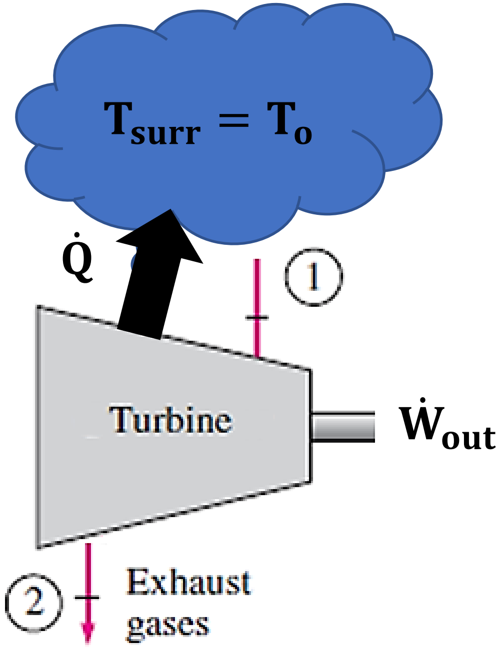
$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady} - \text{state, steady} - \text{flow)}$$

$$\dot{m}s_1 - \dot{m}s_2 - \left(\frac{\dot{Q}}{T_{surr}}\right) + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \left(\frac{\dot{Q}}{T_{surr}}\right) \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \left(\frac{T_o}{T_o}\right)\dot{Q} \geq 0$$

$$\text{But}: \dot{m}(h_1 - h_2) - \dot{Q} = \dot{W}_{out}$$



$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) + \dot{m}(h_1 - h_2) - \dot{W}_{out} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = [\dot{m}(ex_1 - ex_2)] - \dot{W}_{out}$$

T_{surr}: is the surrounding temperature

T_o: is the dead-state temperature

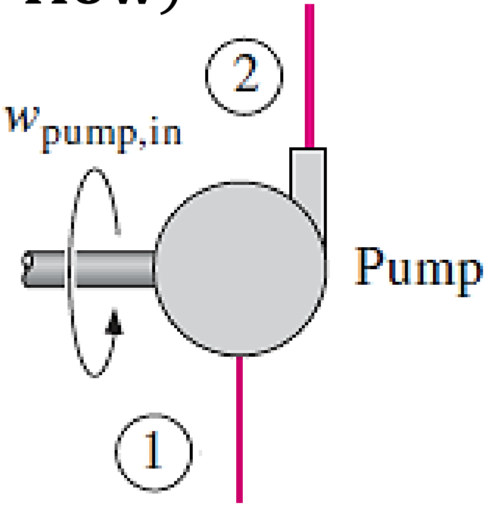
Second-Law Analysis

Pump:

Exergy balance:

$$\mathbf{E\dot{X}_{in} - E\dot{X}_{out} - E\dot{X}_{loss} = \frac{dEX_{sys}}{dt} = 0\ (steady - state, steady - flow)}$$

$$\mathbf{\dot{m}ex_1 + \dot{W}_{in} - \dot{m}ex_2 - E\dot{X}_{loss} = 0}$$



$$\mathbf{E\dot{X}_{loss} = E\dot{X}_{supplied} - E\dot{X}_{recovered}}$$

$$\mathbf{E\dot{X}_{loss} = \underbrace{\boxed{\dot{W}_{in}}}_{E\dot{X}_{supplied}} - \underbrace{\boxed{\dot{m}(x_2 - x_1)}}_{E\dot{X}_{recovered}}}$$

T_o: is the dead-state temperature.
Pumping liquids is not usually associated with heat transfer

Second-Law Analysis

Pump:

The second-law efficiency:

$$\eta_{II,P} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

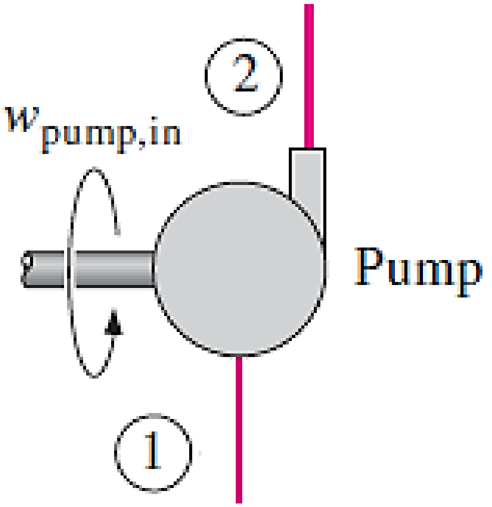
$$\eta_{II,P} = \frac{\dot{m}(ex_2 - ex_1)}{\dot{W}_{in}}$$

$$\eta_{II,P} = \frac{\dot{m}[(h_2 - h_1) - T_o(s_2 - s_1)]}{\dot{W}_{in}}$$

But the first law of thermodynamics requires:

$$\dot{m}(h_2 - h_1) = \dot{W}_{in}$$

$$\eta_{II,P} = \frac{\dot{W}_{in} - \dot{m}T_o(s_2 - s_1)}{\dot{W}_{in}} = \frac{\dot{W}_{rev}}{\dot{W}_{in}} \approx 1$$



T_o: is the dead-state temperature.
Pumping liquids is not usually associated with heat transfer

Second-Law Analysis

Pump:

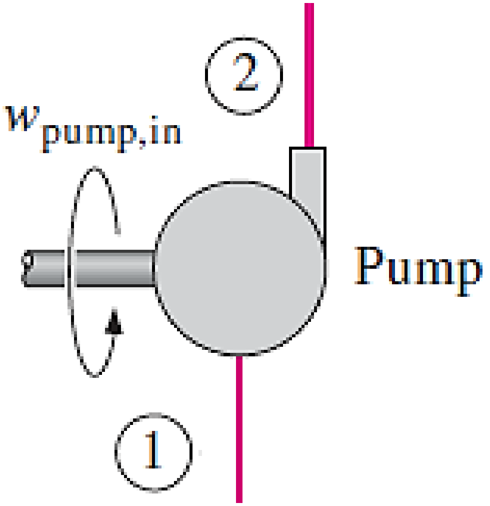
Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady − state, steady − flow)}$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o(s_2 - s_1) \geq 0$$



T_o: is the dead-state temperature. Pumping liquids is not usually associated with heat transfer

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

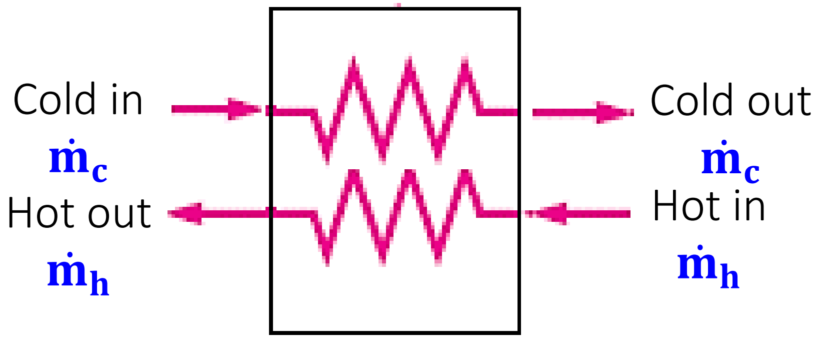
Exergy balance:

$$\dot{EX}_{in} - \dot{EX}_{out} - \dot{EX}_{loss} = \frac{dEX_{sys}}{dt} = 0 \ (\mathrm{steady-state, steady-flow})$$

$$\dot{m}_c ex_{c,in} + \dot{m}_h ex_{h,in} - \dot{m}_c ex_{c,out} - \dot{m}_h ex_{h,out} - \dot{EX}_{loss} = 0$$

$$\dot{EX}_{loss} = \dot{EX}_{supplied} - \dot{EX}_{recovered}$$

$$\dot{EX}_{loss} = \underbrace{\boxed{\dot{m}_h\left(ex_{h,in} - ex_{h,out}\right)}}_{\dot{EX}_{supplied}} - \underbrace{\boxed{\dot{m}_c\left(ex_{c,out} - ex_{c,in}\right)}}_{\dot{EX}_{recovered}}$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

The second-law efficiency:

$$\eta_{II,HX}=\frac{\text{Exergy recovered}}{\text{Exergy supplied}}=1-\frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

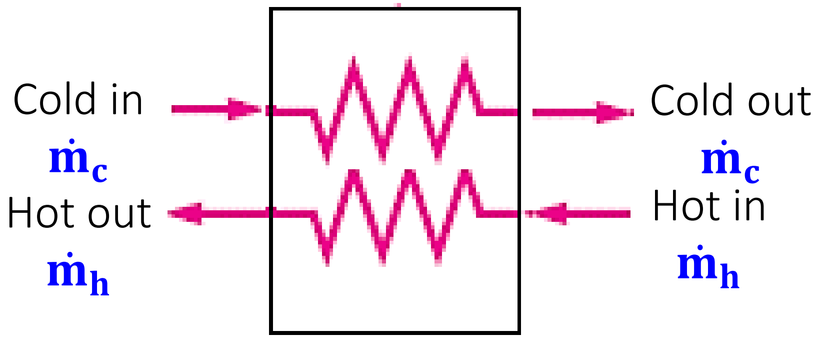
$$\eta_{II,HX}=\frac{\dot{m}_c\left(ex_{c,out}-ex_{c,in}\right)}{\dot{m}_h\left(ex_{h,in}-ex_{h,out}\right)}$$

$$\eta_{II,HX}=\frac{\dot{m}_c\left[\left(h_{c,out}-h_{c,in}\right)-T_o\left(s_{c,out}-s_{c,in}\right)\right]}{\dot{m}_h\left[\left(h_{h,in}-h_{h,out}\right)-T_o\left(s_{h,in}-s_{h,out}\right)\right]}$$

But the first law of thermodynamics requires:

$$\dot{m}_c\left(h_{c,out}-h_{c,in}\right)=\dot{m}_h\left(h_{h,in}-h_{h,out}\right)$$

$$\eta_{II,HX}=\frac{\dot{m}_h\left(h_{h,in}-h_{h,out}\right)-\dot{m}_cT_o\left(s_{c,out}-s_{c,in}\right)}{\dot{m}_h\left[\left(h_{h,in}-h_{h,out}\right)-T_o\left(s_{h,in}-s_{h,out}\right)\right]}$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Condenser & separator (Heat Exchanger – HX):

Entropy balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0\ (\text{steady} - \text{state}, \text{steady} - \text{flow})$$

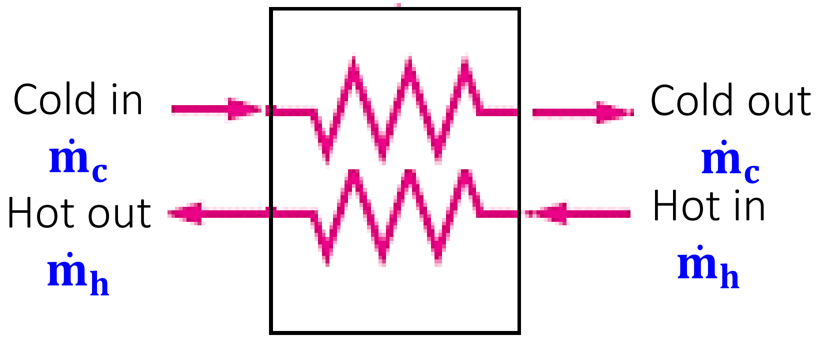
$$\dot{m}_c s_{c,in} + \dot{m}_h s_{h,in} - \dot{m}_c s_{c,out} - \dot{m}_h s_{h,out} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_c(s_{c,out} - s_{c,in}) - \dot{m}_h(s_{h,in} - s_{h,out}) \geq 0$$

$$\dot{m}_c(s_{c,out} - s_{c,in}) \geq \dot{m}_h(s_{h,in} - s_{h,out})$$

$$\dot{EX}_{loss} = T_o \dot{S}_{gen}$$

$$\dot{EX}_{loss} = \dot{m}_c T_o(s_{c,out} - s_{c,in}) - \dot{m}_h T_o(s_{h,in} - s_{h,out}) \geq 0$$



T_o: is the dead-state temperature.
In heat exchangers, heat transfer to surrounding should be minimized

Second-Law Analysis

Binary Cycle (ORC):

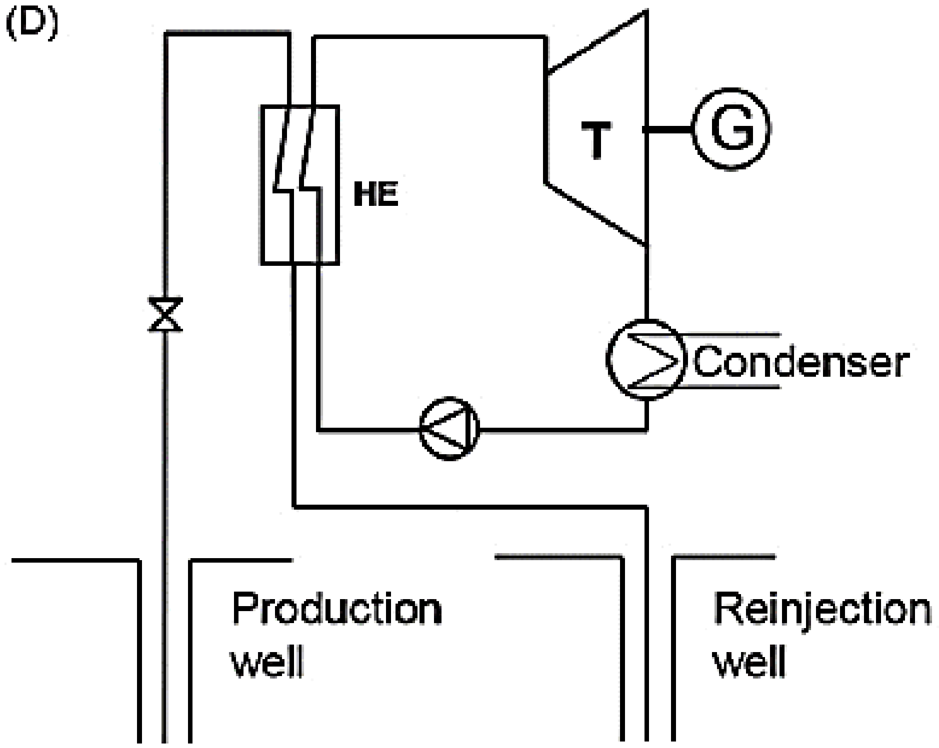
Exergy balance:

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{loss} = \frac{dX_{sys}}{dt} = 0\ (steady - state, steady - flow)$$

$$\dot{m}ex_{production} - \dot{W}_{net} - \dot{m}ex_{reinjection} - \dot{X}_{loss} = 0$$

$$\dot{X}_{loss} = \dot{X}_{supplied} - \dot{X}_{recovered}$$

$$\dot{X}_{loss} = \underbrace{\boxed{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}}_{\dot{X}_{supplied}} - \underbrace{\boxed{\dot{W}_{net}}}_{\dot{X}_{recovered}}$$



Second-Law Analysis

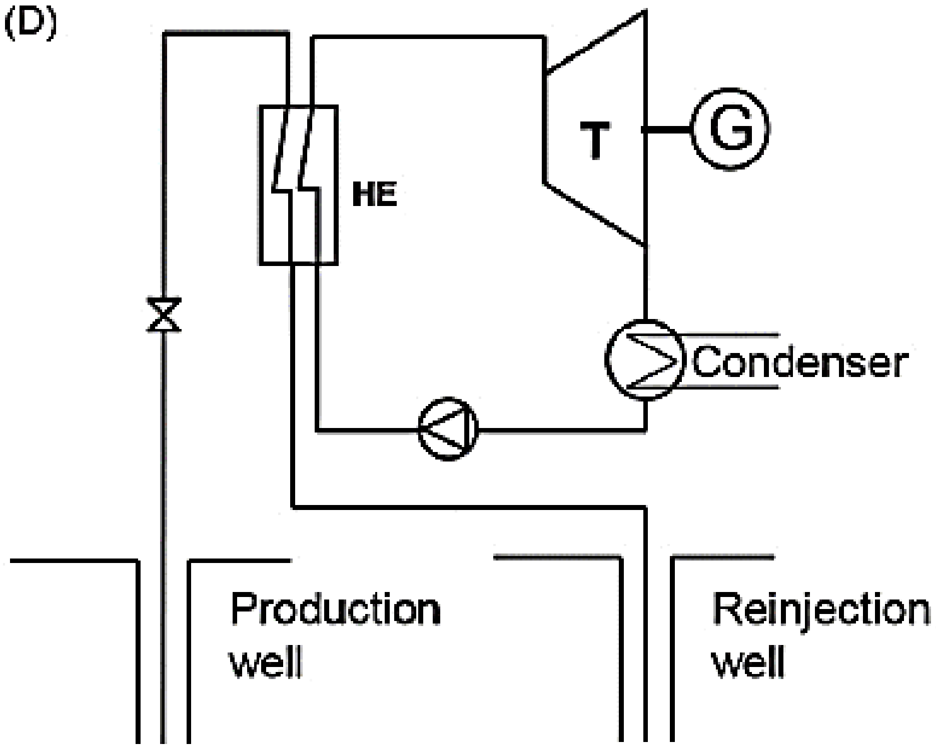
Binary Cycle (ORC):

The second-law efficiency:

$$\eta_{II,R} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy supplied}}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}\left(ex_{production} - ex_{reinjection}\right)}$$

$$\eta_{II,R} = \frac{\dot{W}_{net}}{\dot{m}[\Delta h - T_o \Delta s]}$$



Second-Law Analysis

Binary Cycle (ORC):

Entropy balance:

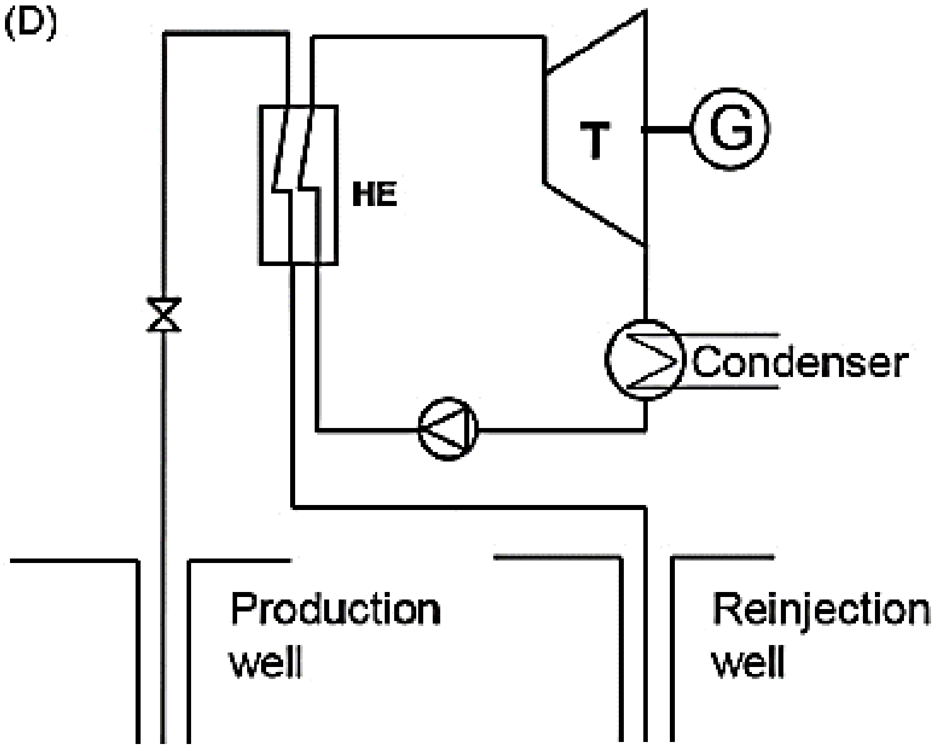
$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt} = 0 \text{ (steady − state, steady − flow)}$$

$$\dot{m}s_{production} - \dot{m}s_{reinjection} - \frac{\dot{Q}_L}{T_L} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}\left(s_{reinjection} - s_{production}\right) + \frac{\dot{Q}_L}{T_L} \geq 0$$

$$\dot{EX}_{loss} = T_o\dot{S}_{gen} = \dot{m}T_o\left(s_{reinjection} - s_{production}\right) + \dot{Q}$$

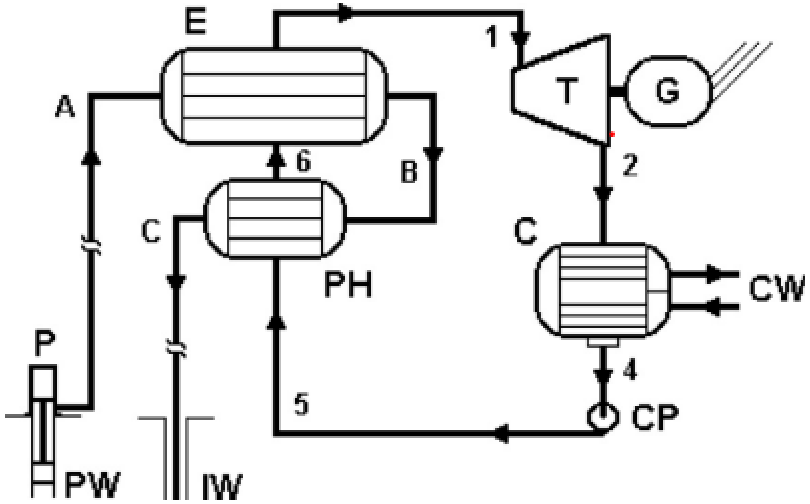
Where $T_o = T_L$



Problem 3

An **ORC** geothermal power plant operates from a reservoir that can provide hot liquid at 440 K (167 ºC)

Net power 1200 kW
Brine inlet temperature, T_A = 440 K (saturated liquid, c_b = 4.19 kJ/kg·K)
Pinch-point temperature difference 5 K
Working fluid: isopentane
Preheater-evaporator pressure, $P_5 = P_6 = P_1$ = 2.0 MPa
Condensing temperature, T_4 = 320 K
Turbine isentropic efficiency 85 %
Feed pump isentropic efficiency 75 %



Estimate:

- Energy analysis. **Thermal Efficiency**
- Exergy analysis. **Exergy efficiency**. Exergy destruction
- **Optimize** preheater-evaporator pressure (**P_5**)

- Environmental conditions: 25 ºC, 1 bar

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

Co-funded by the
Erasmus+ Programme
of the European Union

GEB

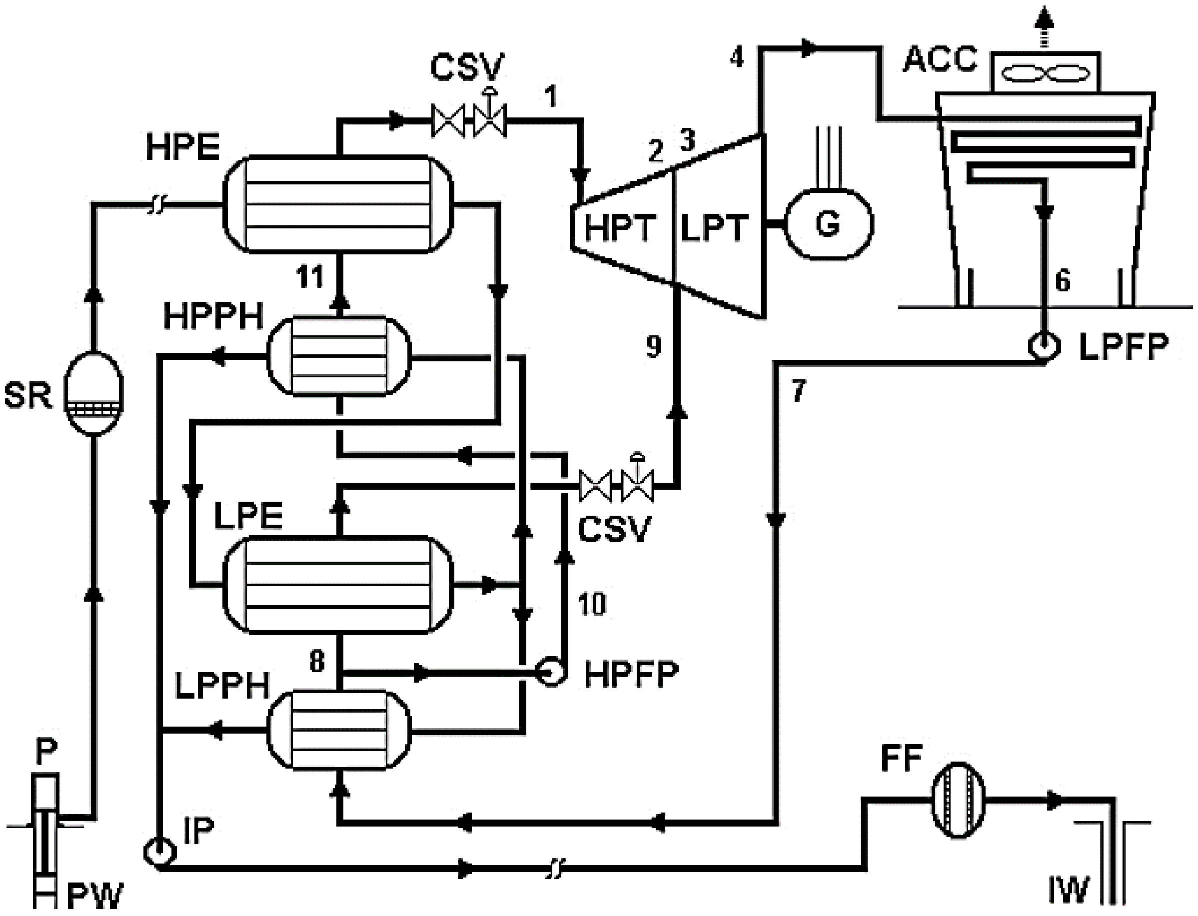
Geothermal Energy Capacity Building in Egypt (GEB)

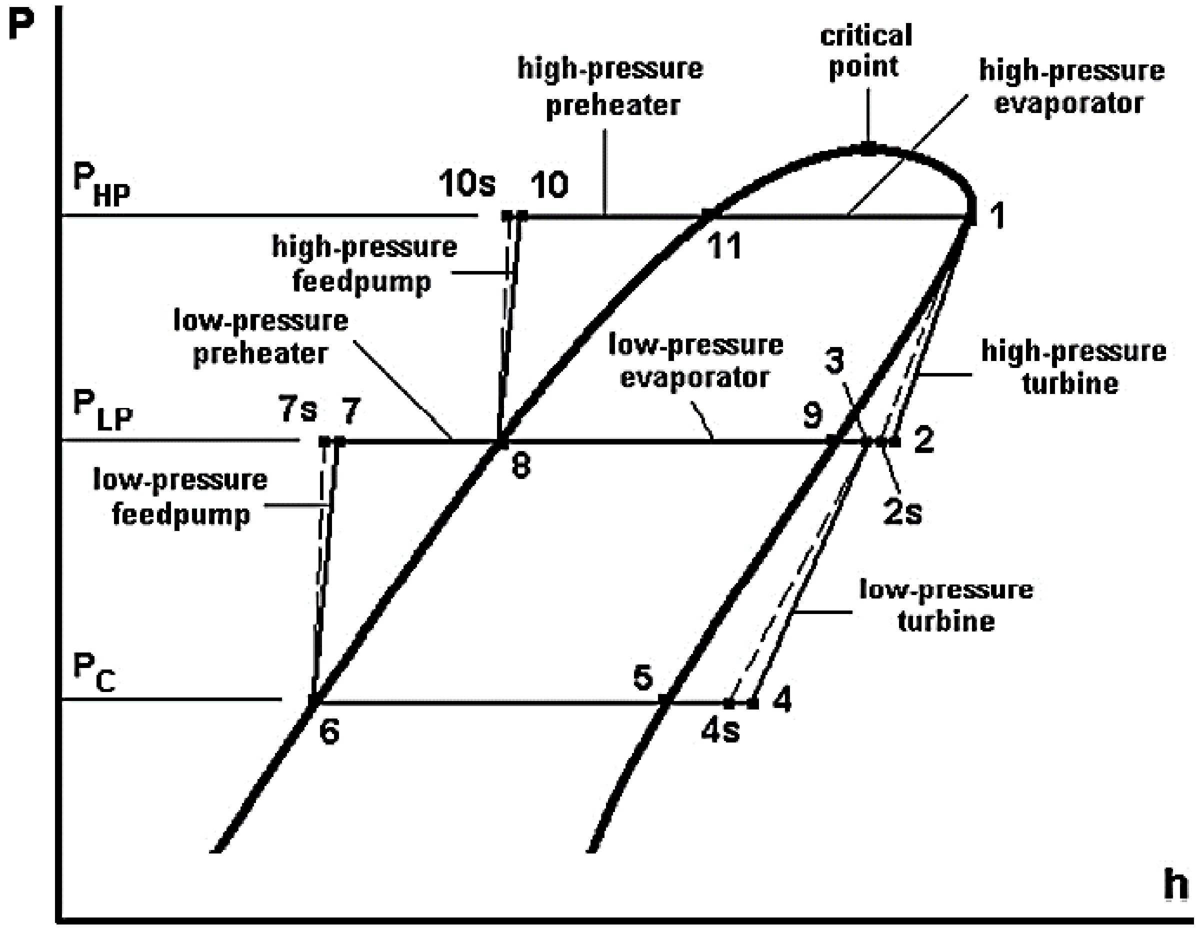
Power generation from low-enthalpy geothermal resources: Binary cycle power plants

Energy and exergy analysis

Co-funded by the
Erasmus+ Programme
of the European Union

Advanced Binary Cycle: Dual Pressure BC





Advanced Binary Cycle: Dual Pressure BC

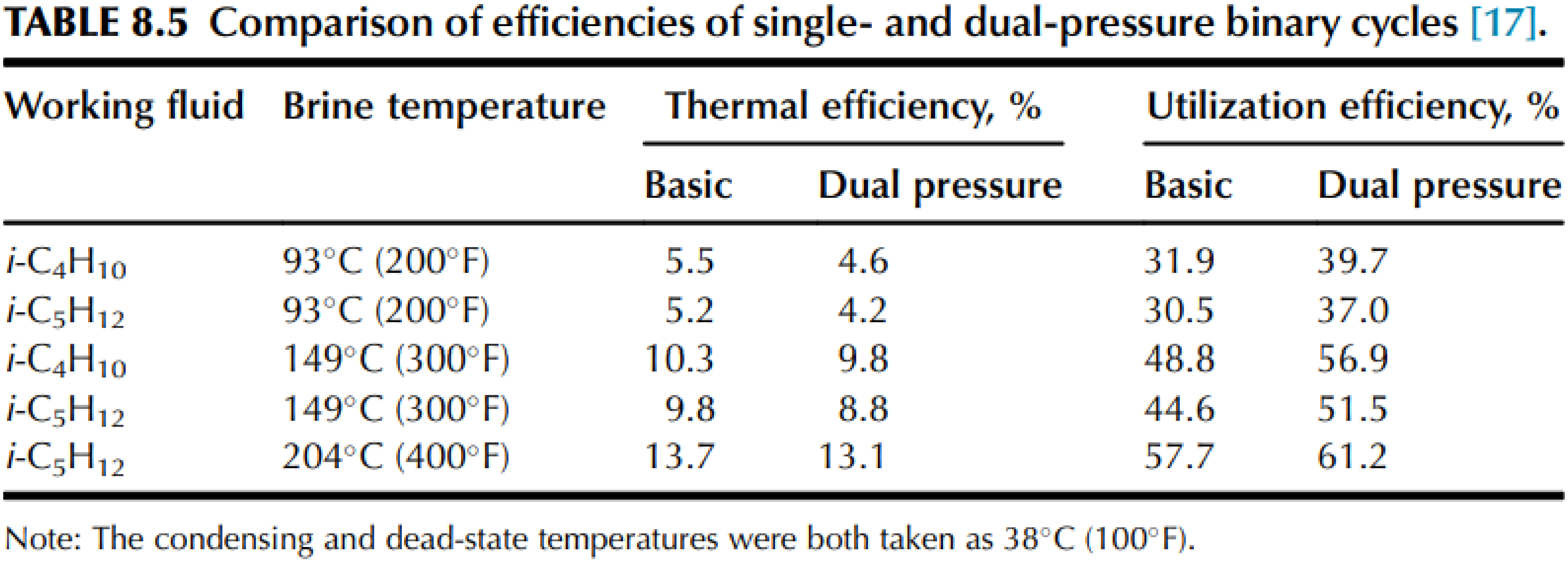
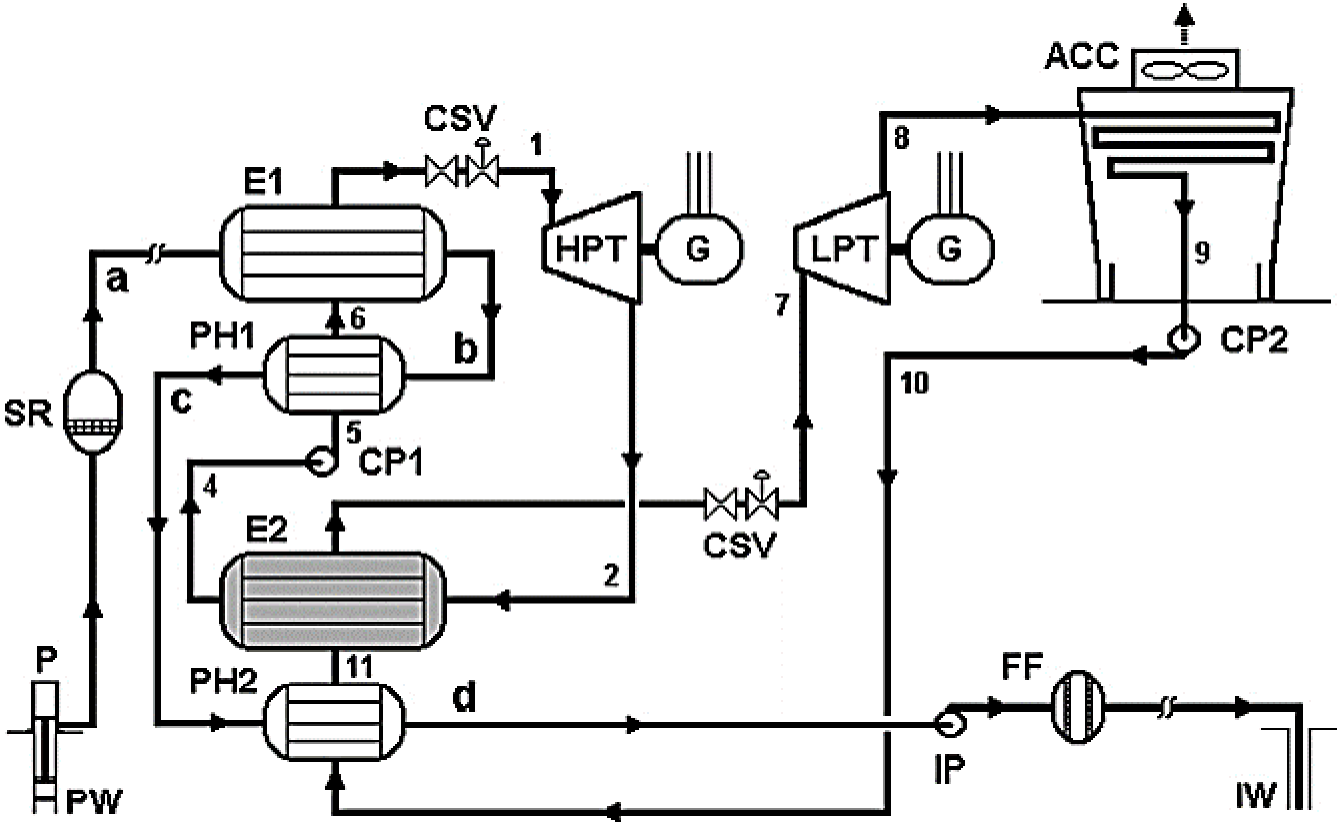


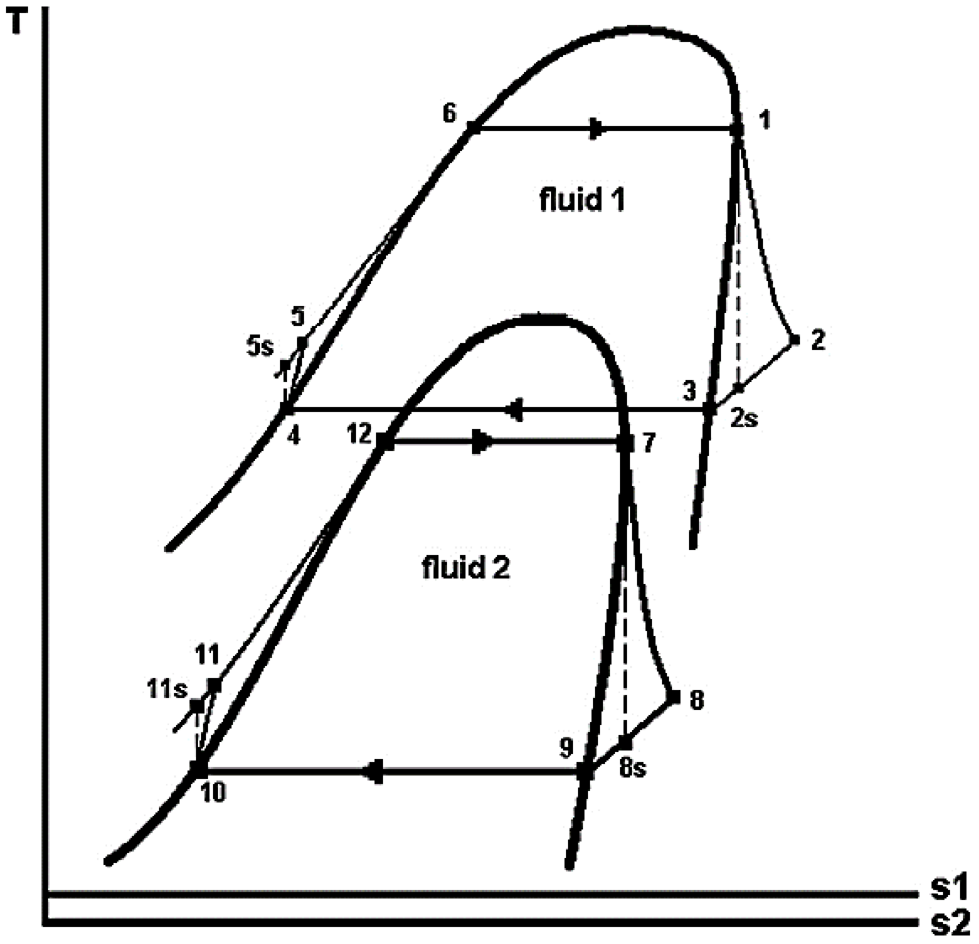
TABLE 8.5 Comparison of efficiencies of single- and dual-pressure binary cycles [17].

Working fluid	Brine temperature	Thermal efficiency, %		Utilization efficiency, %	
		Basic	Dual pressure	Basic	Dual pressure
i-C_4H_{10}	93°C (200°F)	5.5	4.6	31.9	39.7
i-C_5H_{12}	93°C (200°F)	5.2	4.2	30.5	37.0
i-C_4H_{10}	149°C (300°F)	10.3	9.8	48.8	56.9
i-C_5H_{12}	149°C (300°F)	9.8	8.8	44.6	51.5
i-C_5H_{12}	204°C (400°F)	13.7	13.1	57.7	61.2

Note: The condensing and dead-state temperatures were both taken as 38°C (100°F).

Advanced Binary Cycle: Dual Fluid BC





Advanced Binary Cycle: Dual Fluid BC

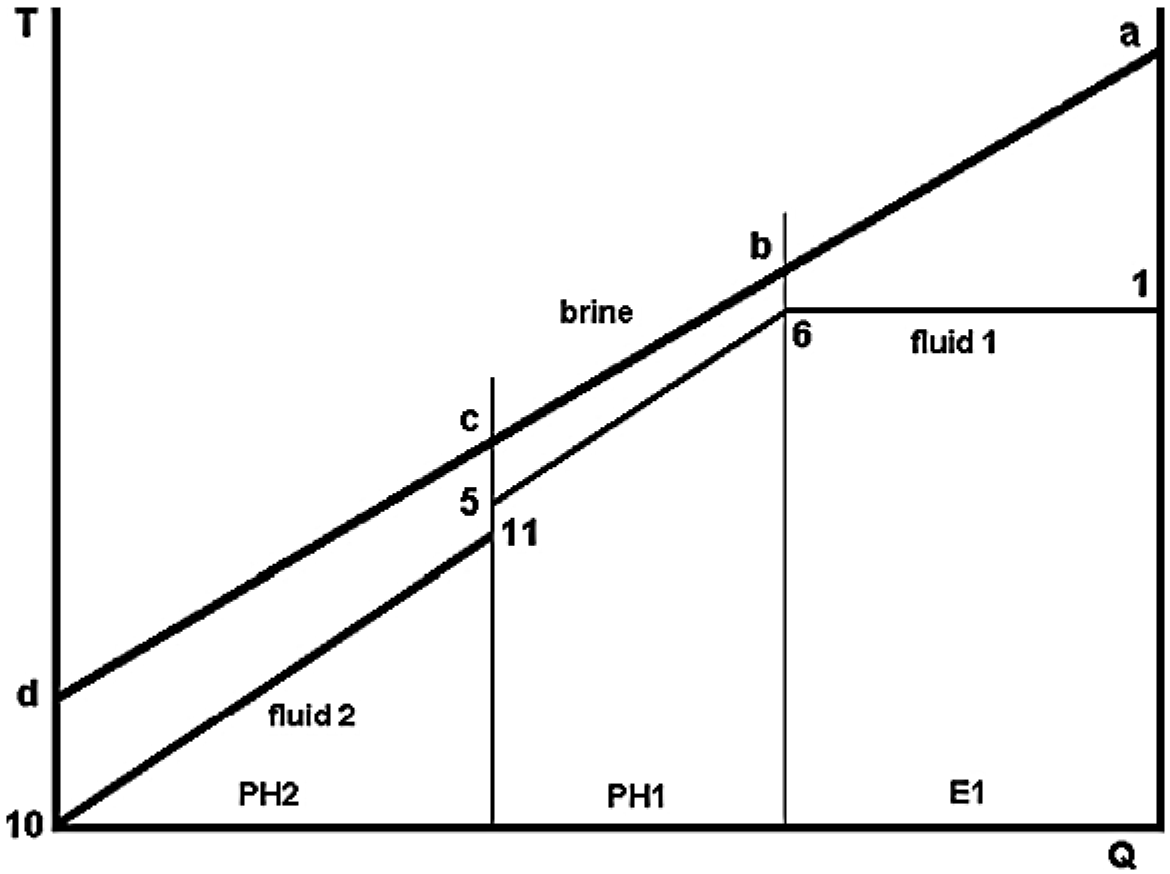


Figure 8.16 Dual-fluid binary plant: Temperature-heat transfer diagram for brine heat exchangers with subcritical working fluid pressures.

Advanced Binary Cycle: Dual Fluid BC

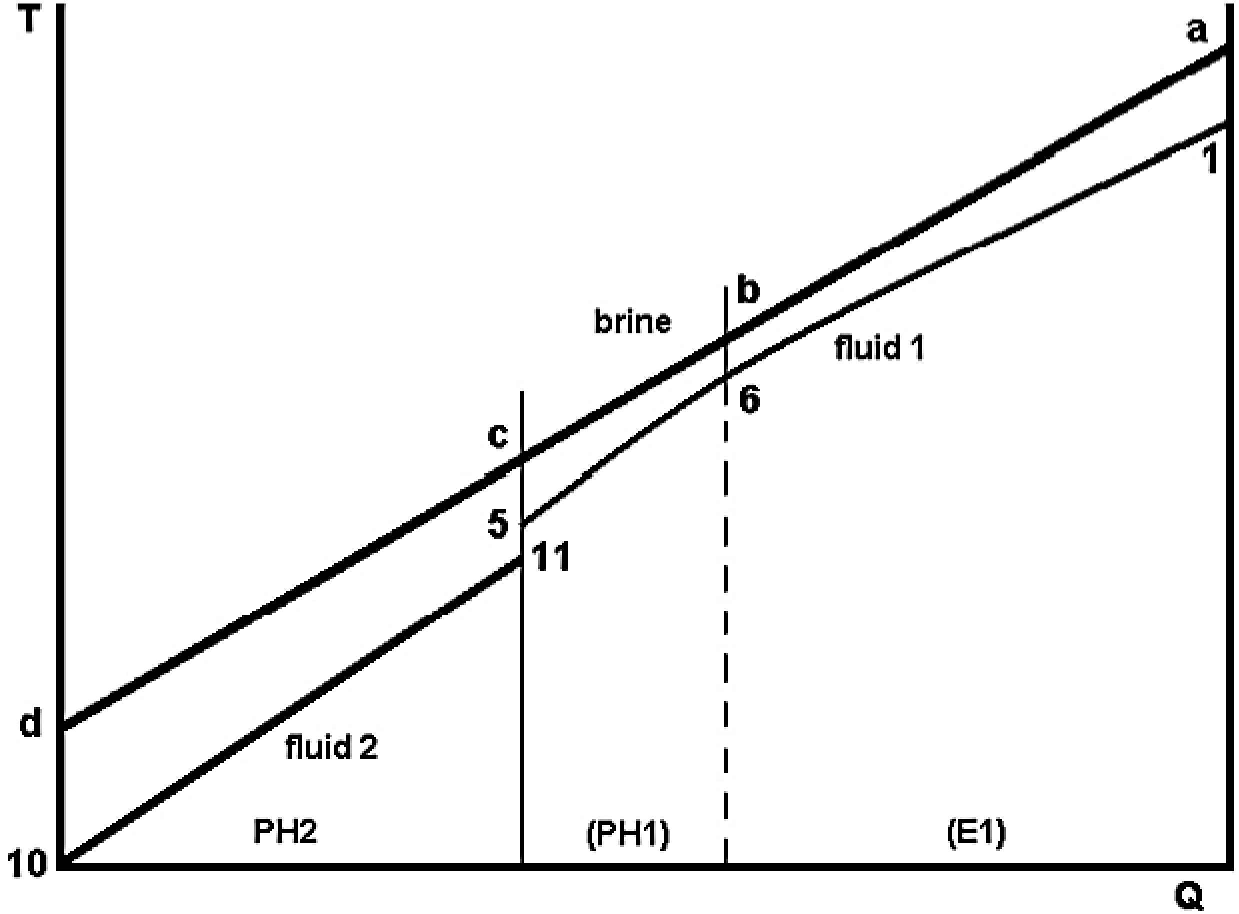


Figure 8.17 Dual-fluid binary plant: Temperature-heat transfer diagram for brine heat exchangers with supercritical pressure for working fluid 1.

Advanced Binary Cycle: Kalina BC

- The working fluid is a binary mixture of H_2O and NH_3
- Evaporation and condensation occur at variable temperature
- Cycle incorporates heat recuperation from turbine exhaust
- Composition of the mixture may be varied during cycle in some versions

- Improved thermodynamic performance of heat exchangers by reducing the irreversibilities associated with heat transfer
- Better match between the brine and the mixture.

Advanced Binary Cycle: Kalina BC

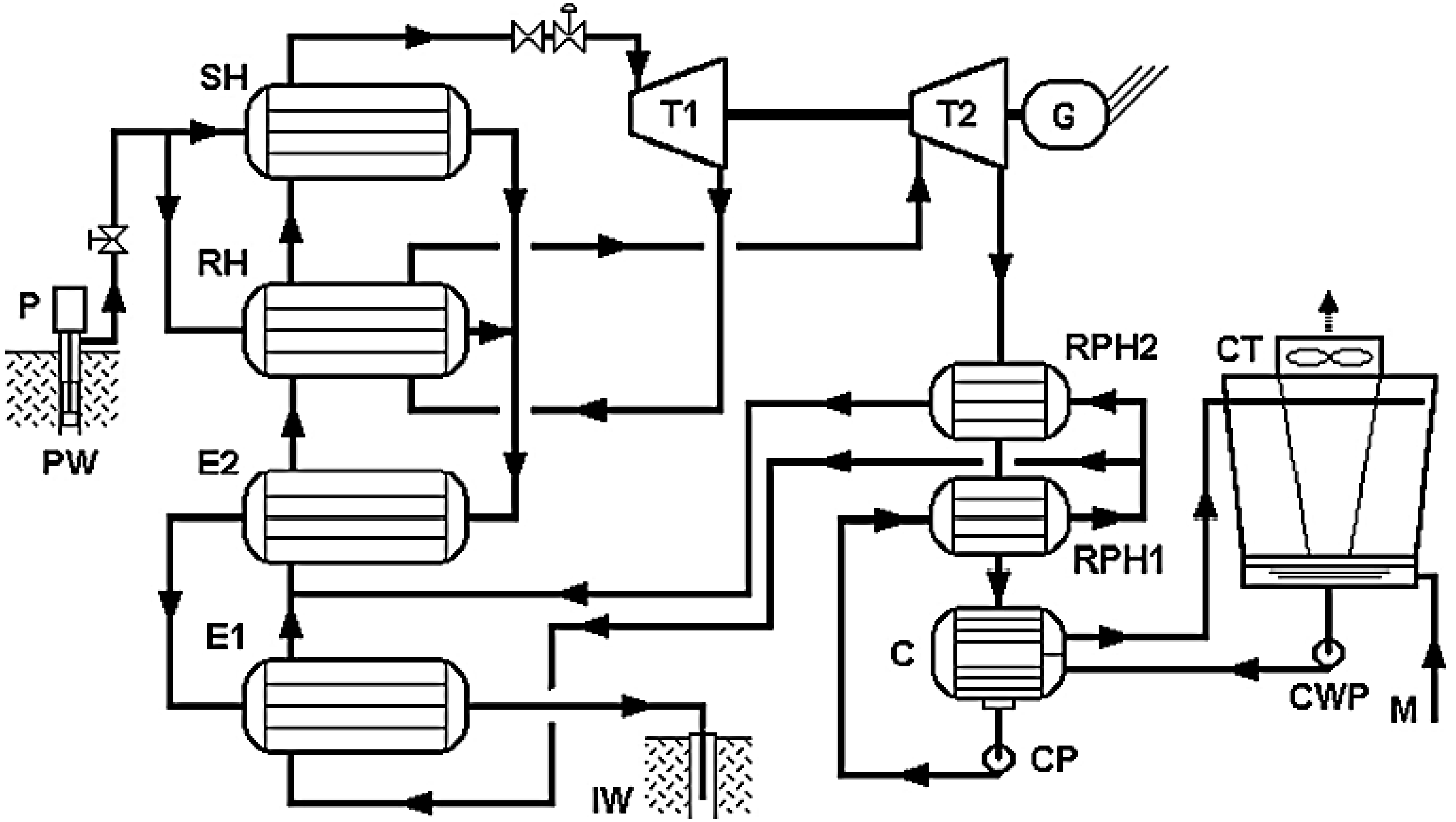
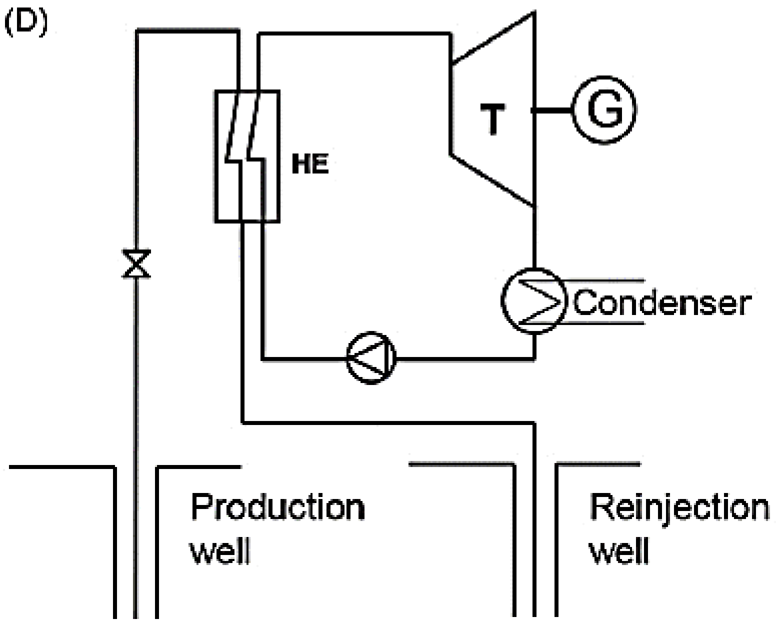


Figure 8.18 Typical Kalina cycle employing a reheater and two recuperative preheaters.

Binary Cycle (ORC)



Operator	Plant	Plant Type	Year	No. of Units[1]	Net Rating MW[2]	Gross Rating MW
Mammoth-Pacific	MP-1	Binary	1984	2	7	10
Mammoth-Pacific	MP-2	Binary	1990	3	10	15
Mammoth-Pacific	PLES-1	Binary	1990	3	10	15

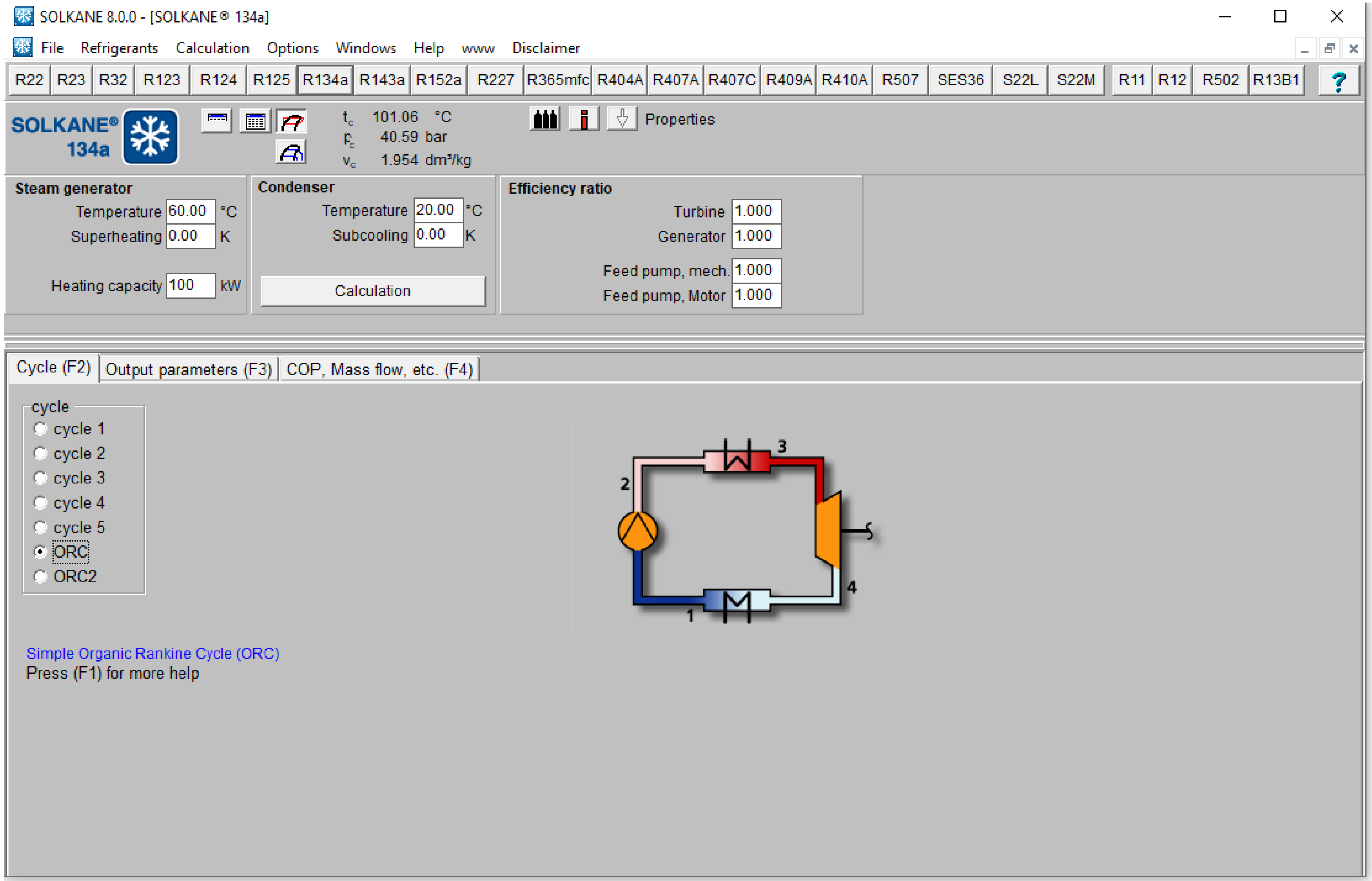
Binary Cycle (74ºC !)

400 kW

1: Hot water at **74 ºC** (57 ºC)
2: R-134a
3: Turbine (13500 rpm)
4: Cold water at **4 ºC** (9 ºC)
5: R 134a
6: Pump

Binary Cycle (ORC)

Binary Cycle (ORC)



t_c 101.06 °C
p_c 40.59 bar
v_c 1.954 dm³/kg

Steam generator
Temperature 60.00 °C
Superheating 0.00 K
Heating capacity 100 kW

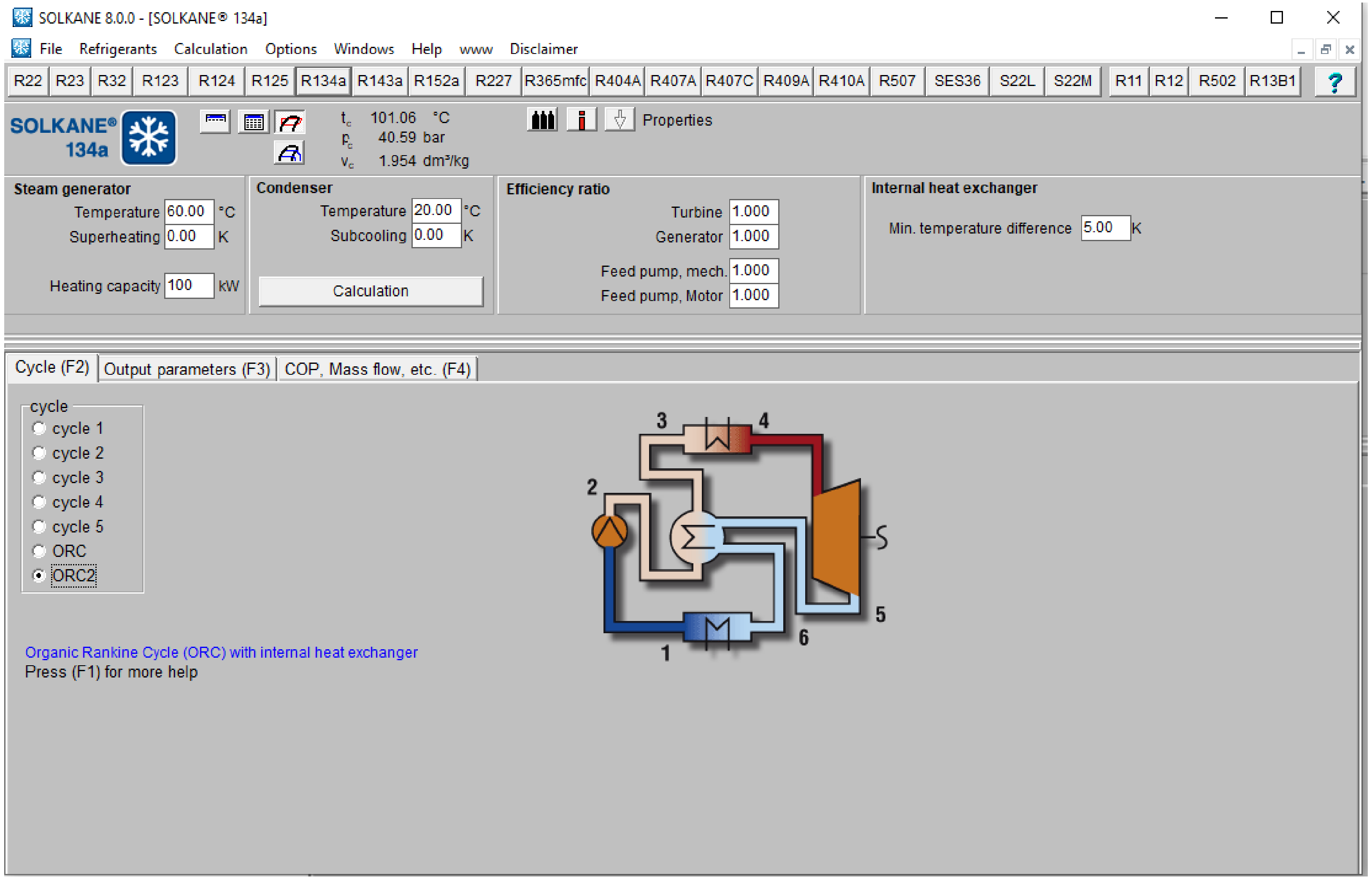
Condenser
Temperature 20.00 °C
Subcooling 0.00 K

Efficiency ratio
Turbine 1.000
Generator 1.000
Feed pump, mech. 1.000
Feed pump, Motor 1.000

cycle
- ○ cycle 1
- ○ cycle 2
- ○ cycle 3
- ○ cycle 4
- ○ cycle 5
- ● ORC
- ○ ORC2

Simple Organic Rankine Cycle (ORC)
Press (F1) for more help

Binary Cycle (ORC)



t_c 101.06 °C
p_c 40.59 bar
v_c 1.954 dm³/kg

Steam generator
Temperature 60.00 °C
Superheating 0.00 K
Heating capacity 100 kW

Condenser
Temperature 20.00 °C
Subcooling 0.00 K

Efficiency ratio
Turbine 1.000
Generator 1.000
Feed pump, mech. 1.000
Feed pump, Motor 1.000

Internal heat exchanger
Min. temperature difference 5.00 K

cycle
- ○ cycle 1
- ○ cycle 2
- ○ cycle 3
- ○ cycle 4
- ○ cycle 5
- ○ ORC
- ● ORC2

Organic Rankine Cycle (ORC) with internal heat exchanger
Press (F1) for more help

Environmental Impact

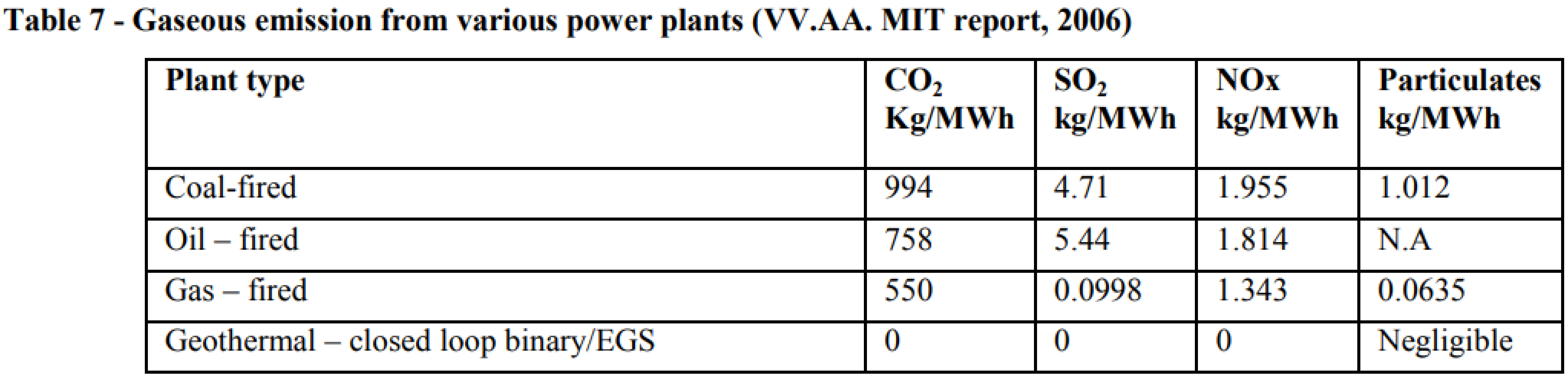


Table 7 - Gaseous emission from various power plants (VV.AA. MIT report, 2006)

Plant type	**CO_2 Kg/MWh**	**SO_2 kg/MWh**	**NOx kg/MWh**	**Particulates kg/MWh**
Coal-fired	994	4.71	1.955	1.012
Oil – fired	758	5.44	1.814	N.A
Gas – fired	550	0.0998	1.343	0.0635
Geothermal – closed loop binary/EGS	0	0	0	Negligible

https://www.geothermal-energy.org/pdf/IGAstandard/WGC/2015/37008.pdf

Geothermal Energy Capacity Building in Egypt (GEB)

Geothermal Power Plants

Co-funded by the
Erasmus+ Programme
of the European Union

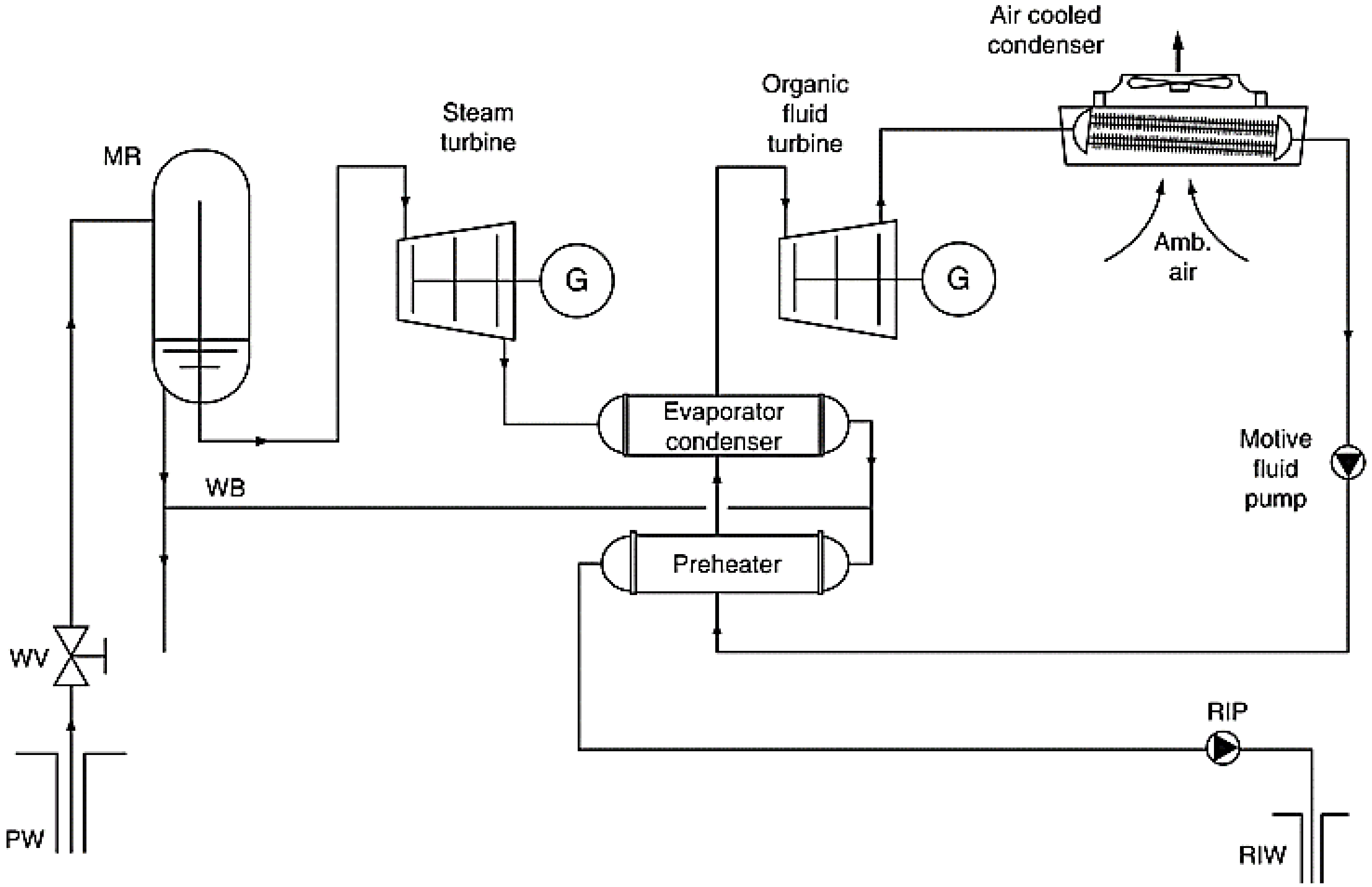
GEB

Geothermal Energy Capacity Building in Egypt (GEB)

Advanced geothermal energy conversion systems
Hybrid geothermal power systems

Co-funded by the
Erasmus+ Programme
of the European Union

Hybrid Flash-Binary Systems



Hybrid Flash - Binary Systems

Rotokawa (New Zealand)

- 4 extraction wells
- 5 reinjection wells
- 1 backpressure turbine (16 MW)
- 4 binary cycles (6.5 MW each)

Solar-Augmented Binary Plants

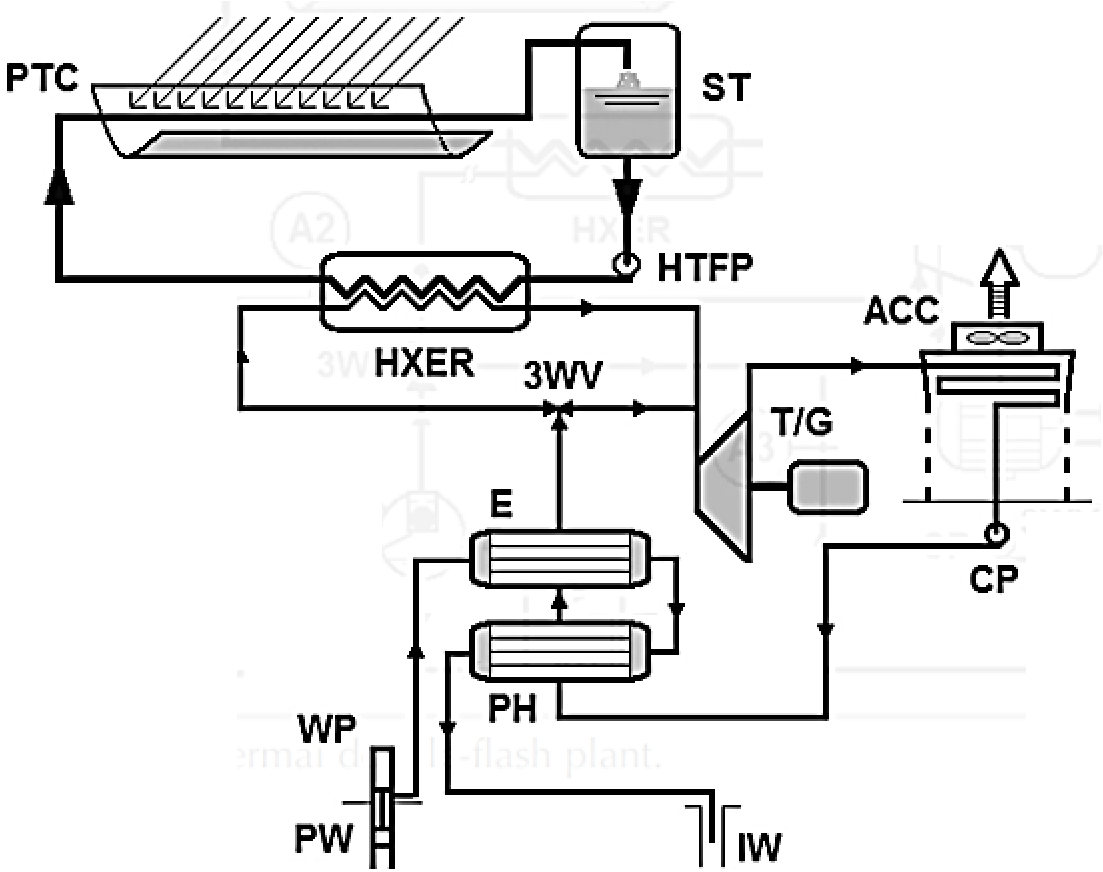


Figure 9.26 Solar-geothermal binary plant with superheating of the binary working fluid.

Hot Dry Rock (HDR)
Enhanced Geothermal Systems (EGS)

Formations with high temperatura and…

Low permeability and lack of water.

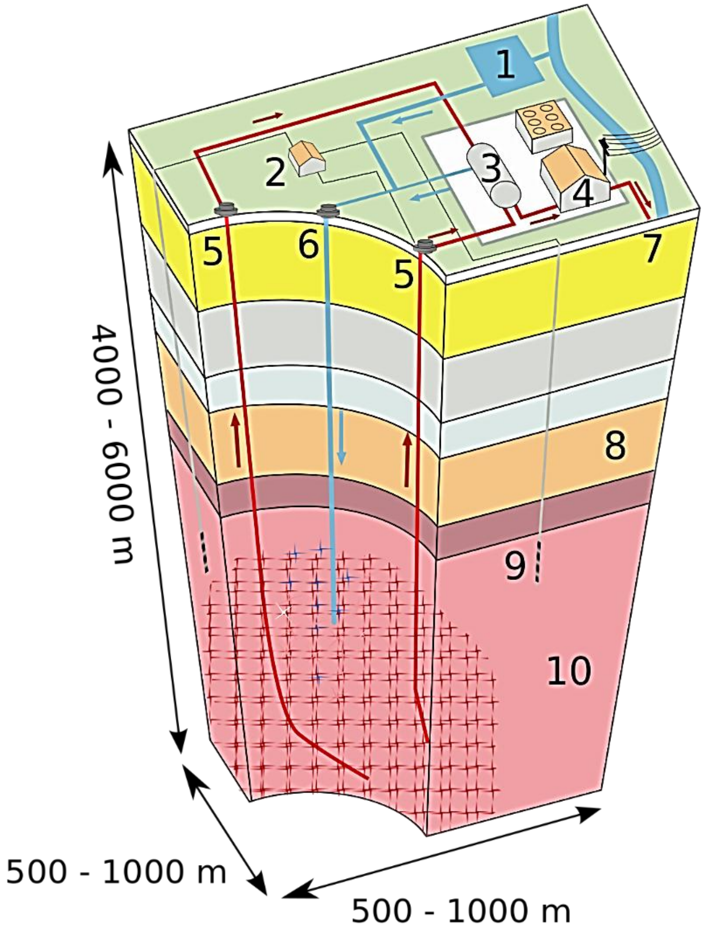
Use: EGS

Experimental phase

Hydraulic fracturing is needed to increase permeability.

Fluid is injected and circulated.

Enhanced Geothermal Systems (EGS)



Enhanced Geothermal Systems (EGS)

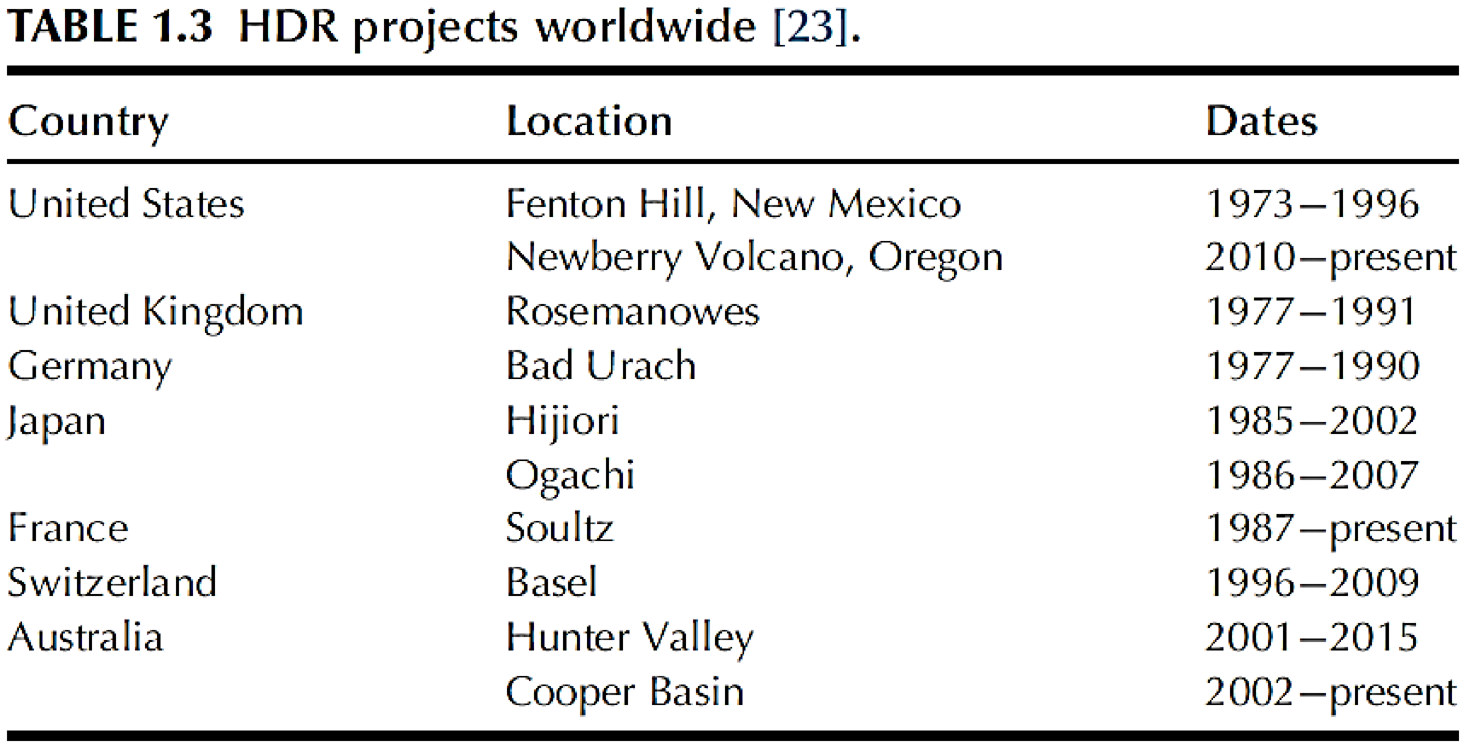
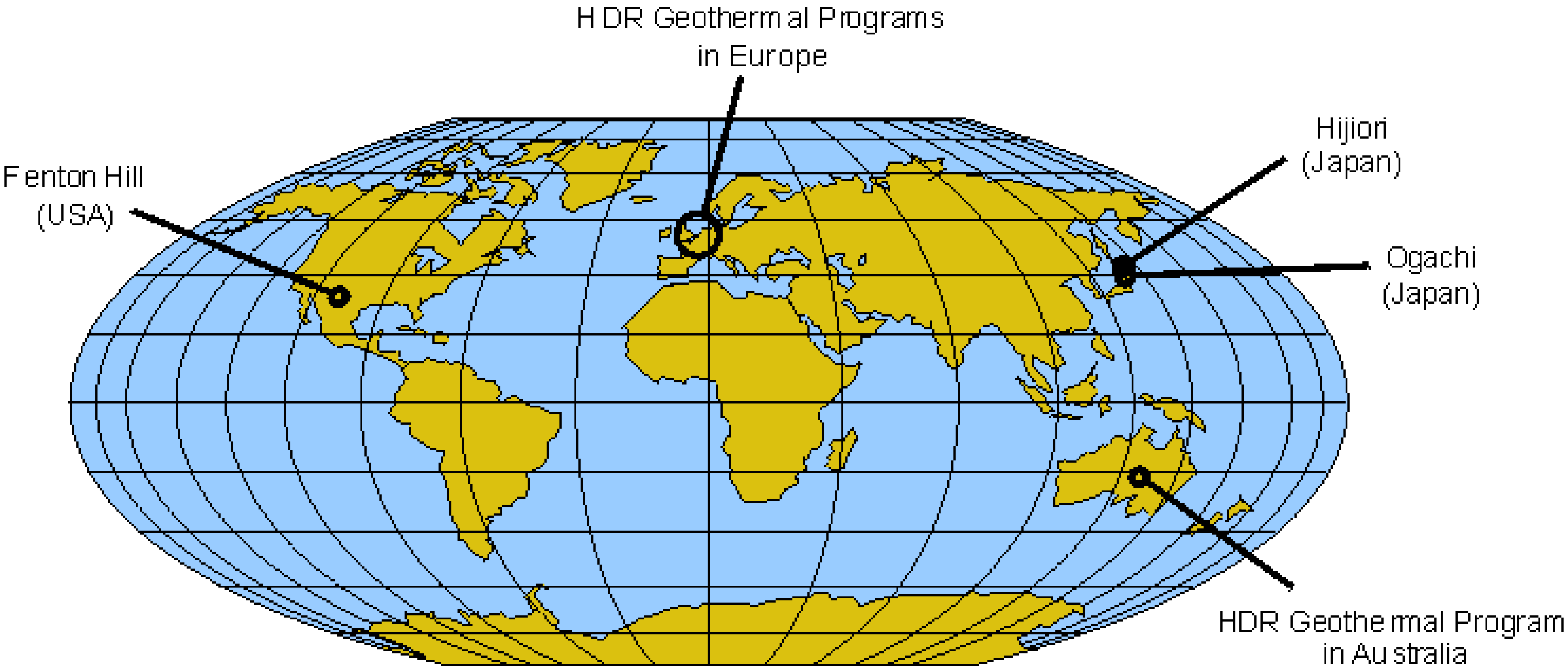


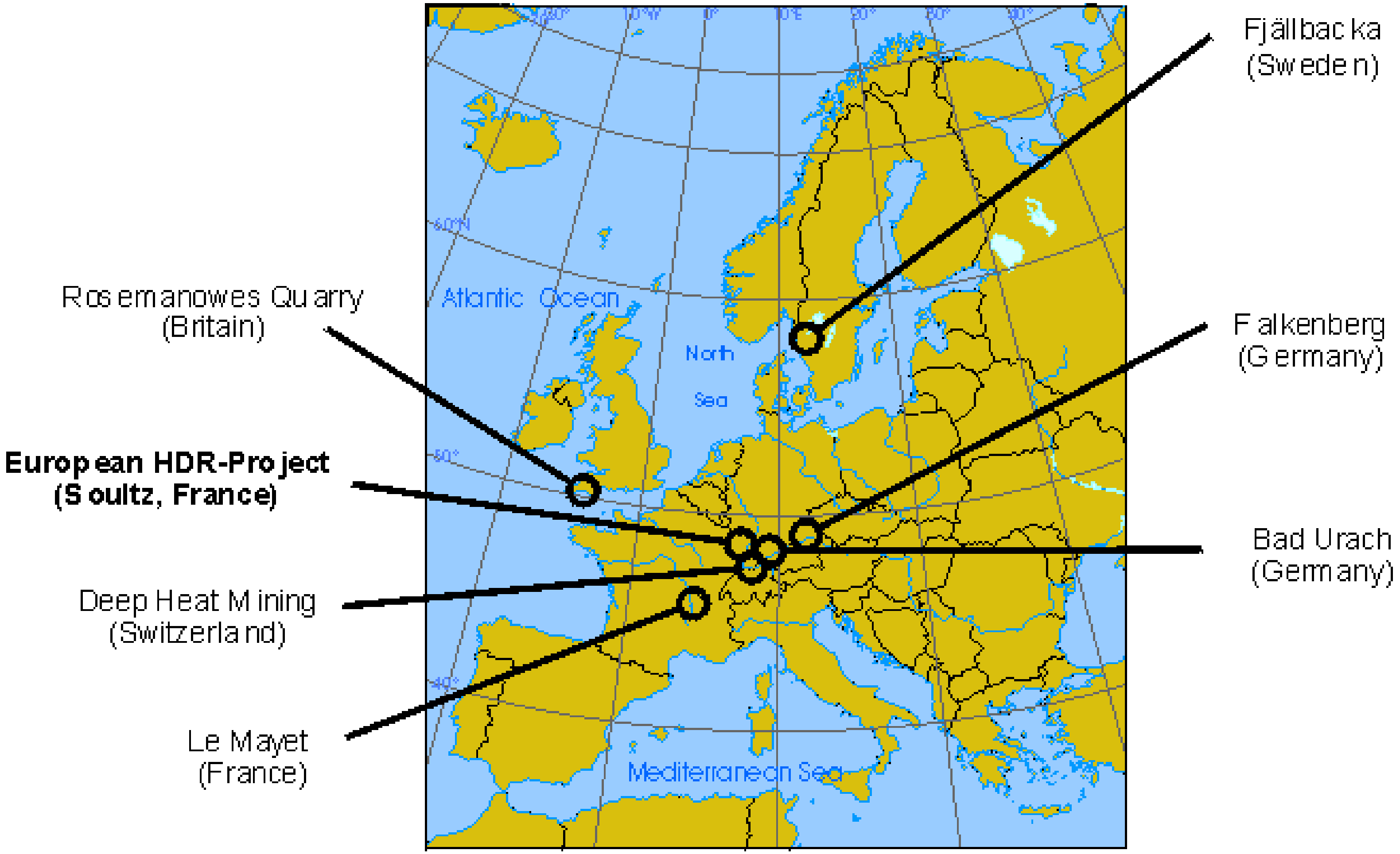
TABLE 1.3 HDR projects worldwide [23].

Country	Location	Dates
United States	Fenton Hill, New Mexico	1973–1996
	Newberry Volcano, Oregon	2010–present
United Kingdom	Rosemanowes	1977–1991
Germany	Bad Urach	1977–1990
Japan	Hijiori	1985–2002
	Ogachi	1986–2007
France	Soultz	1987–present
Switzerland	Basel	1996–2009
Australia	Hunter Valley	2001–2015
	Cooper Basin	2002–present

Enhanced Geothermal Systems (EGS)



Enhanced Geothermal Systems (EGS)



EGS (Soultz-sous-Forêts)

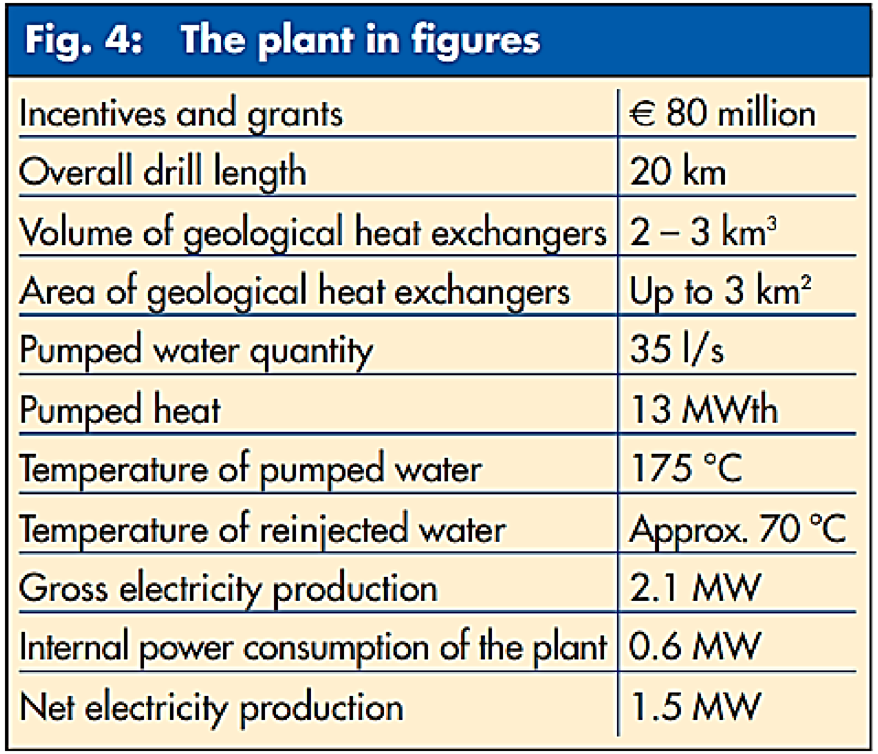


Fig. 4: The plant in figures

Incentives and grants	€ 80 million
Overall drill length	20 km
Volume of geological heat exchangers	2 – 3 km^3
Area of geological heat exchangers	Up to 3 km^2
Pumped water quantity	35 l/s
Pumped heat	13 MWth
Temperature of pumped water	175 °C
Temperature of reinjected water	Approx. 70 °C
Gross electricity production	2.1 MW
Internal power consumption of the plant	0.6 MW
Net electricity production	1.5 MW

http://www.bine.info/fileadmin/content/Publikationen/Englische_Infos/projekt_0409_engl_Internetx.pdf

EGS (Soultz-sous-Forêts)

Fig. 2: The temperature gradient in Soultz

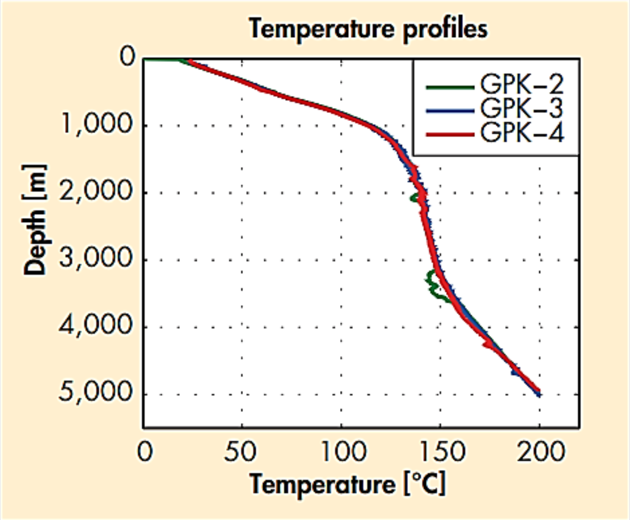


Fig. 3: Position of the various boreholes

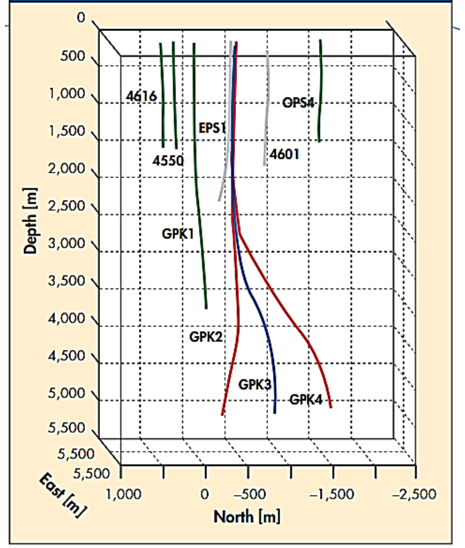
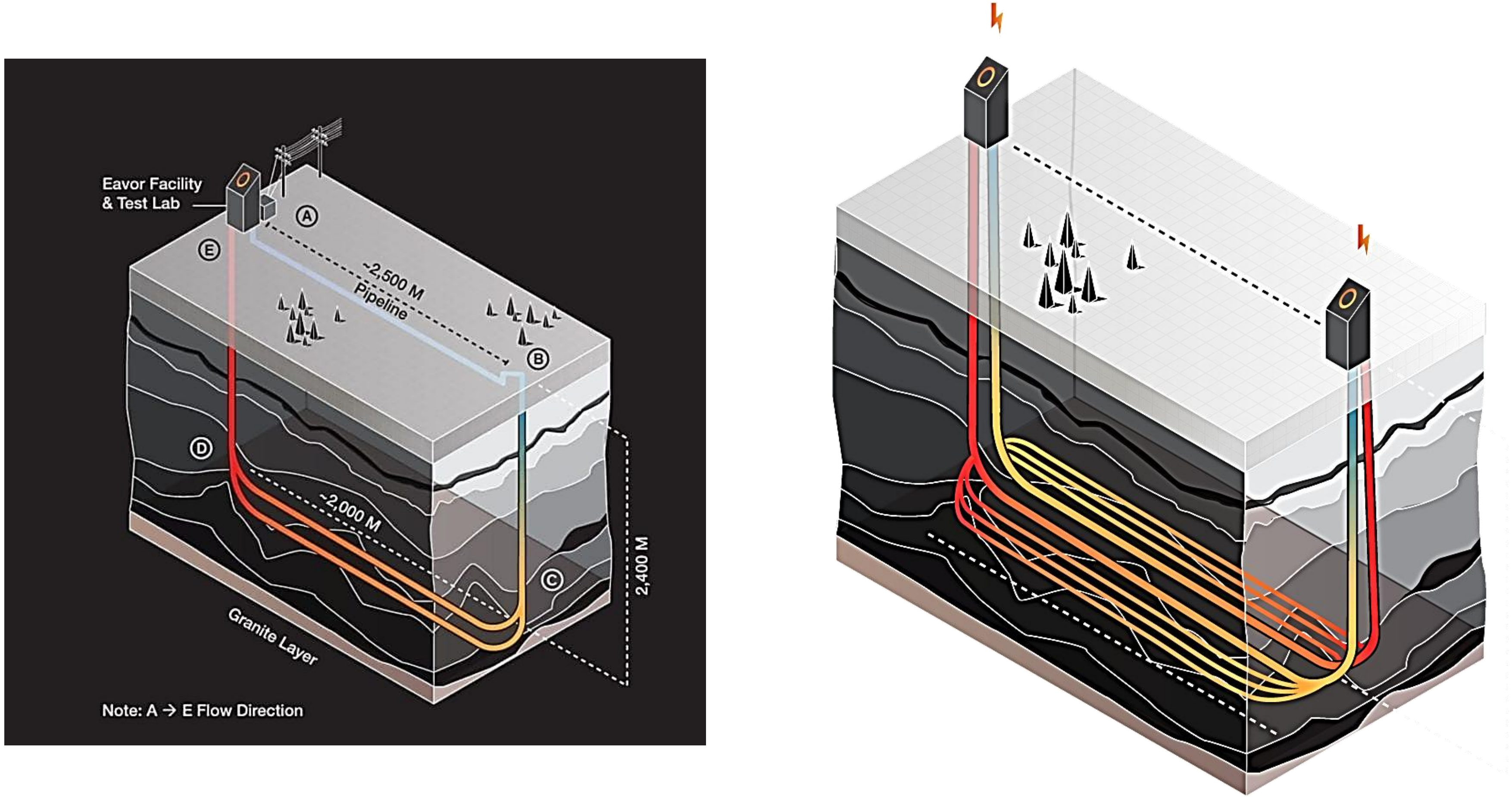


Fig. 9: Power plant and systems above ground

http://www.bine.info/fileadmin/content/Publikationen/Englische_Infos/projekt_0409_engl_Internetx.pdf

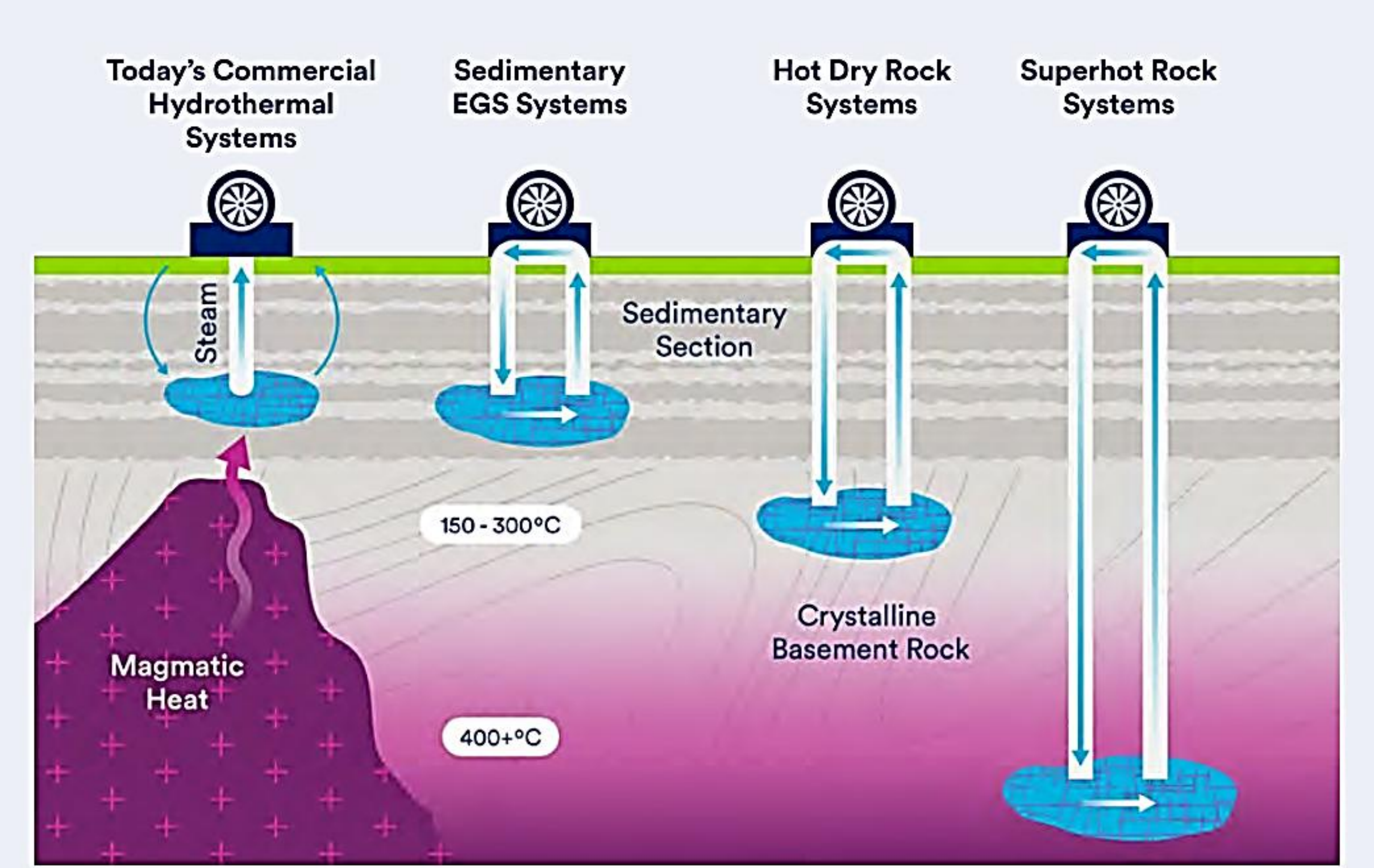
EGS (Eavor-Loop)



Advanced geothermal system technologies focus on using horizontal drilling techniques to drill small, sealed horizontal boreholes between wells to allow for circulation of fluids that bring the heat to the surface. One example is Eavor Technologies' "Eavor-Loop" system, which completed a successful demonstration at the Eavor-Lite facility in Alberta, Canada, in February 2020. Courtesy: Eavor

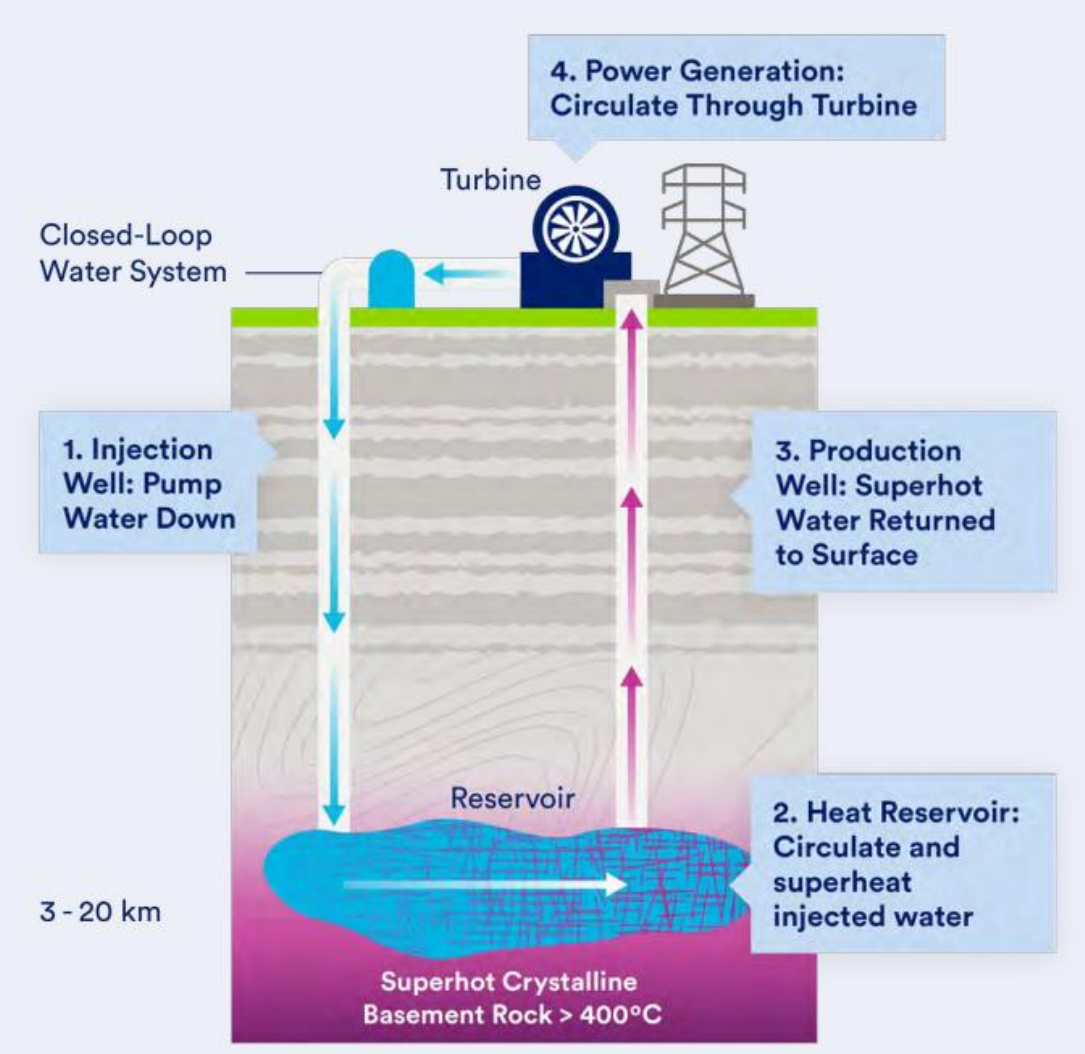
Superhot Rock Systems

https://www.quaise.energy/



https://www.catf.us/wp-content/uploads/2021/09/CATF_SuperhotRockGeothermal_Report.pdf

Superhot Rock Systems



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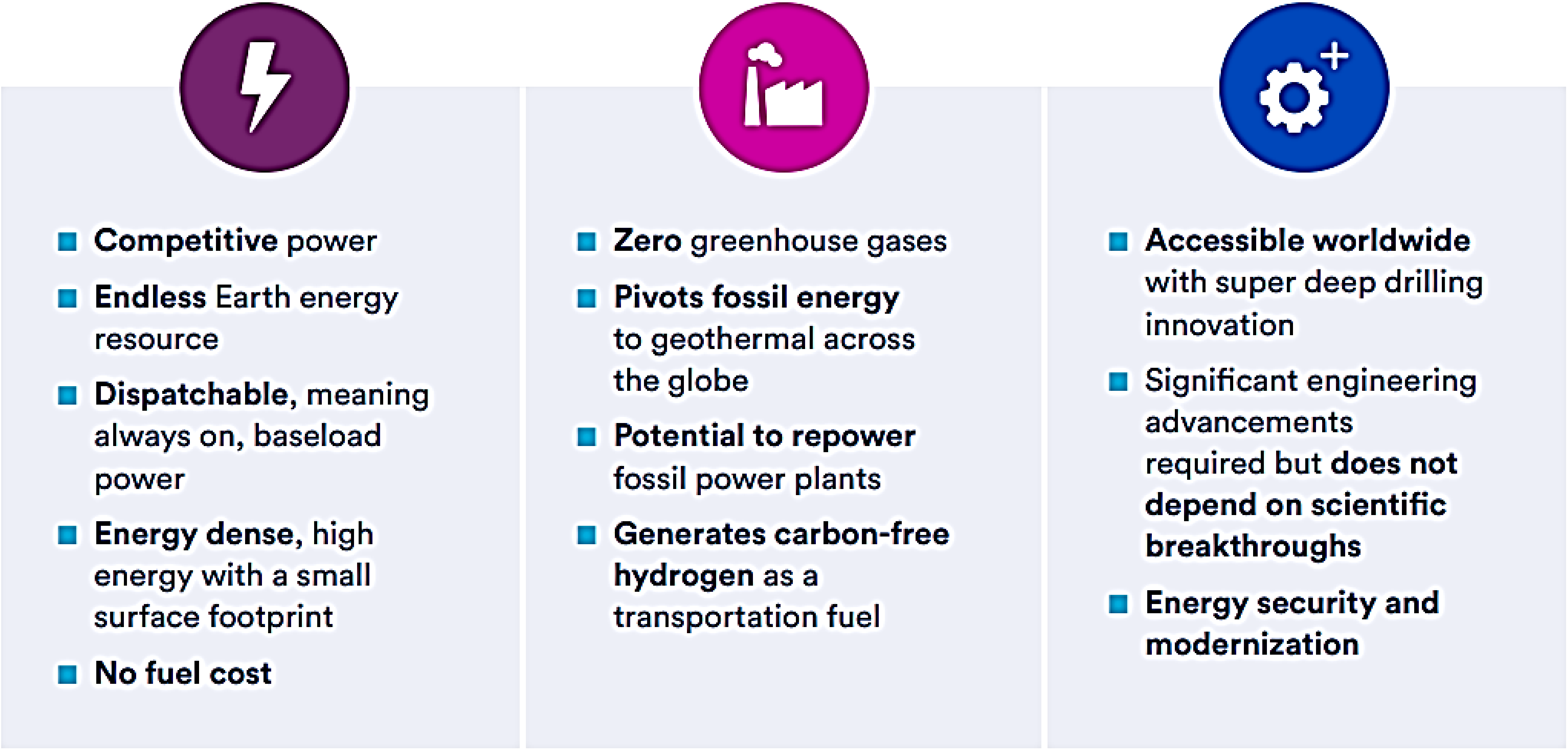
Superhot Rock Systems



- **Competitive** power
- **Endless** Earth energy resource
- **Dispatchable,** meaning always on, baseload power
- **Energy dense,** high energy with a small surface footprint
- **No fuel cost**

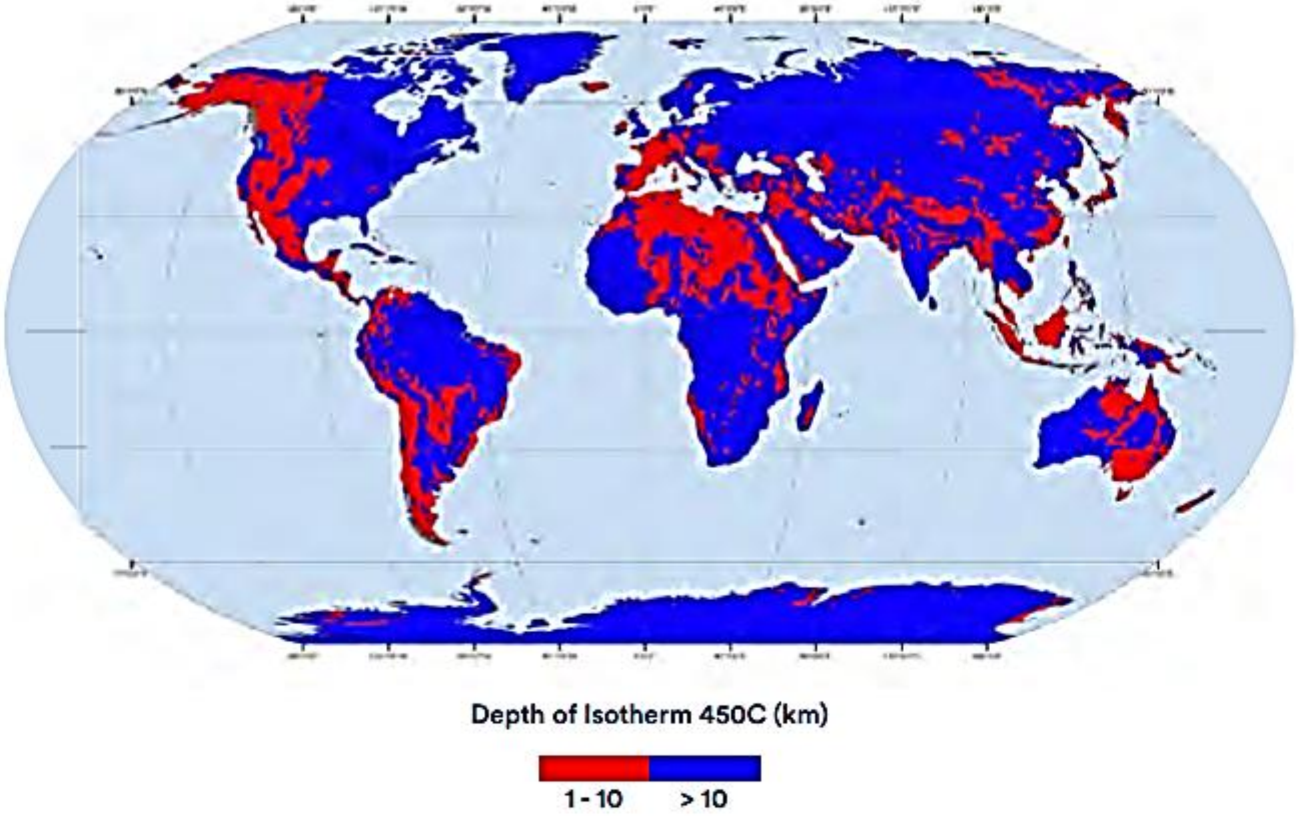
- **Zero** greenhouse gases
- **Pivots fossil energy** to geothermal across the globe
- **Potential to repower** fossil power plants
- **Generates carbon-free hydrogen** as a transportation fuel

- **Accessible worldwide** with super deep drilling innovation
- Significant engineering advancements required but **does not depend on scientific breakthroughs**
- **Energy security and modernization**

https://www.catf.us/wp-content/uploads/2021/09/CATF_SuperhotRockGeothermal_Report.pdf

Superhot Rock Systems

Why superhot geothermal matters: areas shaded red are superhot rock resources >450°C that are less than 10 km in depth and may be accessible with enhanced mechanical drilling methods. Energy drilling could reach depths beyond 10 km in the blue regions.



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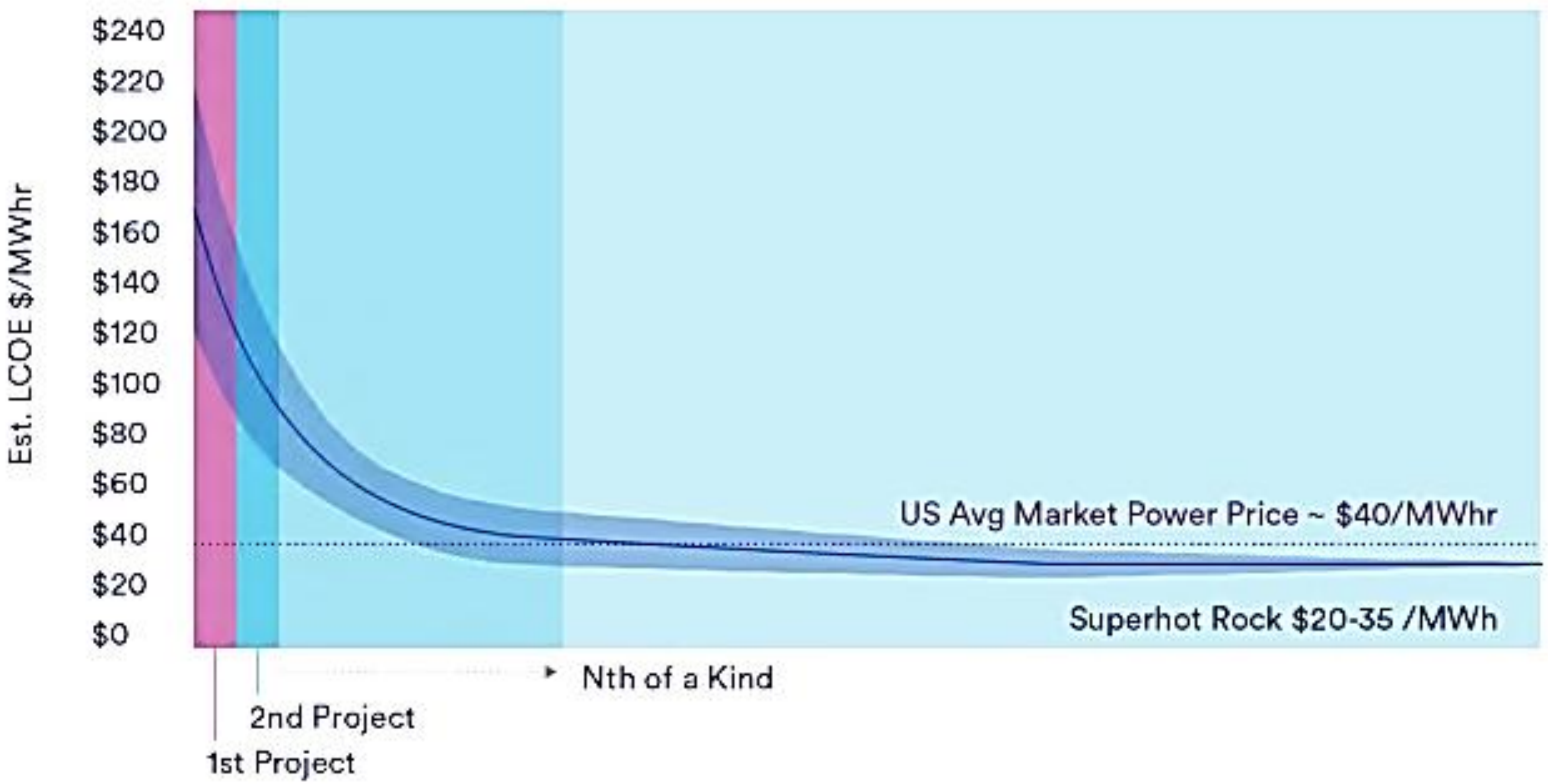
Superhot Rock Systems

Illustrative graph shows how electricity produced from superhot rock is expected to be competitive for “Nth of a kind” plants (levelized cost of electricity after full commercialization). Lucid Catalyst and Hot Rock Energy Research Organization (HERO) have preliminarily estimated that superhot rock geothermal could have an LCOE in the range of $0.02-$0.035 / kWh. This would be competitive with other dispatchable and intermittent energy resources.



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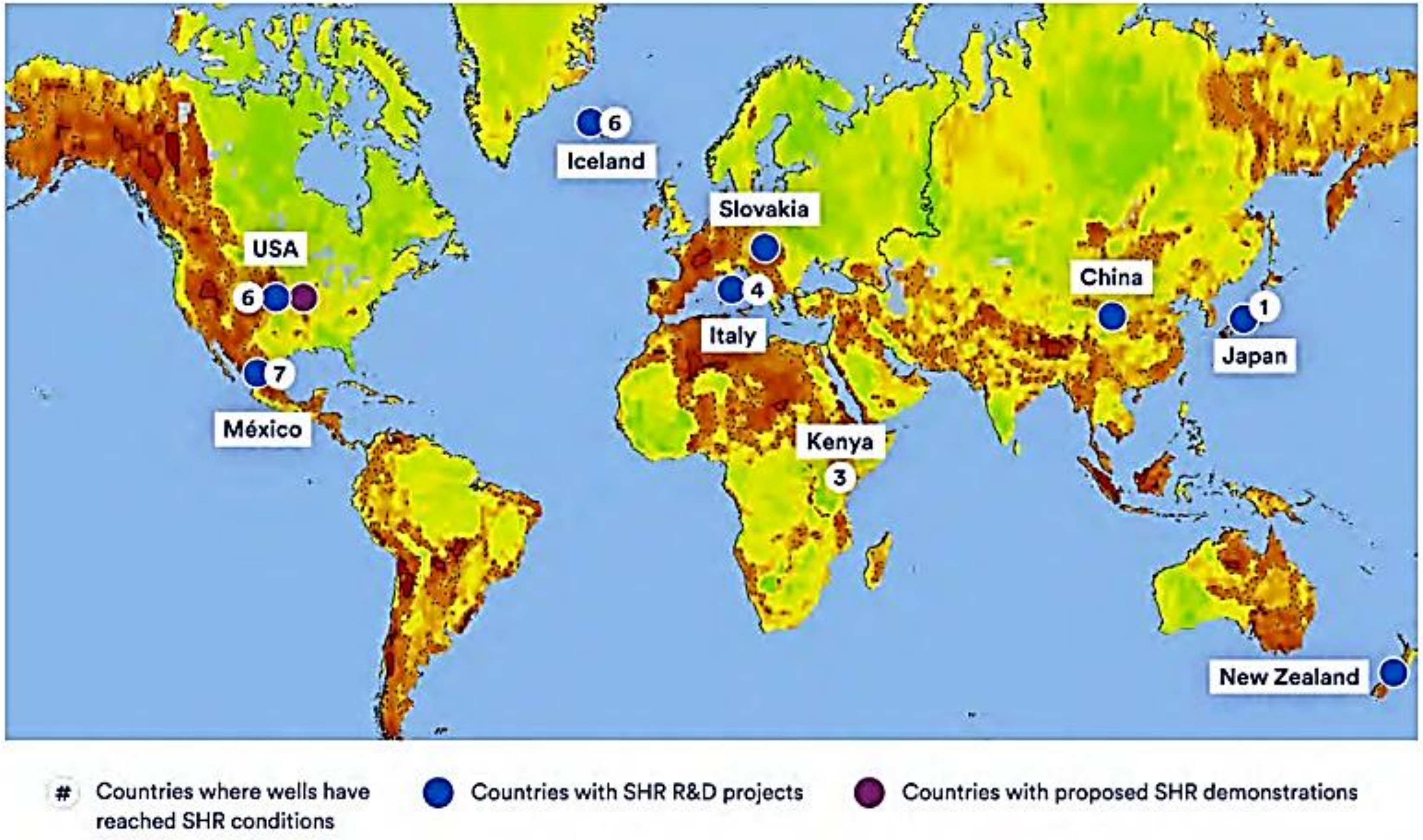
Superhot Rock Systems



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Superhot Rock Systems

- **Japan Beyond Brittle Project**. 1994-1995, reaching to the "brittle-ductile transition zone" where rock is more plastic at temperatures above 500°C at a depth of 3.7 km

- **Iceland Deep Drilling Project**. First test well, IDDP-1 Krafla, completed in 2009, after drilling was terminated when it encountered magma. Projected energy flow of 36 MWe. The second well, IDDP-2 Reykjanes, reached its objective of supercritical (superhot) conditions at 426°C in 2017. IDDP is currently planning a third superhot well.

- **DESCRAMBLE**. Larderello (Italy) EU project 2015-18 to drill into superhot. Larderello's Venelle-2 is the hottest geothermal well on record, registering 514°C at a depth of 2.9 km

- **GEMex**. EU-supported program focused on HDR/EGS development and SHR systems. It drilled several wells at the Acoculco geothermal field, reaching "well above" 300°C in dry wells

- **Hotter and Deeper**. New Zealand. Since 2009 The project hopes to investigate potential reservoir systems in the superhot plastic brittle-ductile transition zone at about 7 km where geophysics suggests there is little seismic activity

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