

# THERMODYNAMIC CYCLES FOR RENEWABLE ENERGY

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# **LECTURE OUTLINE**

- Power Cycles in Renewable Energy
- The Organic Rankine Cycle (ORC)
- How to Design an ORC?
- Geothermal Application
- Solar Rankine Cycle
- Biomass Power Plant
- Ocean Thermal Energy Conversion (OTEC)

# Applications



## Biogas, Landfill gas

- Enhancement of biogas engine via exhaust, water jacket or both
- Direct biogas to electricity conversion with boiler



## Solar

- Solar CHP with CSP field
- Solar CHP with CSP and heat storage



## Biomass

- Biomass to electricity
- Biomass CHP
- Isolated site



## Geothermal

- Natural hot sources
- Medium temperature wells (from 80 °C)

# Renewable Energies



## Industrial Waste Heat Recovery

- Process Heat
- Exhaust gases
- Waste steam



## Diesel genset

- Efficiency enhancement via exhaust, water jacket or both



## Transportation

- Sea and River transportation
- Railroad
- Heavy Duty Trucks

# Energy Efficiency

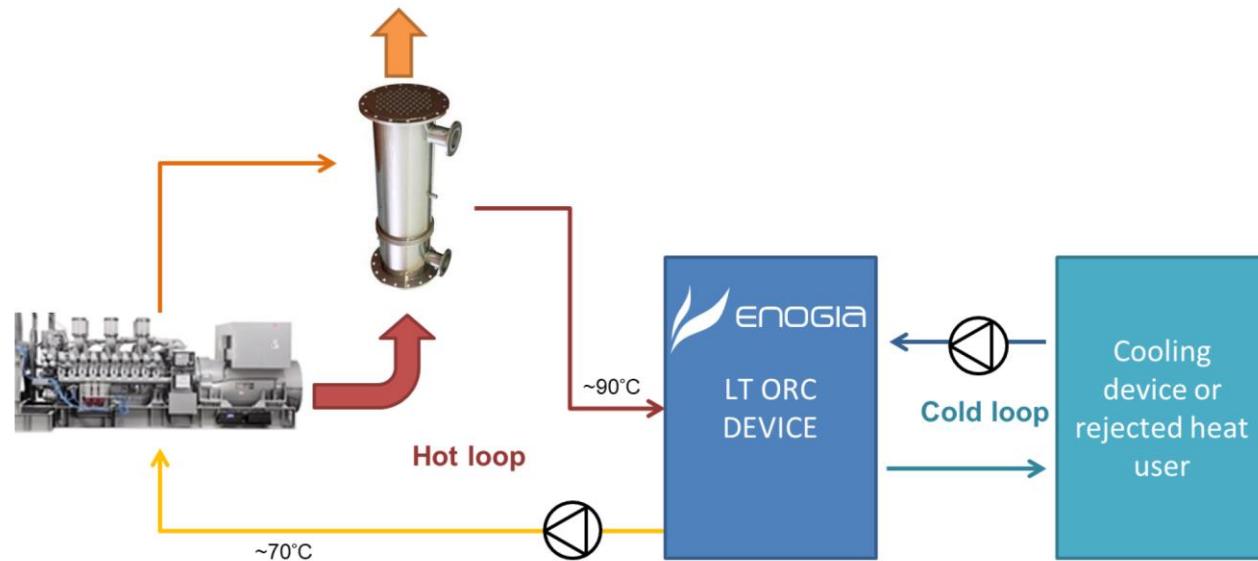


## Educational and research

- With boiler simulating heat source

## Applications

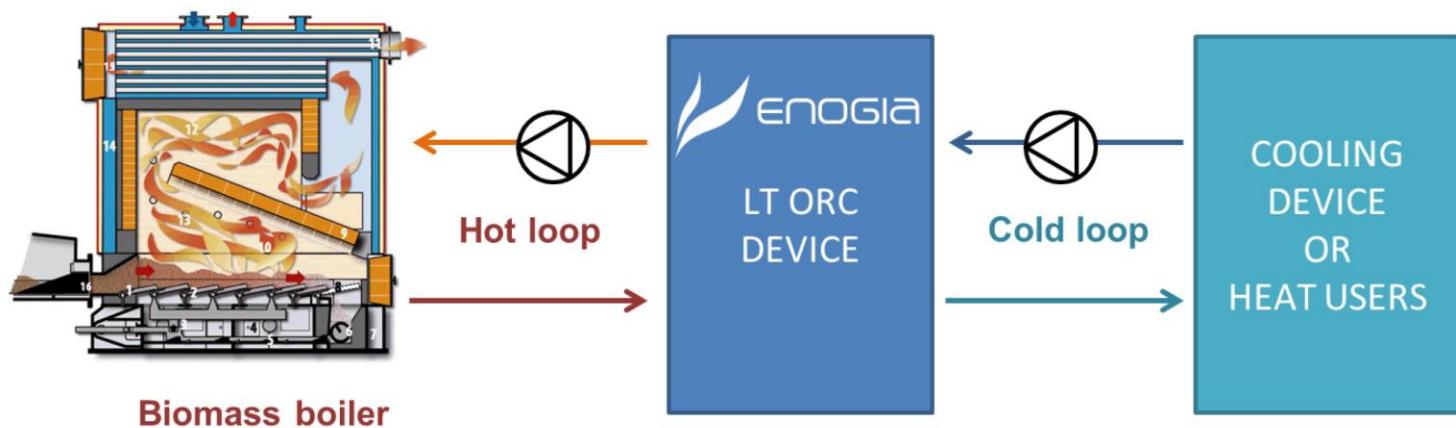
ORC is used to enhance Biogas CHP efficiency using both jacket water and exhaust heat recovery :



Combined Heat and Power (CHP) Application

## Applications

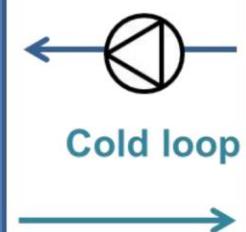
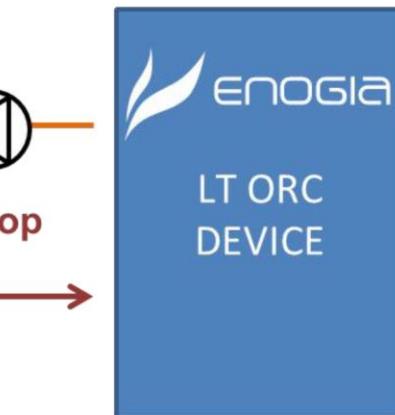
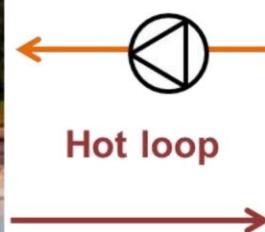
ORC is used to generate electricity from biomass boiler heat or as a bottoming cycle



## Applications

ORC is used to produce electricity from geothermal natural source or geothermal well :

**Hot water from geothermal natural or enhanced source**



# The Organic Rankine Cycle (ORC)

Organic Rankine Cycle systems for large-scale waste heat recovery to produce electricity. Organic Rankine Cycle

The Organic Rankine Cycles are variants of the water steam cycles, which are used when the hot source is at low or medium temperature.

Under these conditions their performance of the water steam cycles deteriorates, it becomes preferable to use other thermodynamics fluids.

These fluids are derived from organic chemistry (carbonà and can be hydrocarbons, oils, refrigerants,...

# Simple Organic Rankine Cycle & Working Fluids

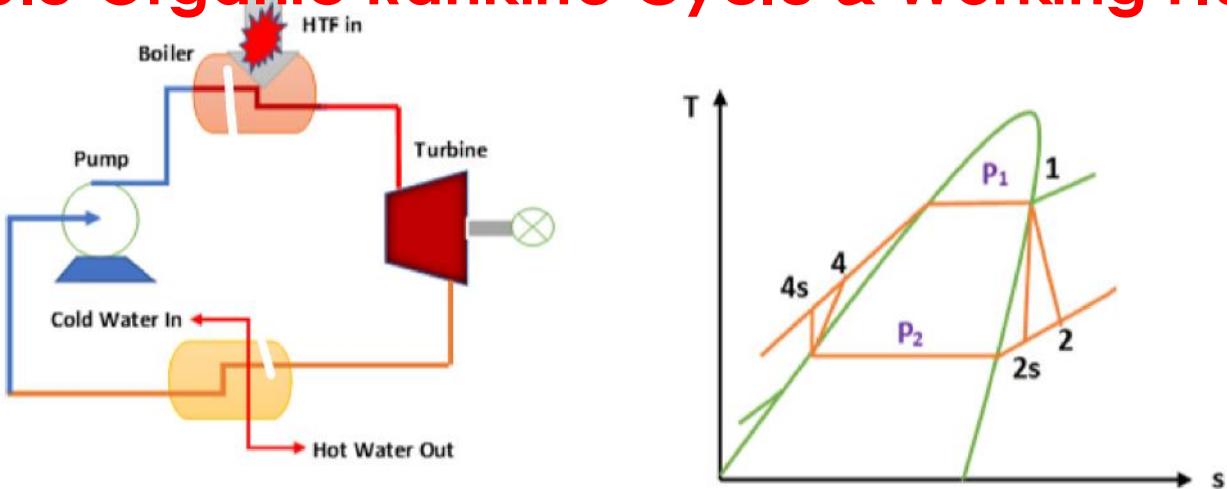


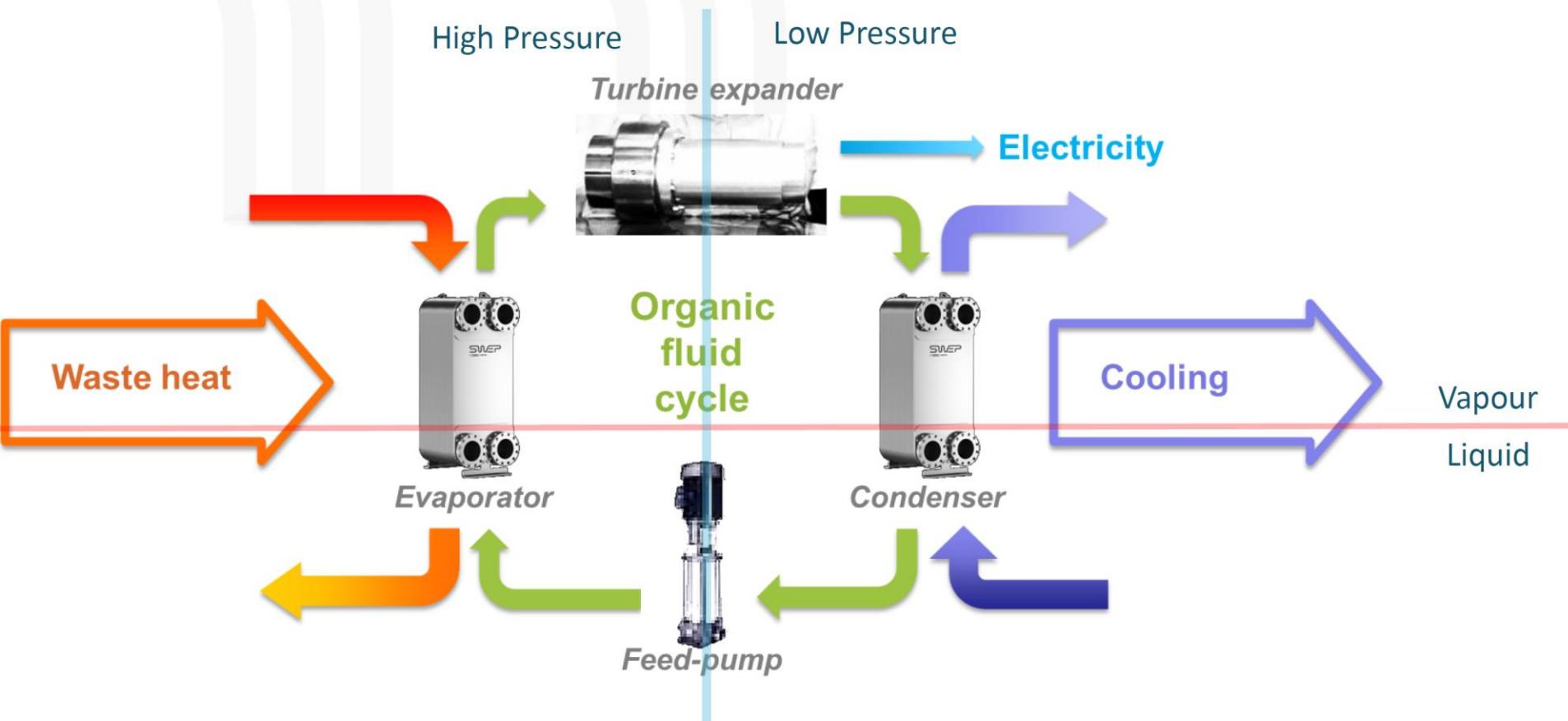
Figure 4.10. (a) Simple organic Rankine cycle. (b) T-s diagram.

Table 4.1 Basic properties of working fluids. Source: [ethermo.us/Refrigerant](http://ethermo.us/Refrigerant).

Working fluid	Critical temperature (°C)	Critical pressure (kPa)	Critical density (kg m <sup>-3</sup> )
R134a	101	4059	511.90
R227ea	102.8	2925	594.25
R245fa	154	3651	516.08
R123	183.68	3661.8	550
R600	152	3797	228
Toluene	319	4126	291
Iso-butane	134.7	3650	
Iso-pentane	187.2	3629	

R245a = Pentafluoro propane

# How does an ORC work?



# How to select the working fluids?

In general, four main categories of criteria can be distinguished in all classifications:

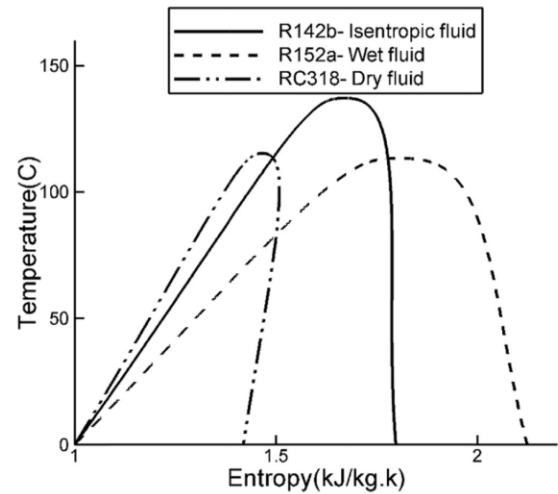
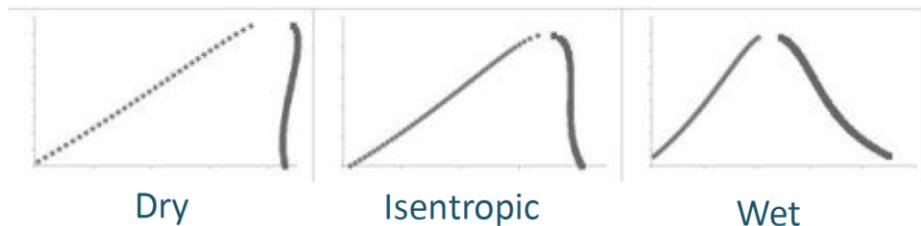
- performance criteria,
- technical and economic criteria;
- Operating limit criteria
- environmental and safety criteria

# How to select the working fluids?

There are three types of fluids: dry fluids, isentropic fluids and wet fluids.

This distinction is made by the value of the slope  $\delta$  on the expansion in the entropic diagram, defined by  $\delta=dS/dT$  on the saturation vapor curve

- If  $\delta>0$ , dry fluid;
- If  $\delta=0$ , Isentropic fluid;
- If  $\delta<0$ , Wet fluid;



# How to select the working fluids?

Technical and economic criteria:

- Price per Kg
- Implementation

Operating limit criteria:

- Maximum temperature and pressure.
- Partial vaccuum acceptable or not

Environmental and safety criteria

- GWP: Global warming potential
- ODP: Ozone Depletion Potential
- Fluid toxicity and flammability

# How to select the working fluids?

## Exemple

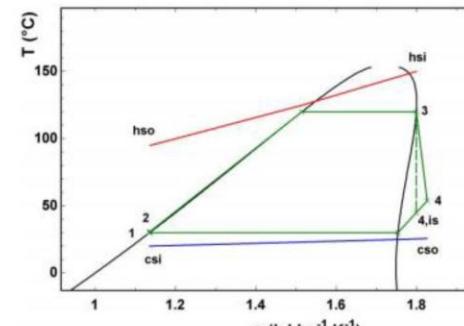
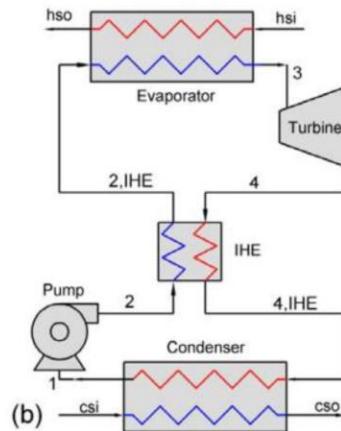
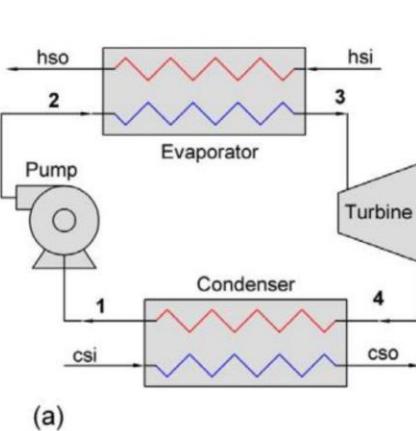
Case:

- Hot loop temperature Inlet: 200 °C
- Cold loop temperature Outlet: 70°C
- Application: production Heat and electricity supply to household

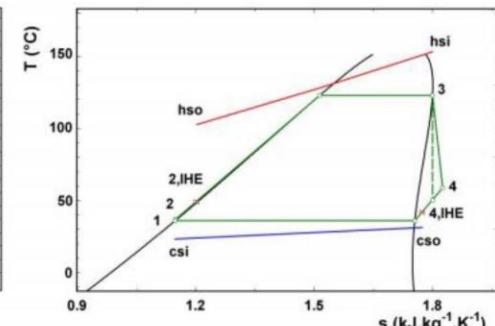
Physical parameter	Novec 649	Cyclopentane	R365mfc
Boiling temperature (° C)	165	188	180
Critical temperature (° C)	168	239	187
Boiling Pressure (Bar)	17,5	22,11	29,08
Superheating (° C)	5	5	5
Condensing temperature (° C)	73	73	73
Condensing pressure (Bar)	2,18	2,1	2,90
Subcooling (° C)	1	3	3
Pressure Ratio	8,02	10,52	10
Mass flow (Kg/s)	0,22	0,05	0,1
Power production (kW)	2,39	2,99	2,73
$\eta_{cycle}$ (%)	10,85	13,62	12,4
Global Warming Potential	1	11	1110
Ozone Depletion Potential	0	0	0
Atmospheric Lifetime (year)	0.014	0.008	8.6
Flammability/Toxicity	No/Low	High/High	Yes in vapor phase

# What is the regenerative cycle

In the case of a "dry fluid", the cycle can be improved using a regenerator: since the fluid has not reached the two-phase state at the end of the expansion, its temperature at this point is higher than the condensing temperature. This higher temperature fluid can be used to preheat the liquid before it enters the evaporator.

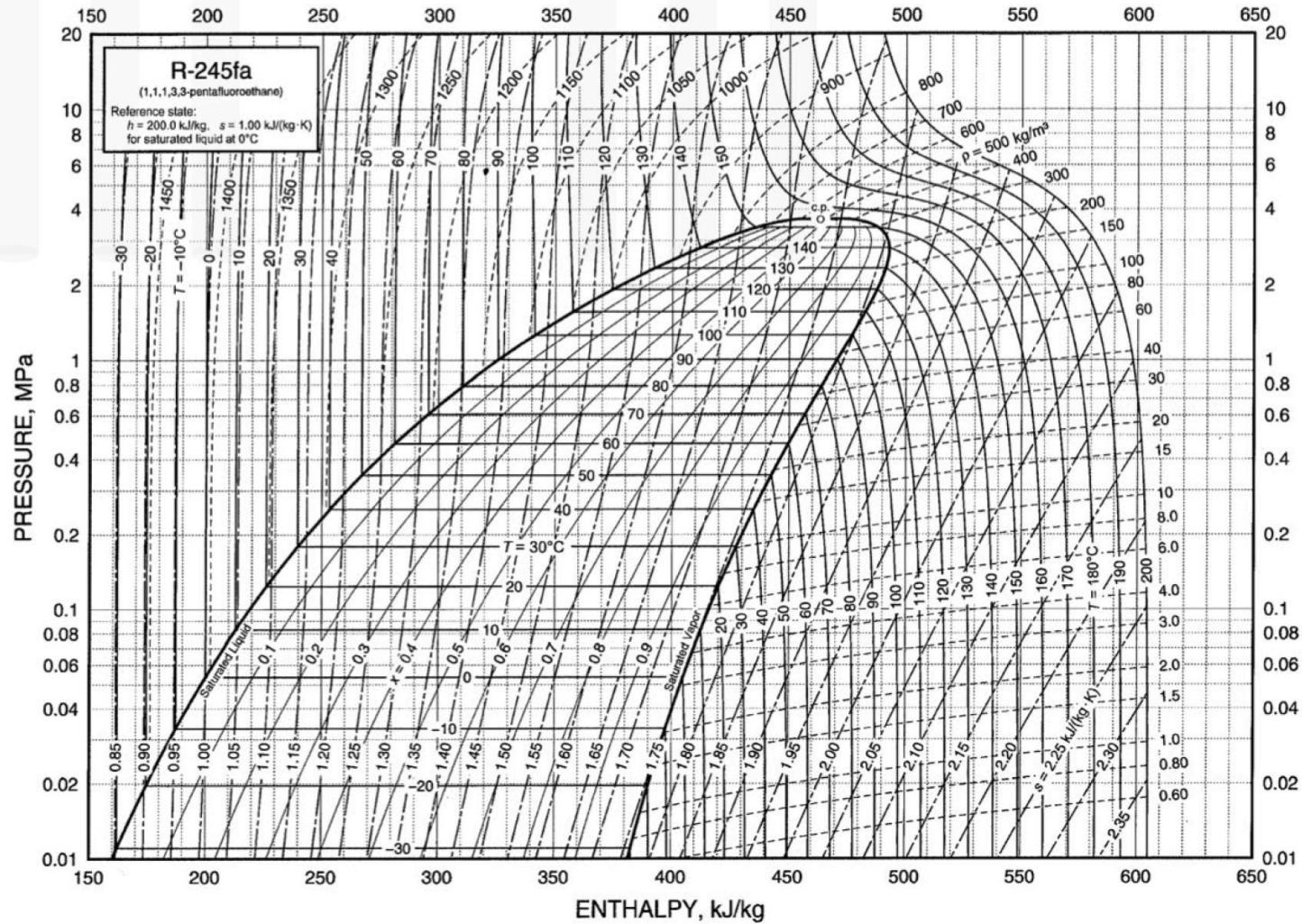


(a)

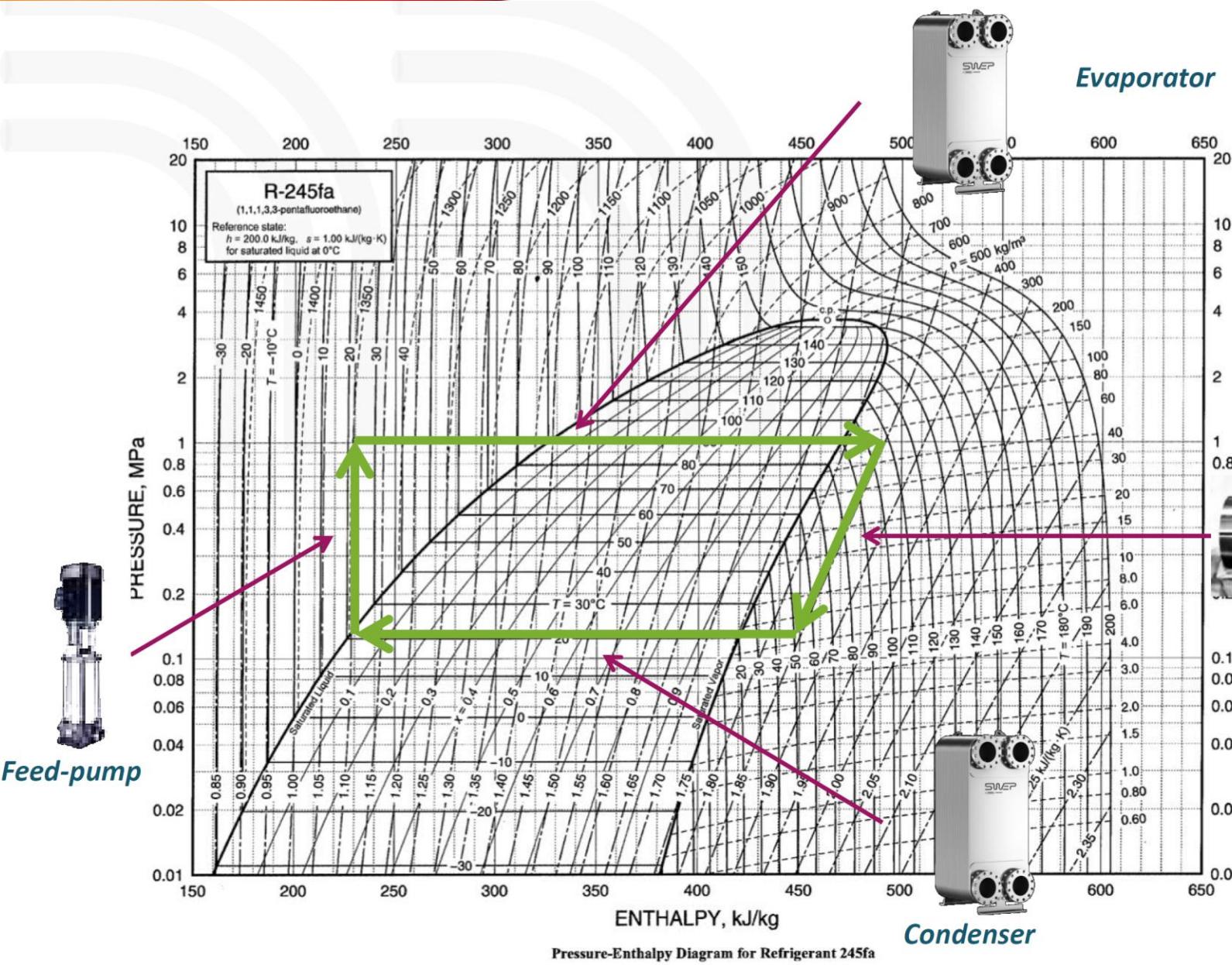


(b)

# Mollier Diagram

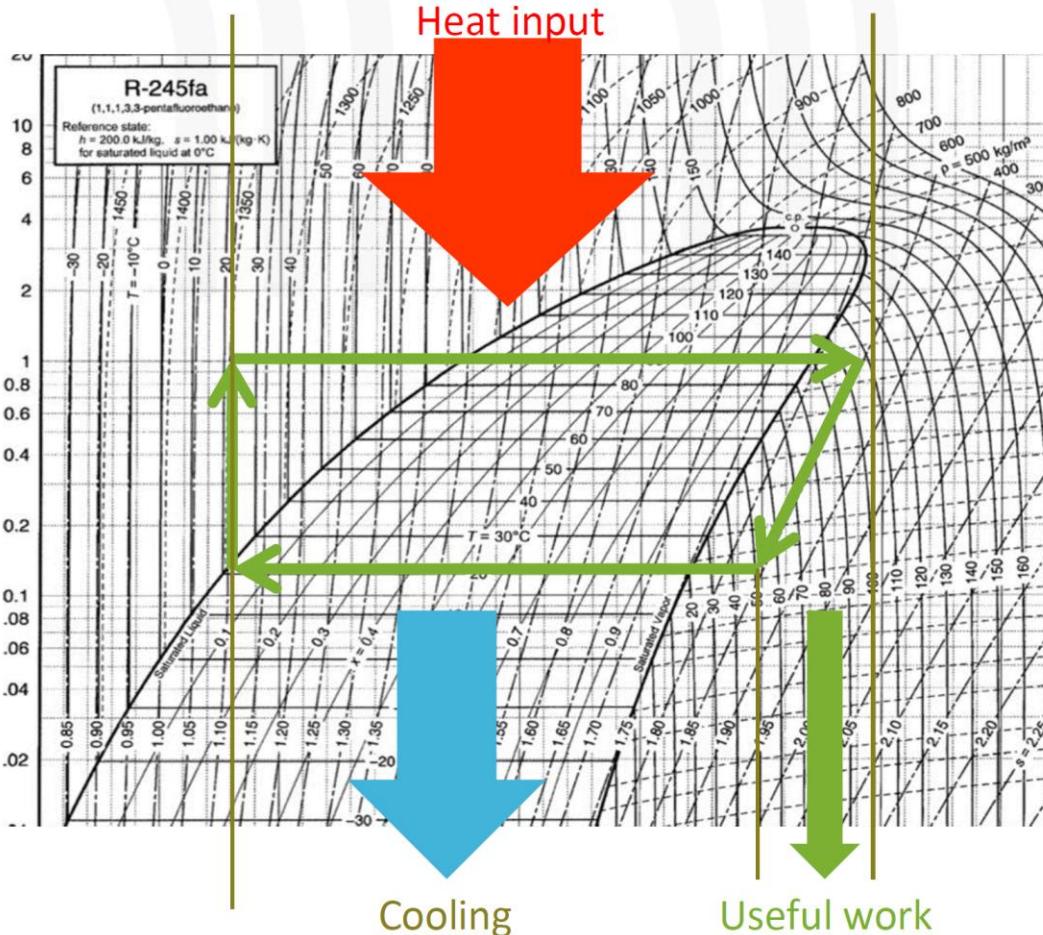


### Pressure-Enthalpy Diagram for Refrigerant 245fa



# How to design an ORC?

## Isentropic expansion

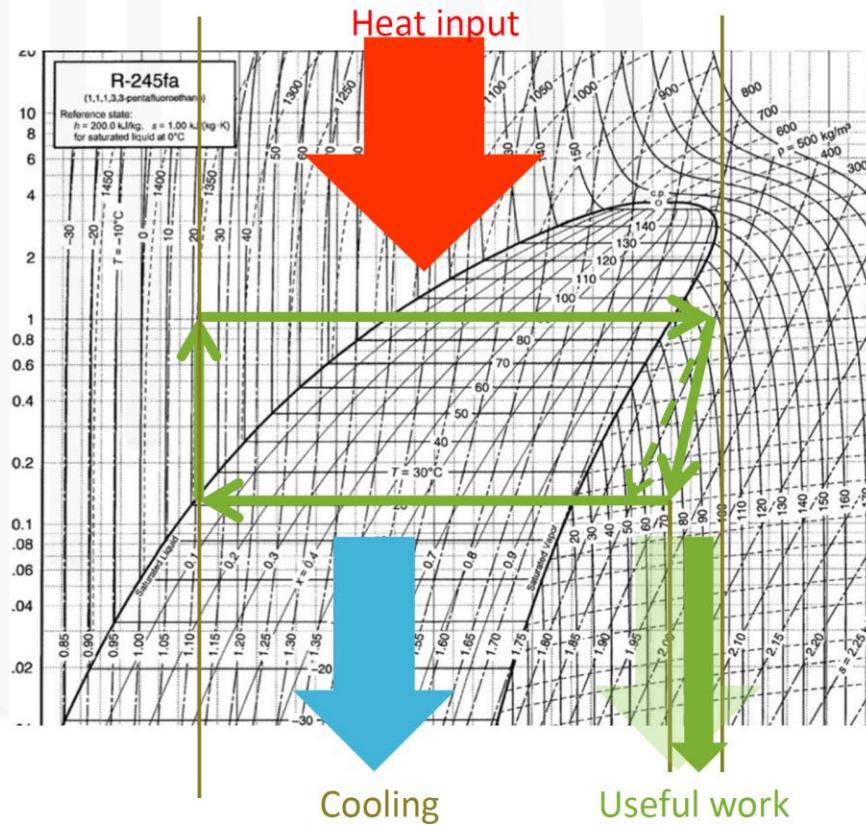


Theoretical cycle efficiency:

$$\frac{\text{Isentropic expansion enthalpy}}{\text{Heating enthalpy}}$$

# How to design an ORC?

## Real expansion



$$\text{Real Cycle efficiency} = \frac{\text{Real expansion enthalpy}}{\text{Heating enthalpy}}$$

$$\begin{aligned}\text{Real Cycle efficiency} \\ = \text{Theoretical cycle efficiency} \times \text{Turbine efficiency}\end{aligned}$$

$$\text{Turbine efficiency} = \frac{\text{Real expansion enthalpy}}{\text{Isentropic expansion enthalpy}}$$

# How to design an ORC?

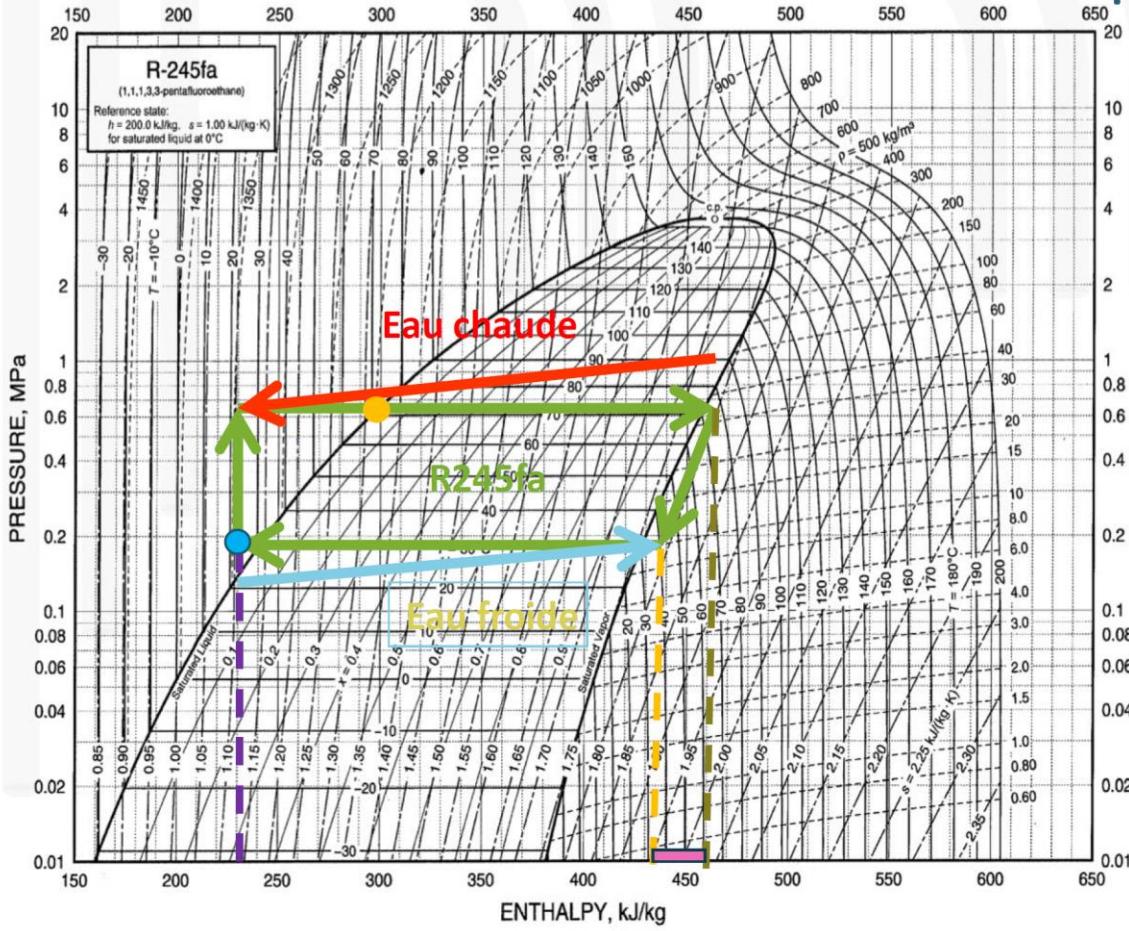
## Exemple

Step:

1. Define boiling and condensation temperature and pressure.
2. Calculate the heating enthalpy
3. Calculate the expansion enthalpy
4. Define the theoretical cycle efficiency
5. Calculate the real cycle efficiency
6. Calculate the electrical power output

# How to design an ORC?

## Exemple



- Boiling Point: 70°C  
6 bar
- Condensing point: 33°C  
2 bar
- Enthalpy inlet turbine: 456 kJ/kg enthalpie
- Enthalpy outlet the condenser: 243 kJ/kg enthalpie
- Theoretical expansion enthalpie: 21 kJ/kg

# **How to design an ORC?**

## **Example: Calculation of efficiency and power output of 100 kW unit.**

- Heat enthalpy =  $456 - 243 = 213 \text{ KJ/kg}$
- Turbine expansion work =  $21 \text{ KJ/Kg}$
- Turbine efficiency (assumed) =  $55\%$
- Theoretical cycle efficiency =  $21/213 = 10\%$
- Real cycle efficiency =  $10\% \times 55\% = 5.5\%$
- Electrical output of the ORC =  $5.5 \text{ kW}$

The Organic Rankine Cycle (ORC) is a thermodynamic cycle similar to the conventional Rankine cycle but uses an organic working fluid such as R245fa. This report calculates the efficiency of an ORC using R245fa as the working fluid, with specified conditions: evaporator pressure of 1MPa and condenser pressure of 0.15MPa, assuming an ideal cycle initially. Modifications will then be made to account for turbine efficiency and pinch points.

## Ideal Case

In the ideal case, we assume:

- Isentropic compression in the pump.
- Isentropic expansion in the turbine.
- No pinch points in the evaporator and condenser.

## Given Data

- Working fluid: R245fa.
- Evaporator pressure: 1MPa.
- Condenser pressure: 0.15MPa.
- Saturated liquid at condenser exit.

The thermodynamic states at key points in the cycle are as follows:

- **State 1:** Saturated liquid at 0.15MPa, with enthalpy  $h_1 = 235.23 \text{ kJ/kg}$  and specific volume  $v_1 = 0.001244 \text{ m}^3/\text{kg}$ .
- **State 2:** After pumping to 1MPa, the enthalpy  $h_2$  is calculated using:

$$W_{\text{pump}} = v_1(P_2 - P_1)$$

$$W_{\text{pump}} = 0.001244 \times (1,000 - 150) = 1.054 \text{ kJ/kg}$$

$$h_2 = h_1 + W_{\text{pump}} = 235.23 + 1.054 = 236.284 \text{ kJ/kg}$$

- **State 3:** Saturated vapor at 1MPa, with enthalpy  $h_3 = 439.89 \text{ kJ/kg}$ .
- **State 4:** Isentropic expansion to 0.15MPa, with isentropic enthalpy  $h_4 = 297.26 \text{ kJ/kg}$ .

### Efficiency Calculation (Ideal Case)

The heat added in the evaporator is:

$$Q_{\text{in}} = h_3 - h_2 = 439.89 - 236.284 = 203.606 \text{ kJ/kg}$$

The work done by the turbine is:

$$W_{\text{turbine}} = h_3 - h_4 = 439.89 - 297.26 = 142.63 \text{ kJ/kg}$$

The net work is:

$$W_{\text{net}} = W_{\text{turbine}} - W_{\text{pump}} = 142.63 - 1.054 = 141.576 \text{ kJ/kg}$$

The thermal efficiency of the ideal cycle is:

$$\eta_{\text{ideal}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{141.576}{203.606} = 0.6954 \text{ or } 69.54\%$$

## Modified Case: Turbine Efficiency and Pinch Points

Now, we modify the cycle to include:

- **Turbine Efficiency:** 55%.
- **Evaporator Pinch Point:** 2 K below the evaporator temperature.
- **Condenser Pinch Point:** 3 K above the condenser temperature.

## Revised State Points

- **State 1:** Saturated liquid at 0.15MPa, with enthalpy  $h_1 = 235.23\text{kJ/kg}$ .
- **State 2:** After pumping to 1MPa, with enthalpy  $h_2 = 236.284\text{kJ/kg}$ .
- **State 3:** Saturated vapor at 1MPa, but the evaporator pinch point lowers the temperature by 2K. The revised enthalpy  $h_3 = 438.89\text{kJ/kg}$ .
- **State 4:** The expansion is non-ideal with 55% turbine efficiency. The actual enthalpy  $h_4$  is calculated as:

$$h_4 = h_3 - \eta_{\text{turbine}} \times (h_3 - h_4^{\text{isentropic}})$$

$$h_4 = 438.89 - 0.55 \times (438.89 - 297.26) = 360.99 \text{ kJ/kg}$$

## Efficiency Calculation (Modified Case)

The work done by the turbine is now:

$$W_{\text{turbine}} = h_3 - h_4 = 438.89 - 360.99 = 77.90 \text{ kJ/kg}$$

The net work is:

$$W_{\text{net}} = W_{\text{turbine}} - W_{\text{pump}} = 77.90 - 1.054 = 76.846 \text{ kJ/kg}$$

The thermal efficiency is:

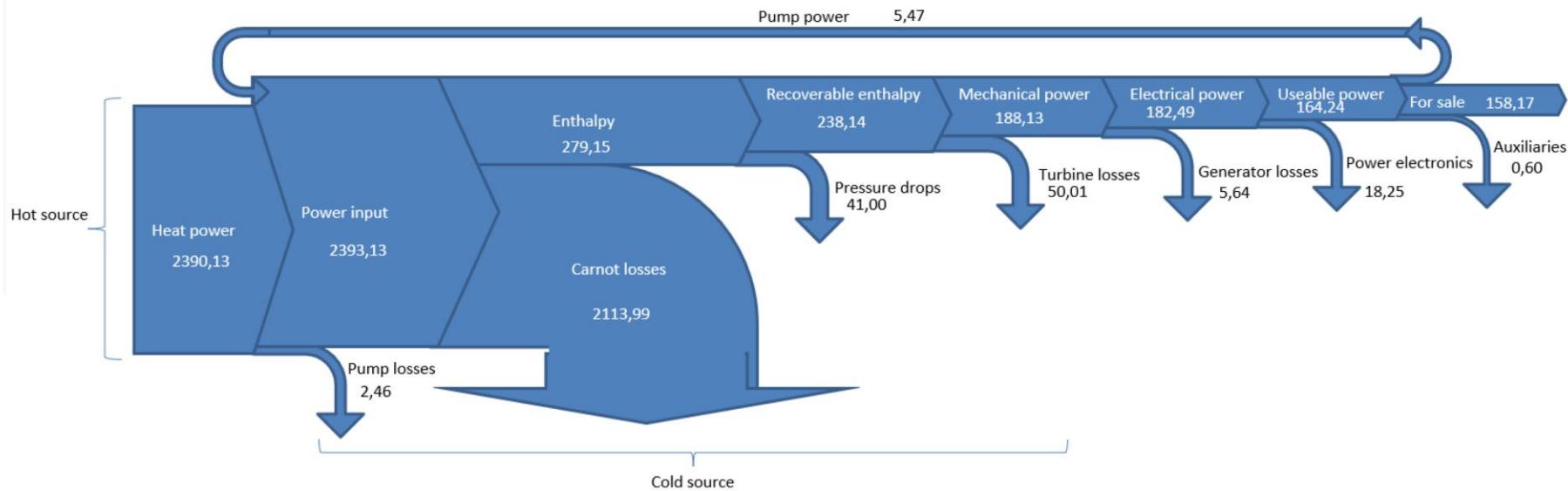
$$\eta_{\text{modified}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{76.846}{202.606} = 0.3793 \text{ or } 37.93\%$$

## Conclusion

In the ideal case, the ORC efficiency was calculated as 69.54%. However, after incorporating a 55% turbine efficiency and considering evaporator and condenser pinch points, the efficiency decreases to 37.93%. These factors are critical in determining the real-world performance of an ORC system.

# How to design an ORC?

## Sankey diagram



# Heat recovery in geothermal context

## Constraints

- A geothermal power plant is a risky and capital-intensive investment
- Geothermal fluids are characterized as a solution of pure water plus dissolved minerals (i.e., Fe, Zn, Cu) and dissolved gases (i.e., H<sub>2</sub>CO<sub>3</sub>, H<sub>2</sub>S), which varies greatly field by field and well by well,
- Due to some of the chemical components of the geothermal fluid, the exploitation of geothermal resources may cause scaling and corrosion in geothermal plants
- The geothermal environment is critical for the design of electrical components due to the presence of sulfidic components: corrosion of copper electrical equipment in a geothermal environment from hydrogen sulfide contamination.



Rack of material sample on the Chaunoy 40 wells

# Heat recovery in geothermal context

- Operate an existing facility. (oil facility).
- For each geology, it is necessary to identify the most resistant and cheapest material.
- Being able to exploit small geothermal wells (Low flow and low temperature) but with a good return on investment



Chaunoy (France)



Grasteinn (Iceland)



Soultz Sous Forêts (France)

# Solar Organic Rankine Cycle

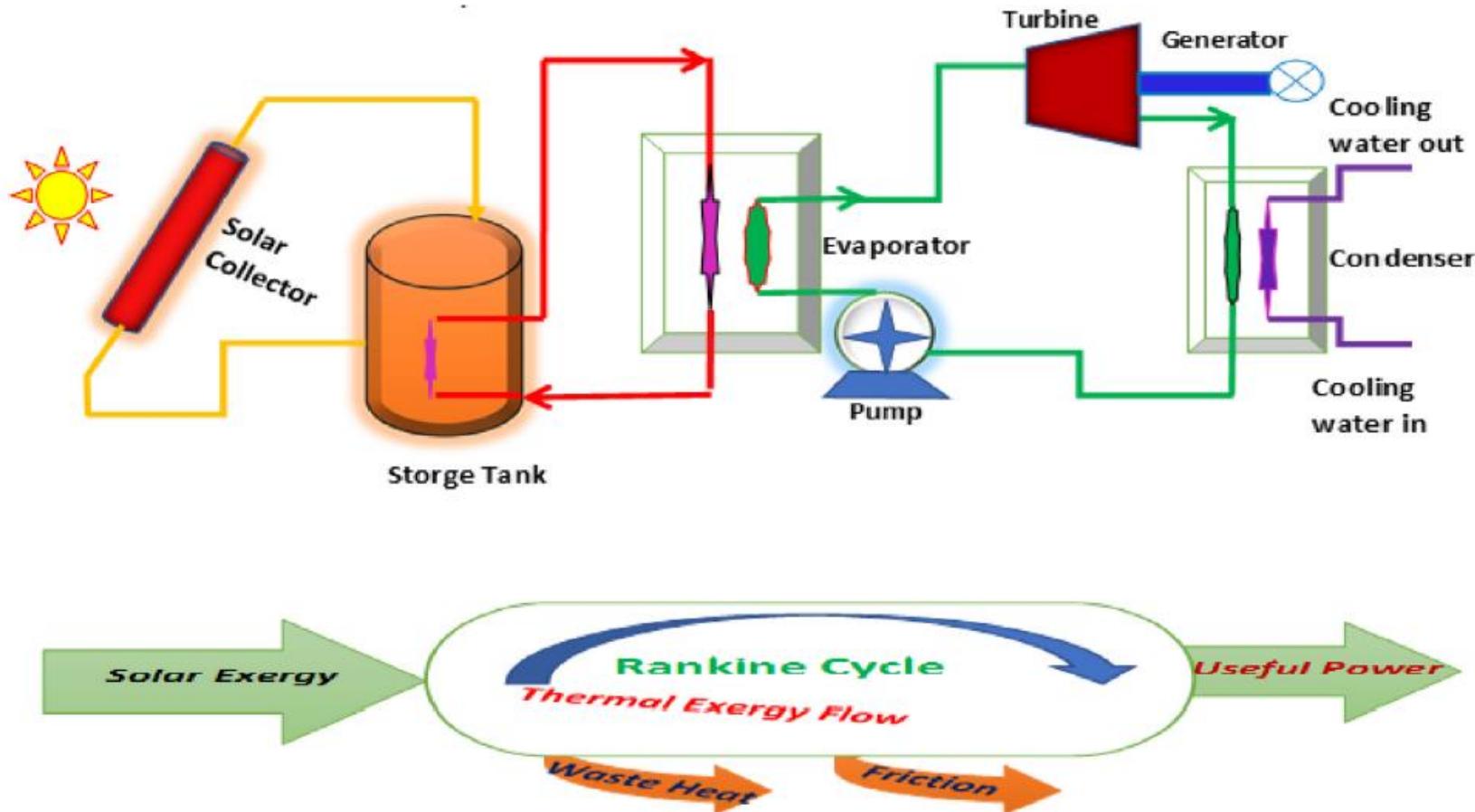


Figure 4.12. Thermal energy storage for solar-powered organic Rankine cycle and exergy flow diagram.

# Example on Solar Organic Rankine Cycle

**EXAMPLE 8** Consider an engine operating on a solar Rankine cycle and Refrigerant 134a is its working fluid. The saturated vapor enters the turbine at 60 °C and exits as a saturated liquid-vapor mixture and enters the condenser. The condenser pressure is 6 bar and the working fluid exits the condenser as a saturated liquid. The R-134a fluid is then pressurized in the pump before entering the constant solar heater. The energy input to the collectors from solar radiation is 0.4 kW m<sup>-2</sup>. Find the minimum possible solar collector surface area, in m<sup>2</sup>, per kW of power produced by the plant.

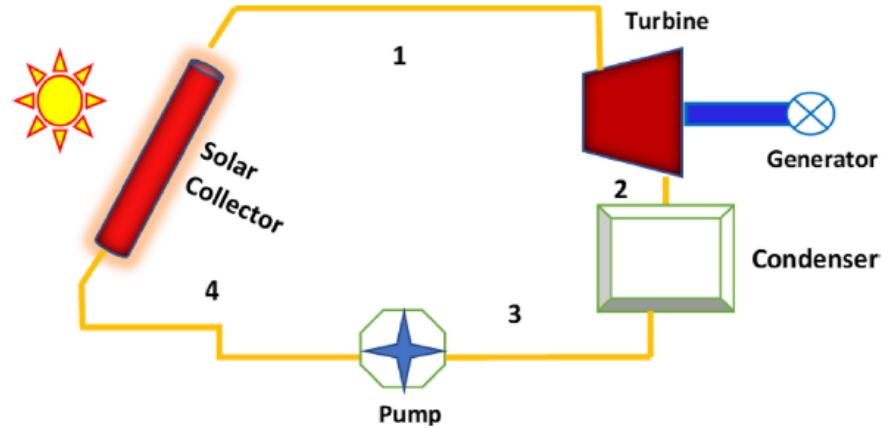


Figure 4.13. The schematic diagram of the solar power plant.

# Analysis of Solar Organic Rankine Cycle

**TABLE A-10**

Pressure Conversions:  
1 bar = 0.1 MPa  
=  $10^5$  kPa

Properties of Saturated Refrigerant 134a (Liquid–Vapor): Temperature Table

Temp. °C	Press. bar	Specific Volume m <sup>3</sup> /kg		Internal Energy kJ/kg		Enthalpy kJ/kg		Entropy kJ/kg · K		Temp. °C
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor $v_g$	Sat. Liquid $u_f$	Sat. Vapor $u_g$	Sat. Liquid $h_f$	Evap. $h_{fg}$	Sat. Vapor $h_g$	Sat. Liquid $s_f$	
48	12.526	0.8989	0.0159	117.22	251.79	118.35	153.33	271.68	0.4243	0.9017
52	13.851	0.9142	0.0142	123.31	253.55	124.58	148.66	273.24	0.4432	0.9004
56	15.278	0.9308	0.0127	129.51	255.23	130.93	143.75	274.68	0.4622	0.8990
60	16.813	0.9488	0.0114	135.82	256.81	137.42	138.57	275.99	0.4814	0.8973

# Thermodynamic properties of R-134a at 60°C (state 1) and at 6 bar (state 3)

# Given values T1 = 60 # Temperature at state 1 (°C)

h1 = 275.99 # Enthalpy at state 1 (kJ/kg)

s1 = 0.8973 # Entropy at state 1 (kJ/kg · K)

P1 = 16.813 # Pressure at state 1 (bar)

P4 = P1\*100 # Pressure at state 4 (kPa)

TABLE A-11

## Properties of Saturated Refrigerant 134a (Liquid-Vapor): Pressure Table

Pressure Conversions:  
1 bar = 0.1 MPa  
=  $10^5$  kPa

Press. bar	Temp. °C	Specific Volume m <sup>3</sup> /kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		Press. bar
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor $v_g$	Sat. Liquid $u_f$	Sat. Vapor $u_g$	Sat. Liquid $h_f$	Evap. $h_{fg}$	Sat. Vapor $h_g$	Sat. Liquid $s_f$	Sat. Vapor $s_g$	
3.2	2.48	0.7770	0.0632	53.06	228.43	53.31	195.35	248.66	0.2089	0.9177	3.2
3.6	5.84	0.7839	0.0564	57.54	230.28	57.82	192.76	250.58	0.2251	0.9160	3.6
4.0	8.93	0.7904	0.0509	61.69	231.97	62.00	190.32	252.32	0.2399	0.9145	4.0
5.0	15.74	0.8056	0.0409	70.93	235.64	71.33	184.74	256.07	0.2723	0.9117	5.0
6.0	21.58	0.8196	0.0341	78.99	238.74	79.48	179.71	259.19	0.2999	0.9097	6.0

P3 = 6 # Pressure at state 3 (bar)

P3 = 600.0 # Pressure at state 3 (kPa)

v3 = 0.0008196 # Specific volume at state 3 (m<sup>3</sup>/kg)

h3 = 79.48 # Enthalpy at state 3 (kJ/kg)

sf = 0.2999 # Saturated liquid entropy at 6 bar (kJ/kg · K)

sg = 0.9097 # Saturated vapor entropy at 6 bar (kJ/kg · K)

sfg = sg-sf # Entropy difference between vapor and liquid at 6 bar (kJ/kg · K) = 179.71

hf = h3 # Saturated liquid enthalpy at 6 bar (kJ/kg)

```
net_work = 1.0 # Net power output (kW)
heat_transfer_coefficient = 0.4 # Heat transfer
coefficient (kW/m2)
```

```
# Calculate quality (x2)
x2 = (s1 - sf) / sfg = 0.980
```

```
# Calculate enthalpy (h2)
h2 = hf + x2 * hfg = 255.54 kJ/kg
```

```
# Pump work
wp = -v3 * (P4 - P3) = -0.88623 kJ/kg # Work is in kJ/kg
```

```
# Enthalpy at state 4
h4 = h3 - wp = 80.37 kJ/kg
```

# Turbine work

$$wt = h_1 - h_2 = 20.45 \text{ kJ/kg}$$

# Heat input

$$q_{in} = h_1 - h_4 = 195.62 \text{ kJ/kg}$$

# Heat rejected in the condenser

$$q_{out} = h_2 - h_3 = 176.06 \text{ kJ/kg}$$

# Mass flow rate calculation

$$\text{mass\_flow\_rate} = \text{net\_work} / (wt - wp) = 0.04686 \text{ kg/s}$$

# Total heat addition

$$q_{in\_total} = \text{mass\_flow\_rate} * q_{in} = 9.167 \text{ kW}$$

# Required surface area

$$A = q_{in\_total} / \text{heat\_transfer\_coefficient} = 22.917 \text{ m}^2$$

# The Biomass Power Plant

- ❑ Biomass is green and does not increase the CO<sub>2</sub> level because the atmospheric CO<sub>2</sub> gas is captured in photosynthesis.
- ❑ Biomass is available worldwide and can be used to produce electricity on a wide scale (from 1MW to 15 MW).
- ❑ Commonly for largescale biomass power generation, the most effective way to generate power is co-fired with coals, achieving 45% efficiency.
- ❑ The low working pressures in the organic Rankine cycle power plants have overcome the problem of high costs for requirements such as steam boilers.
- ❑ A further advantage is the engine's long working life due to the working fluid's characteristics which are different from steam as it does not corrode the turbine blades.
- ❑ The organic Rankine cycle overcomes the issue of the small quantity of input fuel available in many regions because a small-sized power plant utilizes an efficient cycle.

# Example

**EXAMPLE 11** Biomass is burnt to generate water steam in an electric power plant. Assume the system runs at an ideal Rankine cycle as shown in figure 4.17. The inlet turbine condition is 12 MPa and the saturated vapor mass flow rate is  $120 \text{ kg s}^{-1}$ . The condenser operates at 4.0 kPa. Determine:

- The power output and the heat transfer in the boiler in kW,
- The thermal efficiency,
- The flow rate of the condenser cooling water in  $\text{kg s}^{-1}$  if the cooling water is subject to a temperature increase of  $20^\circ\text{C}$  with a constant pressure through the condenser.

**Properties**  $T_2 = 300\text{K}$   $P_1 = 12\text{MPa}$   $P_2 = 4.0\text{kPa}$   $m_{t,\text{in}} = \text{s}^{-1}$   
Specific heat of water ( $C_p$ ) at  $300\text{K}$  is  $4.179\text{ kJ kg}^{-1}$ .

**Analysis** The model used in this example is a steam power plant powered by biomass fuel. The diagram is shown in figure 4.15.

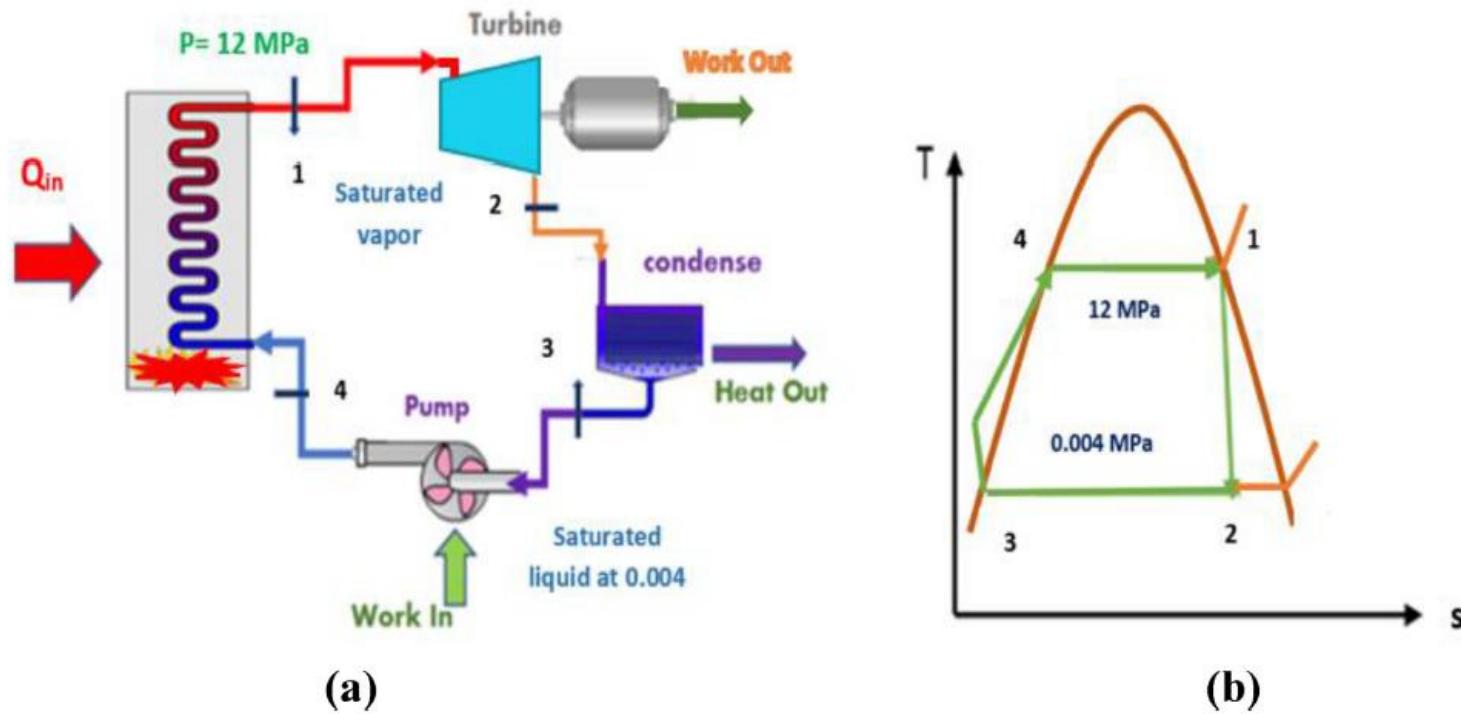


Figure 4.17. (a) Schematic representation of steam cycle. (b)  $T$ - $s$  diagram.

## Turbine Inlet

TABLE A-3

(Continued)

Pressure Conversions:  
1 bar = 0.1 MPa  
=  $10^5$  kPa

Press. bar	Temp. °C	Specific Volume m <sup>3</sup> /kg		Internal Energy kJ/kg		Enthalpy kJ/kg		Entropy kJ/kg · K		Press. bar
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor $v_g$	Sat. Liquid $u_f$	Sat. Vapor $u_g$	Sat. Liquid $h_f$	Evap. $h_{fg}$	Sat. Vapor $h_g$	Sat. Liquid $s_f$	
120.	324.8	1.5267	0.01426	1473.0	2513.7	1491.3	1193.6	2684.9	3.4962	5.4924
130.	330.9	1.5671	0.01278	1511.1	2496.1	1531.5	1130.7	2662.2	3.5606	5.4323
140.	336.8	1.6107	0.01149	1548.6	2476.8	1571.1	1066.5	2637.6	3.6232	5.3717
150.	342.2	1.6581	0.01034	1585.6	2455.5	1610.5	1000.0	2610.5	3.6848	5.3098
160.	347.4	1.7107	0.009306	1622.7	2431.7	1650.1	930.6	2580.6	3.7461	5.2455

## Condenser Inlet

0.04	28.96	1.0040	34.800	121.45	2415.2	121.46	2432.9	2554.4	0.4226	8.4746	0.04
0.06	36.16	1.0064	23.739	151.53	2425.0	151.53	2415.9	2567.4	0.5210	8.3304	0.06
0.08	41.51	1.0084	18.103	173.87	2432.2	173.88	2403.1	2577.0	0.5926	8.2287	0.08
0.10	45.81	1.0102	14.674	191.82	2437.9	191.83	2392.8	2584.7	0.6493	8.1502	0.10
0.20	60.06	1.0172	7.649	251.38	2456.7	251.40	2358.3	2609.7	0.8320	7.9085	0.20

# Solution

# Known values

```
mass_flow_rate = 120 # Steam flow rate (kg/s)
P_turbine_in = 12 * 1000 # Inlet turbine pressure (kPa)
P_condenser = 4.0 # Condenser pressure (kPa)
delta_T_cooling = 20 # Temperature increase in cooling water (°C)
c_water = 4.18 # Specific heat of water (kJ/kg ·°C)
```

# Given values from the tables

```
s2_wet = 5.4974 # Specific entropy at state 2 (kJ/kg ·K)
sf_2 = 0.4226 # Specific entropy of saturated liquid at 0.004 MPa
                (kJ/kg ·K)
sg_2 = 8.4746 # Specific entropy of saturated vapor at 0.004 MPa
                (kJ/kg ·K)

hf_2 = 121.46 # Specific enthalpy of saturated liquid at 0.004 MPa (kJ/kg)
hg_2 = 2432.9 # Specific enthalpy of saturated vapor at 0.004 MPa
                (kJ/kg)

h1_g = 2684.9 # Specific enthalpy at turbine inlet (kJ/kg)
```

# (a) Calculate the quality of steam (x2)

$$x_2 = (s_2_{\text{wet}} - s_{f2}) / (s_{g2} - s_{f2}) = 0.6303 \text{ or } 63.03 \%$$

# (b) Calculate specific enthalpy at state 2 (h2\_wet)

$$h_{2\text{wet}} = h_{f2} + x_2 * (h_{g2} - h_{f2}) = 1578.25 \text{ kJ/kg}$$

# (c) Work done per kg of steam entering the turbine

$$\text{work\_turbine} = h_{1g} - h_{2\text{wet}} = 1106.65 \text{ kJ/kg}$$

# (d) Turbine power output

$$W_t = \text{mass\_flow\_rate} * \text{work\_turbine} = 132797.66 \text{ kW}$$

# (e) Heat transfer rate in the boiler

$$Q_{in} = \text{mass\_flow\_rate} * (h_{1g} - h_{f2}) = 307612.80 \text{ kW}$$

# (f) Thermal efficiency

$$\text{thermal\_efficiency} = W_t / Q_{in} = 43.17 \%$$

# (g) Heat rejected in the condenser

$$Q_{out} = \text{mass\_flow\_rate} * (h_2_{wet} - h_f_2) = 174815.14 \text{ kW}$$

# (h) Mass flow rate

$$\text{mass\_flow\_rate\_cooling} = Q_{out} / (c_{water} * \Delta T_{cooling}) = 2091.09 \text{ kg/s}$$

# Ocean Thermal Energy Conversion (OTEC)



Figure 4.18. Ocean thermal energy source.

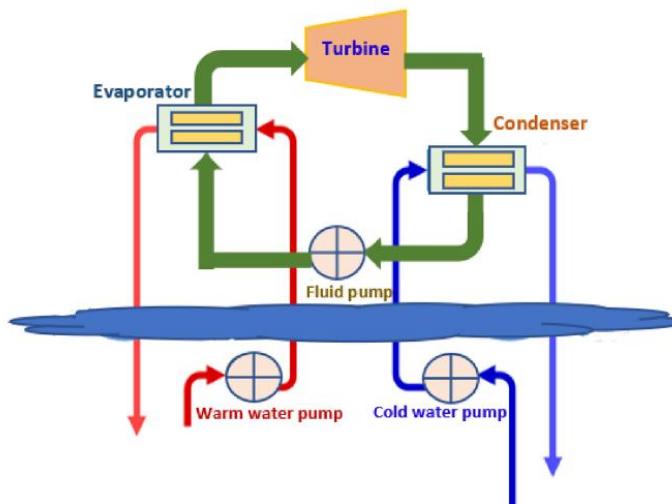


Figure 4.19. Closed cycle ocean thermal energy conversion.

# Ocean Thermal Energy Conversion (OTEC)

- ❑ The world's ocean surface absorbs about 80% of the Sun's solar energy and is replenished daily under any weather condition.
- ❑ Processes in a closed cycle ocean thermal energy conversion start with water flowing through a large pipe into the heat exchanger.
- ❑ A liquid such as ammonia is heated with a low boiling point, thus creating steam, it turns a turbine generator to produce electricity.
- ❑ A second pipe draws out cool water from the deep ocean. This cold water condenses the steam and back to liquid.
- ❑ The warm surface ocean water temperature varies seasonally at 24–30 °C, while the cold deep ocean water remains 5-9 °C.
- ❑ The temperature difference in the ocean water needs approximately 20 °C to keep the ocean thermal energy conversion system at constant power.

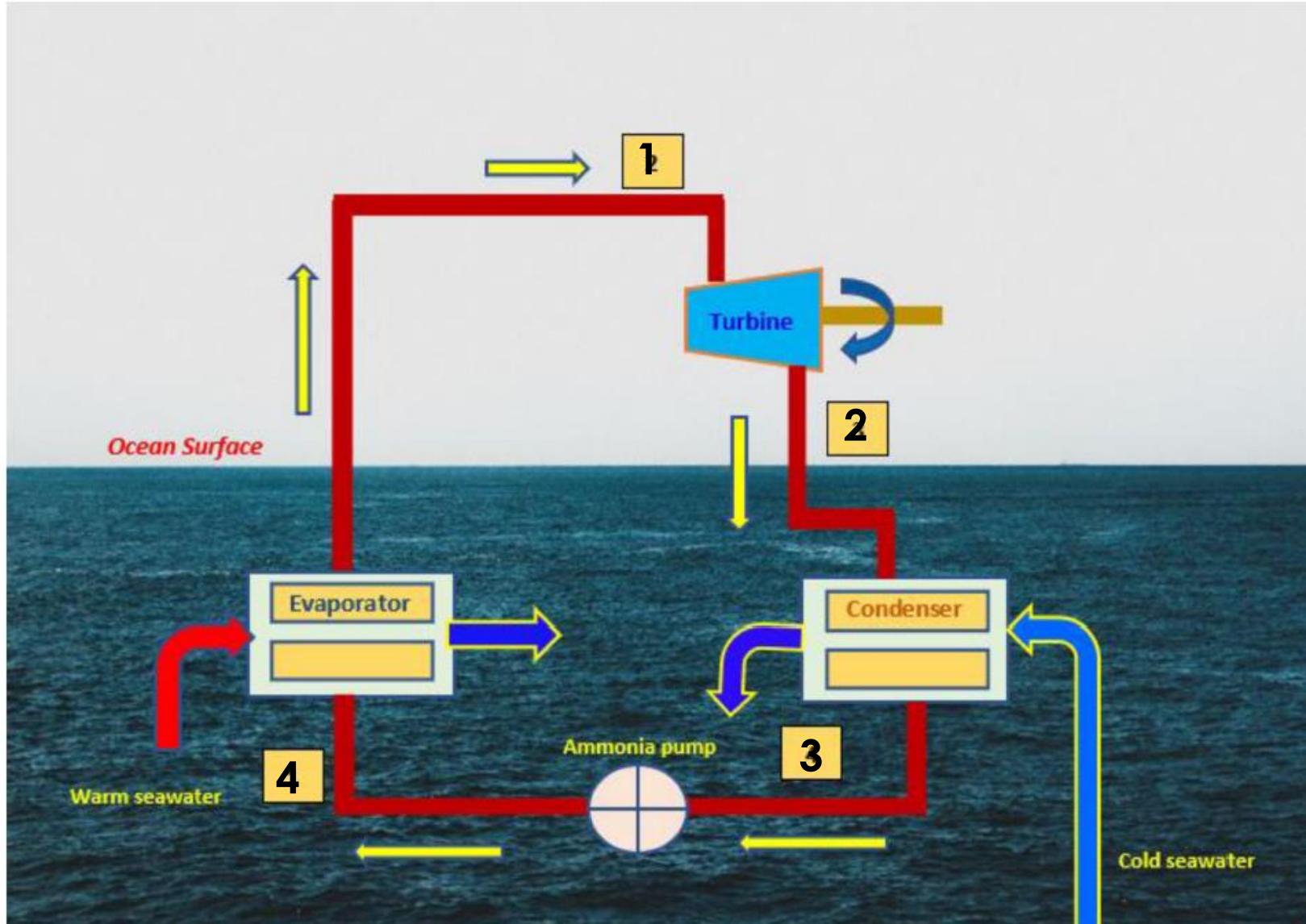


Figure 4.22. Ocean thermal energy conversion (OTEC).

## OTEC Example

**EXAMPLE 12** Ocean thermal energy conversion (OTEC), shown below, is a renewable energy conversion technology that makes use of the temperature difference between surface seawater and deep seawater for power generation. OTEC is based on the Rankine cycle that uses a refrigerant as the working fluid to produce work with heat from low-temperature thermal energy. Let us consider using R-134a for OTEC. The pump increases the pressure of R-134a to 500 kPa, the turbine inlet temperature is at 25 °C, and the condenser pressure is 360 kPa. Calculate the working fluid mass flow rate needed if the cycle output is 1 MW of power.

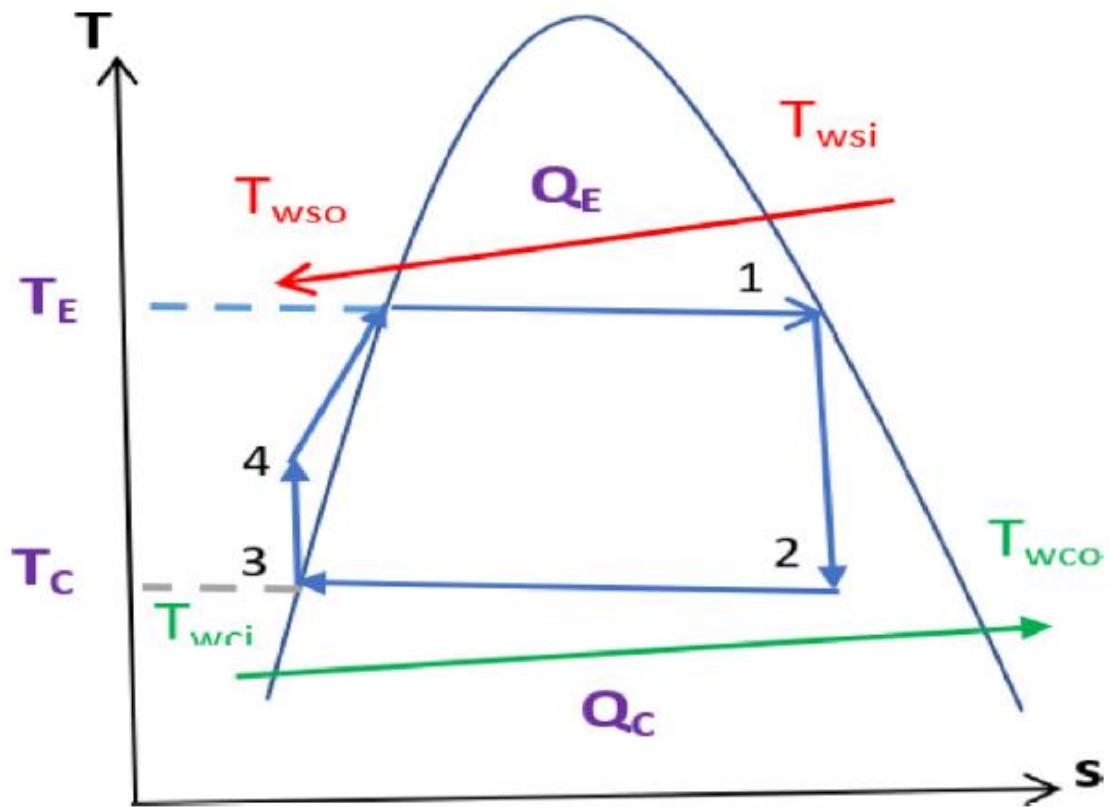


Figure 4.20. Ocean thermal energy conversion cycle T-s diagram.

# Analysis

## Liquid Phase Data

### Data on Saturation Curve

Temperature (C)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)
5.8412	0.36000	1275.2	0.00078418	207.61	207.90	1.0284
15.735	0.50000	1240.8	0.00080595	221.10	221.50	1.0759

## Vapor Phase Data

### Data on Saturation Curve

Temperature (C)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)
5.8412	0.36000	17.623	0.056744	381.54	401.97	1.7240
15.735	0.50000	24.317	0.041123	386.91	407.47	1.7197

## Solution:

$T_1 = 25^\circ C = 298 K$ ,  $P_1 = P_4 = 500 \text{ kPa}$ ,  $P_2 = P_3 = 360 \text{ kPa}$

$T_3 = 5^\circ C$ , power output = 1 MW.

$h_3 = h_f$  at 360 kPa = 207.9 kJ/kg,

$v_3 = v_f$  at 360 kPa = 0.000784 m<sup>3</sup>/kg

$$w_p = v_3 (P_4 - P_3)$$

$$w_p = 0.000784 (500 - 360) = 0.1097 \text{ kJ/kg}$$

### Isobaric Data for P = 0.50000 MPa

Temperature (C)	Pressure (MPa)	Density (kg/m <sup>3</sup> )	Volume (m <sup>3</sup> /kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)
25.000	0.50000	23.125	0.043243	394.78	416.40	1.7501

### Isobaric Data for P = 0.36000 MPa

Temperature (C)	Pressure (MPa)	Density (kg/m <sup>3</sup> )	Volume (m <sup>3</sup> /kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)
12.000	0.36000	17.067	0.058591	386.54	407.63	1.7441
13.000	0.36000	16.982	0.058885	387.34	408.54	1.7473
14.000	0.36000	16.898	0.059178	388.15	409.46	1.7505
15.000	0.36000	16.816	0.059469	388.96	410.37	1.7537

$$h_4 = w_p + h_3 = 0.1097 + 207.9 = 208 \text{ kJ/kg}$$

$$h_1 \text{ at } 500 \text{ kPa, } 25^\circ\text{C} = 416.4 \text{ kJ/kg}$$

$$s_1 \text{ at } 500 \text{ kPa, } 25^\circ\text{C} = 1.7501 \text{ kJ/kg K}^{-1}$$

$$h_2 \text{ at } 360 \text{ kPa } \& s_2 = 1.75 \text{ kJ/kg K}^{-1} = 409.46 \text{ kJ/kg}$$

$$\text{power} = \dot{m} * W_{\text{net}} = \dot{m}[(h_1 - h_2) - (h_4 - h_3)] = 1000$$

$$\dot{m} = 1000 / (416.4 - 409.46 - 0.1097) = 1000 / 6.83$$

$$\dot{m} = 146.4 \text{ kg/s}$$