

Czech Technical University in Prague

Faculty of Mechanical Engineering

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Hygroscopic Cycle

By

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A bachelor work submitted to the

Faculty of Energy Engineering, Czech Technical University in Prague.

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I. Personal and study details

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II. Bachelor's thesis details

Bachelor's thesis title in English:

A basic comparism of the hygroscopic cycle with the casical and organic Rankine cycle

Bachelor's thesis title in Czech:

Základní porovnání hygrokopického oběhu s klasickým a organickým Rankinovým oběhem

Guidelines:

The current effort to reduce energy dependence on the fossil fuels necessarily leads to the search for different approaches to electricity generation in smaller sources. One of the promising energy cycle that could be used in decentralized energy production, for example in local biomass combustion, is the hygroscopic cycle. The work is focused on the basic comparison of the properties of the classical steam cycle and the hygroscopic cycle. For a wider overview, the organic Rankine cycle was also included in the comparison. The comparison is made for an output power of 12.5 MWe.

The work will contain:

1. Current state of knowledge about hygroscopic cycles
2. Schematic of all three cycles
3. Calculation of all cycles with the real fluid properties
4. Discussion of results

Bibliography / sources:

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III. Assignment receipt

The student acknowledges that the bachelor's thesis is an individual work. The student must produce his thesis without the assistance of others, with the exception of provided consultations. Within the bachelor's thesis, the author must state the names of consultants and include a list of references.

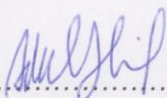
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Statement

I declare that I have worked out this thesis independently assuming that the results of the thesis can also be used at the discretion of the supervisor of the thesis as its co-author. I also agree with the potential publication of the results of the thesis or of its substantial part, provided I will be listed as the co-author.

Prague, 29.04.2022

A handwritten signature in blue ink, appearing to read 'M. J. H.', written over a horizontal dotted line.

Signature

Abstract

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1 Introduction...

1.1 Motivation

In today's economy, there is an ever-urgent demand for the change in type industrial bio footprint on our planet, where emphasis is on developing sustainable and recyclable materials and methods of manufacturing and consumption. And there is no more crucial area of development than the way we produce the energy that allows our civilization to thrive, namely, electricity, especially in areas where the production of electricity is more difficult than others, namely areas with lack of water resources and hot climates.

1.2 Problem Statement

Our problem is to choose what type of cycle should a 12 MW powerplant operate on, in a geography of hot climates and scarce water resources, where temperatures can reach a walloping 45-49 degrees of Celsius. This powerplant needs to be efficient, does not consume much water, and be environmentally friendly and non-toxic. It is important to note that the parameters used for the other cycles are recommended values and are far from optimized

1.3 Goal of thesis work

The goal of our thesis is to introduce the Hygroscopic cycle and the principle behind its workings, to develop a model that can simulate such a cycle with the aid of fluid properties tables, to compare the efficiency of our model with already established models of the Steam Rankine Cycle and the Organic Rankine Cycle, and to compare our results with the experimental data available from the research.

1.4 Structure of thesis work

We try to follow a fairly organized and methodical structure, namely, to first introduce the thermodynamic quantities that are necessary for our computations, and with it introduce all the laws that govern the interdependence of such quantities on each other, then introduce the properties of ideal fluids, and diagrams that describe the processes such fluids go through, especially when subjected to heat, then introduce the power cycles including the Hygroscopic cycle, and finally try to compute such a cycle and compare efficiencies.

2 Theoretical Background

2.1 Definitions of Thermodynamics quantities

To describe and evaluate systems, one must define the principles and quantities necessary for it. We will mainly deal with the First Law of Thermodynamics for closed and open systems, and the Second Law of Thermodynamics. Thus, introducing quantities that describe systems, such as Internal Energy of a system, Entropy, and Enthalpy of a system. Then we will discuss Fluid properties that are necessary to understand the power cycles that will be introduced later.

2.1.1 First Law of Thermodynamics

It is known that the First Law of Thermodynamics is nothing but the Law of Conservation of Energy, only it is expressed in a way that is suitable for Thermodynamic systems, namely, systems that convert Heat into Work, and Work into Heat.

Such formulation came about by James Joule, in his experiment, where an insulated container with water inside, is stirred by a stirrer, and the temperature of water is monitored by a thermometer, he observed the rise in water temperature as a result of the work done by the stirrer, he was also able in a similar experiment to produce the same

effect of a rise in temperature by a source of heat, then he was able to restore the system back into its initial temperature by surrounding the container with colder water, thus he observed that the heat rejected is the same magnitude as the work done [1].

It is important to also note the sign convention when dealing with work and heat, it is considered positive work when it is done by the system, and positive heat when it is given to the system [1].

The formulation of the First Law of Thermodynamic can be said to be that the algebraic sum of net heat and work interactions between the system and its surroundings in a thermodynamic cycle is zero [1].

Note that Work and heat are both Path Functions, meaning that going between two points describing a state of a system (such as temperature) by different paths does not necessarily have equal work or heat done. According to the First Law of Thermodynamics the cyclic sum of heat and work are equal, with emphasis on the cyclic nature [1].

Mathematically, the First Law of Thermodynamics can be expressed in two useful ways:

$$\sum_{Cycle} Q = \sum_{Cycle} W \quad (2.1)$$

Where

Q Heat [kJ]

W.... Work [kJ]

Another Way of expressing it is through a cyclic integral of infinitely small changes in heat and work.

$$\oint \delta Q - \delta W = 0 \quad (2.2)$$

Internal Energy

From mathematics it is known that a Cyclic integral of a scalar function is zero, therefore equation (2.2) is equal to cyclic integral of one scalar function.

$$\oint dE = 0 \quad (2.3)$$

Therefore

$$\delta Q - \delta W = dE \quad (2.4)$$

Where,

dE is some change in the Internal Energy of a system. [kJ]

For a finite process between state points 1 and 2, expression (2.4) can be expressed as

$$Q_{1-2} - W_{1-2} = E_2 - E_1 \quad (2.5)$$

This means that when heat and work are interacting in a system, although they are path functions, their difference is a scalar function, and that constitutes the definition of Internal Energy, it is a property of a system whose difference is equal to the difference between heat and work. Internal Energy can be in multiple forms present at the same time in a system, in the form of Intermolecular energy, Kinetic Energy, Potential Energy and any other form of energy that could be present in a system [1].

$$dE = dU + d(K.E) + d(P.E) + d(\dots) \quad (2.6)$$

dE Change in Internal Energy

dU Change in Intermolecular Energy

$d(K.E)$ Change in Kinetic Energy

$d(P.E)$ Change in Potential Energy

$d(\dots)$ Change in any other form of energy

For a closed system, where the only significant change in Internal Energy is Intermolecular Energy, and where the change in kinetic energy and potential energy is negligible, the expression for Internal Energy becomes [1],

$$dE = dU \quad (2.7)$$

Consequently, the First Law of Thermodynamics for a finite process can be expressed as,

$$Q_{1-2} - W_{1-2} = U_2 - U_1 \quad (2.8)$$

First law of Thermodynamics in a closed system

If the only work being done is at constant pressure, and expression (2.8) is expressed per unit mass, (2.8) becomes,

$$\frac{Q_{1-2}}{m} = \frac{U_2 - U_1}{m} + \frac{p(V_2 - V_1)}{m} \quad (2.9)$$

In differential specific Form,

$$\delta q = \delta u + p dv \quad (2.10)$$

Where,

q Heat addition per unit mass [kJ/kg]

u Intermolecular Energy per unit mass (Specific Internal Energy) [kJ/kg]

v Volume per unit mass (Specific Volume) [m³/kg]

p Pressure [kPa]

Enthalpy

Another system property which is useful, similar to internal energy U, is Enthalpy H [1], defined as,

$$H = U + pV \quad (2.11)$$

Where,

H.... Enthalpy [kJ]

U.... Internal Energy [kJ]

V.... Volume [m³]

In specific form,

$$h = u + pv \quad (2.12)$$

Where,

h.... specific enthalpy [kJ/kg]

When applied to a closed system with constant pressure work, expression (2.9) becomes

$$q_{1-2} = (u_2 + pv_2) - (u_1 + pv_1) = h_2 - h_1 \quad (2.13)$$

Or in differential form,

$$\delta q = dh \quad (2.14)$$

It can also be expressed in nonspecific form,

$$\delta Q = dH \quad (2.15)$$

$$\delta q = dh - vdp \quad (2.16)$$

(2.16) is the general case of (2.15) expressed per unit mass, where in our closed system with constant pressure $vdp=0$, this can also be obtained from (2.10) by expressing $pdv = d(pv) - vdp$, from the chain rule, and putting $d(u+pv) = dh$, thus obtaining (2.16) [1].

First Law of Thermodynamics in an open system

To describe the First Law of Thermodynamics in an open system, we need to quantify how much energy is associated with the flowing fluid. *Fig. 1.* Shows a control volume with fluid flowing in from the left and going out from the right.

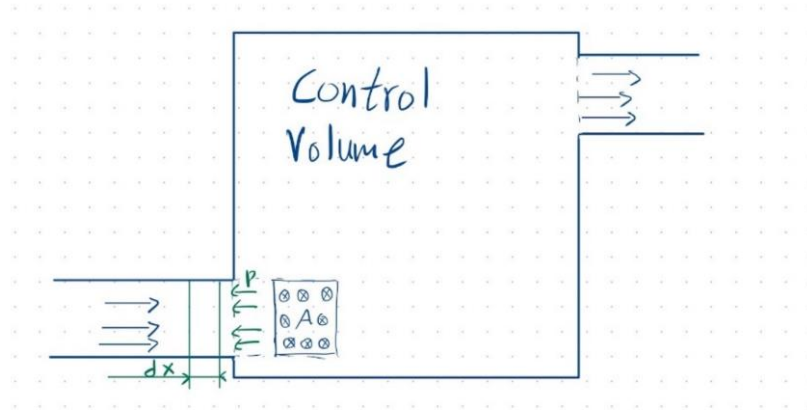


Fig. 1. Open system with Control Volume fluid flow

The force acting on a cross sectional area is,

$$F = pA \quad (2.17)$$

Work done by the force,

$$W = pAdx \quad (2.18)$$

And finally work done per unit mass,

$$\frac{W}{m} = \frac{pAdx}{\rho Adx} = \frac{p}{\rho} = pv \quad (2.19)$$

(2.19) describes the energy carried per unit mass of a flowing fluid

We can now apply the First Law of Thermodynamics on System in Fig. 2.

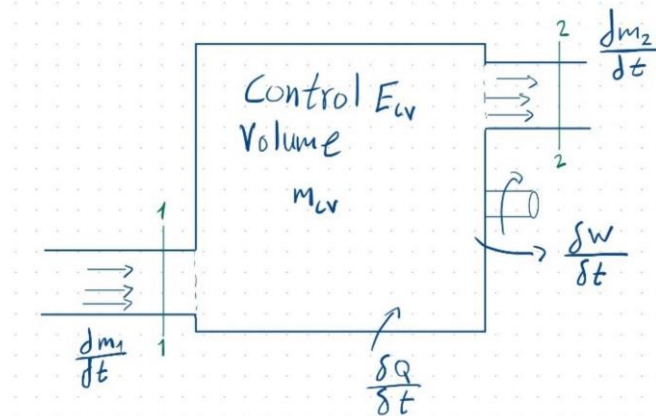


Fig. 2. General case of control volume fluid flow.

From the conservation of mass we have,

$$\frac{dm_1}{dt} - \frac{dm_2}{dt} = \frac{dm_{cv}}{dt} \quad (2.20)$$

Furthermore, Internal Energy of a flowing fluid can be decomposed into: Intermolecular energy, flow work, kinetic energy, and potential energy, that is, in specific form (per unit mass) [1],

$$e = u + pv + \frac{v^2}{2} + gh \quad (2.21)$$

Applying the First Law of Thermodynamic with addition of flowing fluid terms, and respecting the sign convention for work and heat, we can express the total energy in the control volume per unit time as,

$$\frac{dm_1}{dt} e_1 - \frac{dm_2}{dt} e_2 + \frac{\delta Q}{dt} - \frac{\delta W}{dt} = \frac{E_{cv}}{dt} \quad (2.22)$$

At steady state, the change of energy and change of the mass in the control volume must equal zero [1],

$$\frac{dm_1}{dt} = \frac{dm_2}{dt} = \frac{dm}{dt} \quad (2.23)$$

$$\frac{dE_{cv}}{dt} = 0 \quad (2.24)$$

Thus equation (2.22) becomes,

$$\frac{\delta Q}{dt} - \frac{\delta W}{dt} = \frac{dm}{dt} e_2 - \frac{dm}{dt} e_1 \quad (2.25)$$

Plugging in (2.21) in (2.25) yields

$$\frac{\delta Q}{dt} - \frac{\delta W}{dt} = \frac{dm}{dt} \left(u_2 + p_2 v_2 + \frac{v_2^2}{2} + gh_2 \right) - \frac{dm}{dt} \left(u_1 + p_1 v_1 + \frac{v_1^2}{2} + gh_1 \right) \quad (2.26)$$

Multiplying both sides by dt/dm yields,

$$\frac{\delta Q}{dm} - \frac{\delta W}{dm} = \left(u_2 + p_2 v_2 + \frac{v_2^2}{2} + gh_2 \right) - \left(u_1 + p_1 v_1 + \frac{v_1^2}{2} + gh_1 \right) \quad (2.26.1)$$

Substituting (2.12) into (2.26) we get,

$$\frac{\delta Q}{dm} - \frac{\delta W}{dm} = \left(h_2 + \frac{v_2^2}{2} + gh_2 \right) - \left(h_1 + \frac{v_1^2}{2} + gh_1 \right) \quad (2.27)$$

(2.27) is called the steady state flow equation [1]. Usually changes in kinetic energies and potential energies are negligible compared to changes in Enthalpies [1], in such cases, equation (2.27) reduces to,

$$\frac{\delta Q}{dm} - \frac{\delta W}{dm} = h_2 - h_1 \quad (2.28)$$

For a work interacting device, where there are no heat interactions, such as a Turbine, (2.28) becomes,

$$\frac{\delta W}{dm} = h_1 - h_2 \quad (2.29)$$

Similarly for a heat interacting device, where there are no work interactions, such as a boiler, (2.28) becomes,

$$\frac{\delta Q}{dm} = h_2 - h_1 \quad (2.30)$$

2.1.2 The Second Law of Thermodynamics

The Second law of Thermodynamics is one of the most fundamental laws of nature, in the sense that it ascribes a directional constraint on all natural processes. A rotating wheel can be stopped using brakes, which produce heat as a consequence of absorbing the rotational energy, but this frictional energy cannot be flow back naturally into kinetic energy, similarly, water flows from higher elevation to lower elevation, the opposite cannot occur without any external forces. This is an example of processes that have a directional constraint within its nature, but a second class of problems exist, where a system can be brought back into its initial state, such as heating an object, then cooling it down, the directional constraint in this case according to the Second Law of Thermodynamics is that, although the system can be brought back to its initial state, the surrounding of this system suffers permanent change [2].

In other words, The Second Law of Thermodynamics states that all natural phenomena are irreversible, either due to systematic constraints, such as dissipative energies, or due to induced permanent change of its surrounding, this permanent change of the surrounding is called Entropy, which will be discussed in the pages to follow. It is important to note a crucial difference between the first and second class of problems, namely, dissipative effects can be neglected or assume to be nonexistent in ideal situations, however, problems of the second class are not due to some losses, but due to a fundamental reality of nature, that is, even in ideal situations, such permanent changes to the surrounding are governed by a law of nature, and not application limitations [2]. A heat engine is a classic example of problems of the second type, where a device takes heat from a thermal reservoir, converts it into work, according to the Second Law of Thermodynamics, Heat cannot be completely converted into work even in the ideal case, meaning that in a heat engine, has to reject some heat into what is called a sink so

that work done is equal to the difference between heat supplied and heat rejected [2]. Thus, if we want to define the efficiency of a reversible heat engine as a fraction of work done by the heat engine and the heat supplied, we get,

$$\eta = \frac{W}{Q_r} = \frac{Q_s - Q_r}{Q_s} = 1 - \frac{Q_r}{Q_s} \quad (2.31)$$

Where,

W.... Work done by the heat engine

Qs.... Heat supplied to the heat engine

Qr.... Heat rejected by the heat engine

Note that this efficiency of a reversible heat engine is a function of the temperatures of the reservoir and the sink, that is temperature of heat addition and temperature of heat rejection

$$\eta = f(t_s, t_r) \quad (2.32)$$

We may also say that a reversible heat engine has the highest possible efficiency which is larger than any other irreversible engine operating between these 2 temperatures, and all reversible engines that are working between two temperatures, regardless of how they are set up, their efficiency is the same (we are neglecting all practical dissipative effects) [2].

We can see in (2.31) that due to the Second Law of Thermodynamics, a heat engine can never reach 100% efficiency even in the ideal case, as the heat rejected is never zero.

Formally the Second Law of Thermodynamics has many equivalent statements given by various scientists for different areas of applications, we will be sufficed by the Kelvin-Planck Statement which states: *It is impossible for a heat engine to produce net work in a complete cycle if it exchanges heat only with bodies at a single fixed temperature.* It is important to remember that although ideally heat can never be transformed into work, the opposite is true, namely, it is possible to completely convert work into heat [2].

Absolute Thermodynamic scale of temperature

Let us consider two reversible heat engines connected in series as in *Fig. 3*.

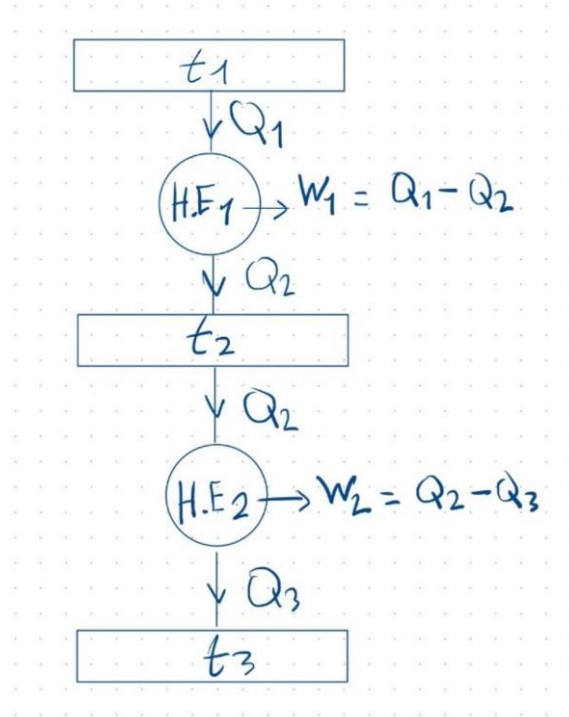


Fig. 3. Two Heat Engines connected in series.

From (2.31), (2.32) applied to the system in Fig. 3. follows,

$$\frac{Q_1}{Q_2} = f(t_1, t_2)$$

$$\frac{Q_2}{Q_3} = f(t_2, t_3)$$

If we consider an equivalent heat engine that operates between \$t_1\$ and \$t_3\$, producing work equal to the sum of our two heat engines we get,

$$\frac{Q_1}{Q_3} = f(t_1, t_3)$$

With some algebra, we can express \$f(t_1, t_2)\$ as a fraction from the other two equations as,

$$f(t_1, t_2) = \frac{f(t_1, t_3)}{f(t_2, t_3)} < = > \frac{\phi(t_1)}{\phi(t_2)} = \frac{T_1}{T_2} \quad (2.33)$$

Where,

\$T_1, T_2\$ are what we define as absolute thermodynamic scale temperature [1].

Which means we can express the efficiency of our first heat engine as,

$$\eta_{H.E.1} = 1 - \frac{Q_1}{Q_2} = 1 - \frac{T_1}{T_2} \quad (2.34)$$

Note that this is only valid for reversible heat engines, since all other irreversible engines have efficiencies lower than (2.34) for heat engines operating between the same t_1 and t_2

Entropy

Entropy is a very wide and rather deep topic; we will suffice with only its mathematical consequences necessary for our computations in the pages to follow.

Consider the first heat engine from our example in *Fig. 3*. From (2.34) we can express

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

Equivalently,

$$\frac{Q_1}{T_1} - \frac{Q_2}{T_2} = \frac{Q_1}{T_1} + \frac{(-Q_2)}{T_2} = 0$$

We respected the minus sign in such a way that our convention would still stand, for Q_2 it is negative since it is rejected from our heat engine, bearing in mind that heat engine operates in a cyclic nature, we can describe our last equation as,

$$\sum_{cycle} \left(\frac{Q}{T} \right) = 0 \quad (2.35)$$

Or in a cyclic integral fashion,

$$\oint \left(\frac{dQ}{T} \right) = 0 \quad (2.36)$$

Like the logic in (2.3), (2.36) must equal a scalar function whose cyclic integral is zero

$$\oint ds = 0 \quad (2.37)$$

Combining (2.36) and (2.37) we get,

$$ds = \frac{dQ}{T} \quad (2.38)$$

Where our scalar function ds is called Entropy [kJ/K] [2].

Specific Heat capacities

An important constant to consider is heat capacity, which describes how much heat is required to raise a substance in an insulated closed vessel by unit degree of temperature

Since heat can be supplied in two different ways, at constant volume or constant pressure (from ideal gas law), we have two specific heat capacities, namely, specific heat at constant volume c_v , and specific heat at constant pressure c_p [2]. Mathematically we can define them as follows,

$$c_v = \left(\frac{dQ}{dT} \right)_{v=const} \quad (2.39)$$

Similarly,

$$c_p = \left(\frac{dQ}{dT} \right)_{p=const} \quad (2.40)$$

Expressing the equation for the first law of thermodynamics (2.10) in nonspecific form, and dividing it by dT to equate it for our case in (2.39), we get,

$$c_v = \left(\frac{dU}{dT} \right)_{v=const} \quad (2.41)$$

Similarly, for (2.16) and (2.40) we get,

$$c_p = \left(\frac{dH}{dT} \right)_{p=const} \quad (2.42)$$

2.1.3 Fluid properties

To be able to understand and analyze vapor power cycles, which is our goal in this case, one must have some understanding of how Fluid behaves with different parameters changes (temperature, pressure, specific volume, entropy, and enthalpy), we are especially interested in what are called Phase Diagrams, they are a type of diagrams that study how a substance changes parameters as it is changing its phases. It is done to predict or evaluate any power cycle in which several such parameters are changed in such a way to produce a desired outcome [2].

p-v Phase Diagram

Suppose we have a solid that undergoes heating, it will start melting till it is liquid, then it will at some point starts boiling, if we keep providing heat, it will keep vaporizing till it is completely in a vapor state, this is done in laboratories, and depending on the diagram we want to get, different parameters are measured, in this case it is a p-v Phase Diagram, here heating is done at constant pressure, and naturally as heating is done, the substance in most cases starts to increase its specific volume, so that the heating process is done at different pressure levels, and important points are designated for each pressure, i.e. Melting Point, boiling point, sublimation point etc., theoretically there

exist infinite such combinations of heating and pressure levels [2], the resulting diagram when all these points are constructed would result in something that looks like,

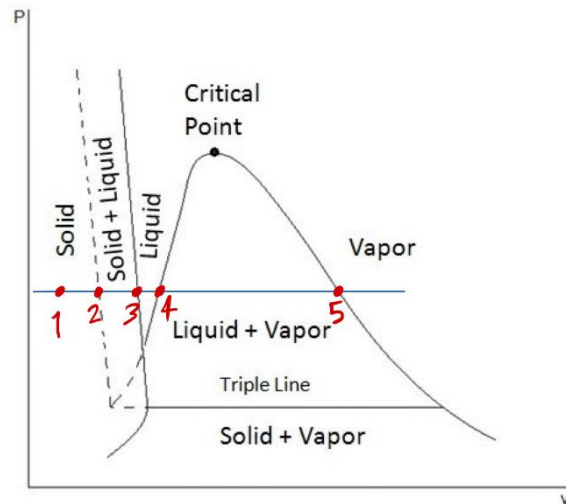


Fig. 4. p - v phase diagram [3].

I have drawn here one of the infinite constant pressure line, representing a heating process, where point 1 represents a starting point, it goes on till it reaches point 2 (melting point), the substance starts melting, as we keep on heating we reach point 3, where the substance is completely melted, similarly point 4 represents the boiling point (also called saturated liquid point), Note that points 3 and 4 lay on two lines which are both called saturated liquid lines. After point 4 the substance is now a mixture of liquid and vapor, we can keep heating till it is completely evaporated at point 5 (also called saturated vapor point), if our substance is past point 5, we call such a state a superheated vapor. We also call the heat needed to go from point 4 to 5, Latent heat, where the substance would have constant temperature as it undergoes phase change.

An interesting observation to make is that, as heating is done at higher pressures, the point between saturated liquid and saturated vapor become closer, they meet at a point called the Critical Point as shown in Fig. 4. At the critical point, the substance exists simultaneously in both liquid and vapor. A similar phenomenon happens at the triple line, but here 3 phases exist simultaneously (solid, liquid and vapor), however that is of no interest to us in our discussion.

Dryness fraction

It is very useful to define what is called dryness fraction, which tells us what the fraction of vapor to liquid is that the substance exists at, this dryness fraction can be geometrically interpreted in *Fig. 4*. as the length of a line connecting 4 and a desired point between point 4 and 5, divided by the length of the line segment 4 and 5 [1].

Mathematically we can express dryness fraction as,

$$x = \frac{m_{vapor}}{m_{liquid} + m_{vapor}} \quad (2.43)$$

Where,

m_{vapor} ... Mass of the vapor in the mixture [kg]

m_{liquid} ... Mass of the liquid in the mixture [kg]

$x = 0$ for pure liquid, $x = 1$ for pure vapor

To calculate entropy, enthalpy, specific volume for states lying between points 4 and 5 in *Fig. 4*. We can use the following formulas [2],

$$v = (1 - x)v_f + xv_g \quad (2.44)$$

Where,

x ... Dryness fraction

v_f ... Specific volume of saturated liquid (specific volume at point 4 in our *Fig. 4*.)

v_g ... Specific volume of saturated vapor (specific volume at point 5 in our *Fig. 4*.)

Similarly for entropy and enthalpy [2],

$$s = (1 - x)s_f + xs_g \quad (2.45)$$

$$h = (1 - x)h_f + xh_g \quad (2.46)$$

Where,

s_f, h_f ... Are corresponding specific entropy and enthalpy for saturated liquid.

s_g, h_g ... Are corresponding specific entropy and enthalpy for saturated vapor.

T-s Diagram

The same heating process at constant pressure can be done again, now recording two different but useful quantities, namely, temperature and specific entropy (entropy per unit of mass). The same principle of infinite constant pressure lines and plotting points of transformation from one phase to another is valid here [2]. If we do this, we get a diagram as such,

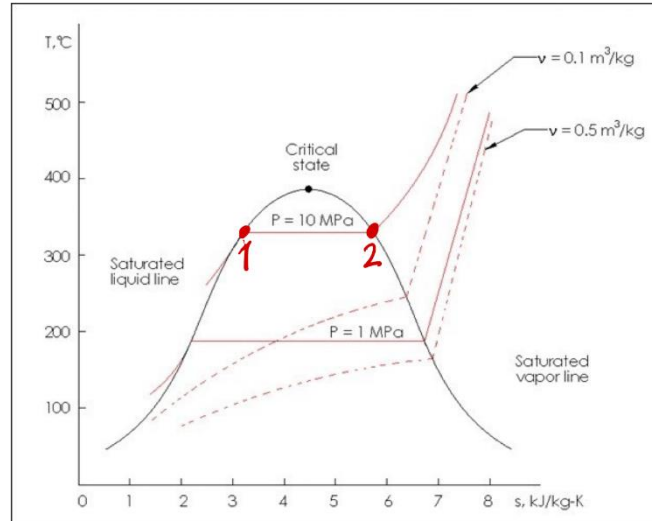


Fig. 5. *T-s diagram of heat addition at constant pressures* [4].

In Fig. 5. Only the part of liquid-vapor phase change is drawn, it is sufficient since in our power cycles we are mostly dealing with liquid to vapor phase changes, in principle there are other saturation lines as in Fig. 4. To represent solid and liquid, solid and vapor phase changes [2]. For demonstration only two constant pressure lines (isobars) are drawn. We can see as discussed in p-v diagram what is called latent heat [2], between point 1 and point 2, mathematically this heat addition can be expressed from,

$$Tds = c_p dT \quad (2.47)$$

Then,

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) = \frac{L}{T} \quad (2.48)$$

Where,

L.... Latent heat [kJ/kg]

T.... Temperature at which phase change happens

In practice we can see a more detailed diagram such as,

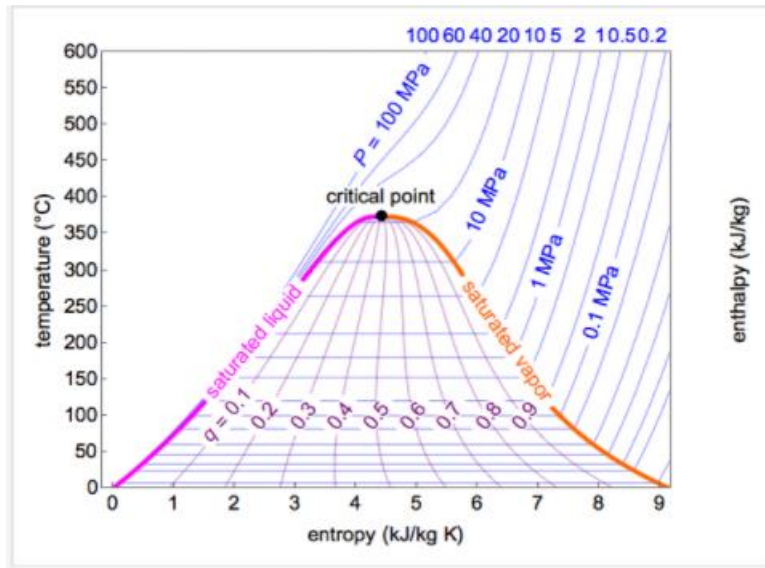


Fig. 6. T-s diagram with Isobars(blue) and dryness fraction lines(purple)[5].

Mollier (h-s) Diagram

A very useful tool for engineers is what is called Mollier diagram [2], it works on the same principle as previous diagrams, the advantage here is that one can read directly the values of enthalpy and entropy which are usually the important parameters for calculations, in this diagram are different lines of constant parameters, such as constant temperature line, constant pressure lines, and constant dryness fraction lines, it looks usually something similar to,

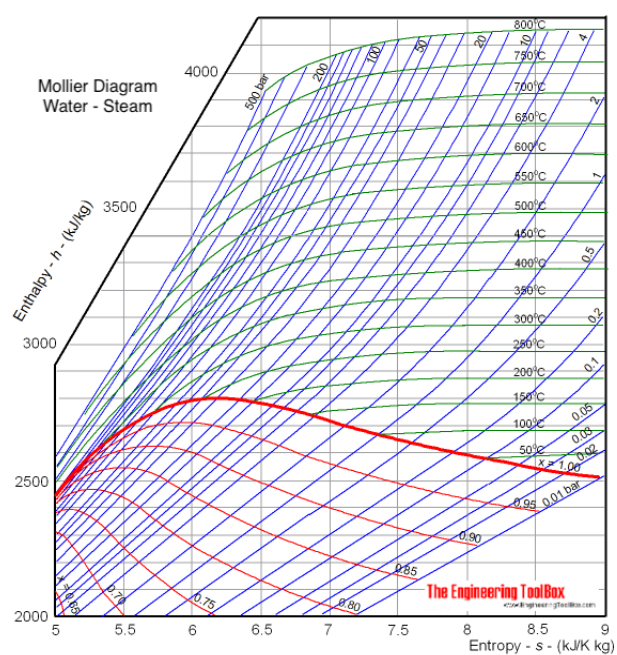


Fig. 7. Mollier diagram with constant pressure(blue), temperature (green) and dryness fraction line(red) [6].

Where $x= 1.00$ is our saturated vapor line.

p-T Diagram

We will briefly show a p-T diagram, mainly to know that it exists, it follows the same logic as the aforementioned diagrams, only interesting here is, for the latent heat phenomenon, our point 1 and 2 from Fig. 6. Is represented by a single point, due to constant Temperatures [2].

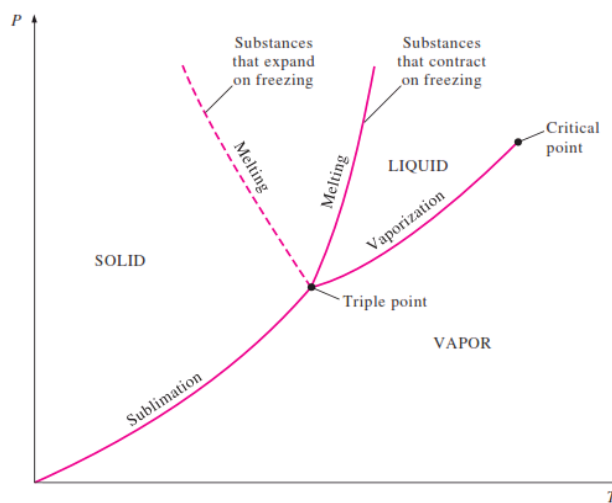


Fig. 8. p-T diagram of phase change of pure substance [7].

Where heat addition at constant pressure is represented here by a horizontal line, similar to p-v diagram [2].

2.2 Water Steam Rankine Cycle (SRC)

2.2.1 Principle of Water Steam Rankine Cycle

Most power generators today are based on a thermodynamic cycle called the Rankine cycle, where water is heated in a boiler till it become saturated vapor or further heating to superheated steam, this steam enters a turbine at high pressure and temperature, expands, resulting in the rotation of the turbine blades, which produce electricity using faraday's law [8], steam then exits the turbine and lower pressure and temperature, in order to recycle the water, a condenser is required to change the phase of steam to liquid water, then an increase in pressure is required before converting it to steam again to enter the turbine, this is achieved by using a feed pump, which feeds the water that is

coming from the condenser to the boiler after increasing its pressure, thus a cycle is obtained by closing this chain of components [9].

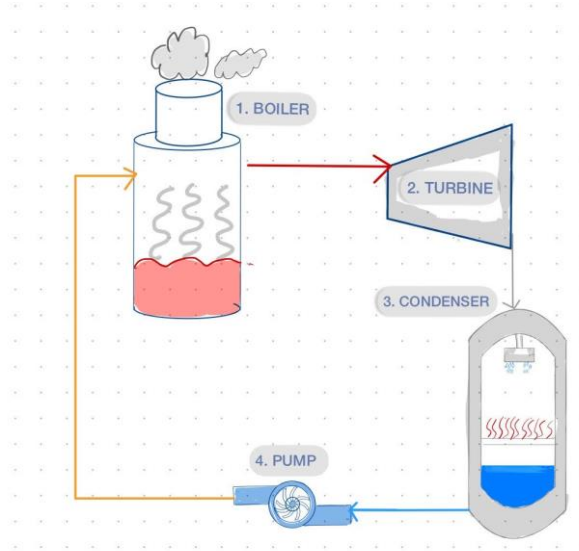


Fig 9. Rankine Cycle diagram.

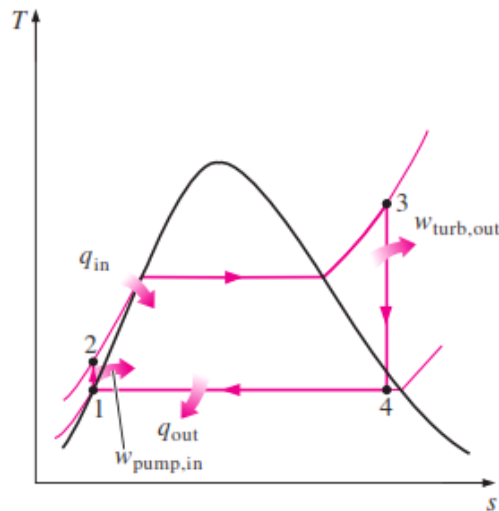


Fig 10. T-s Diagram of a Rankine Cycle [10].

Fig 10. Shows an Ideal Rankine cycle in T-s Diagram, where,
 1-2 Isentropic compression from condensing pressure to turbine pressure
 2-3 isobaric heat addition till steam is superheated
 3-4 isentropic expansion of the steam inside the turbine
 4-1 isobaric condensation of steam (heat rejection)

2.2.2 Efficiency of SRC (Water Steam Rankine Cycle)

The efficiency of the Rankine cycle is defined by how much net work we get from the cycle divided by how much heat is added to the system, in other words, how much energy we get from the system versus how much energy is spent, physically this fraction can never be larger than 1, as energy cannot be created from nothing.

$$\eta = \frac{W_{net}}{Q_{add}} = \frac{W_T - W_P}{Q_{add}} \quad (2.49)$$

$$\eta = \frac{(h_1 - h_2) - (h_3 - h_4)}{h_1 - h_4} \quad (2.50)$$

As seen in *Figure 11*. The efficiency can be increased in three possible ways. Increase the work done by the turbine, and/or decrease the work necessary for the pump, and/or decrease the necessary heat added.

Another way to express the efficiency of a Rankine cycle is the expression for Carnot efficiency,

$$\eta = 1 - \frac{T_{m23}}{T_{41}} \quad (2.51)$$

Where,

T_{m23} Mean temperature of heat addition in step 2-3 in *Fig 10*.

T_{41} Temperature of heat rejection in step 4-1 in *Fig 10*.

Since the Rankine cycle does not add heat isothermally, a mean temperature of heat addition must be used so that the area under the original cycle in T-S diagram and a new rectangular cycle are equal (Carnot equivalent) [11].

2.2.3 Limitations of SRC

2.2.4 Example

Let us consider an example to illustrate how to deal with basic cycle calculations such as masses flow rates, ideal versus real expansions and compressions, and ideal thermodynamic efficiency.

Consider a Power plant with power output of 12.5 MW, inlet steam temperature t_1 of 490 Celsius, at pressure p_1 of 85 Bars, where it is rejected and condensated at a temperature t_2 of 43 Celsius, Temperature outlet of deaerator of 105 Celsius turbine efficiency of 90%, pump efficiency of 75%, generation and mechanical efficiency of 99%, using steam tables, we calculate the input and output work, input heat and output heat, and calculate the ideal efficiency of a water steam Rankine cycle when the cycle is given according to *Fig 12*.

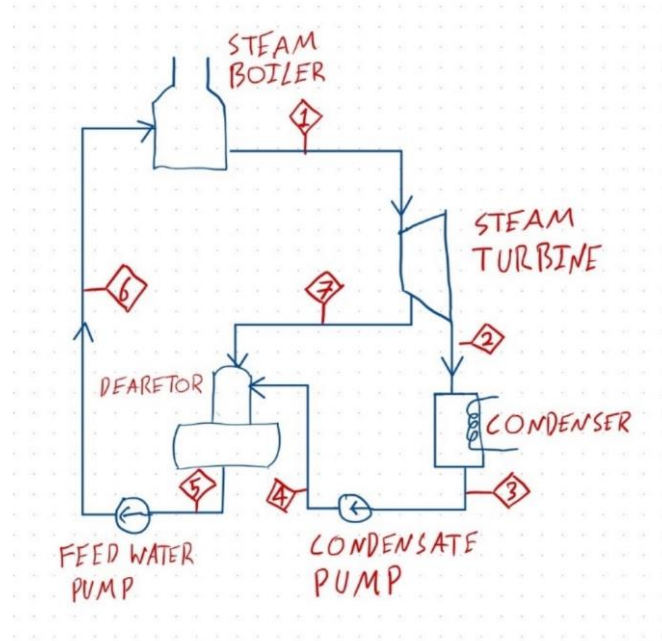


Fig 11. water Rankine cycle power plant.

To calculate all our parameters of work and heat, we need to know the specific enthalpy of each of the points 1 through 7, using the provided initial conditions and steam tables. After expansion, the saturation pressure corresponding to t_2 is,

$$p_{sat}(t_2) = p_2 = 0.0865 \text{ Bars}$$

Now we can proceed to look for h_1 and h_2 from steam tables,

$$h_1(p_1, t_1) = 3386.26 \text{ kJ/kg (Point 1)}$$

$$s_{2Adiabatic} = s_1(p_1, h_1) = 6.659 \text{ kJ/kgK}$$

$$h_{2Adiabatic}(p_2, s_2) = 2092 \text{ kJ/kg}$$

h_2 adiabatic assumes no heat losses in the turbine, to get the real h_2 , we can use the equation for the thermodynamic efficiency of a turbine

$$h_{2real} = h_1 - \eta_{turbine}(h_1 - h_{2Adiabatic}) = 2219.62 \text{ kJ/kg (Point 2)}$$

Checking steam quality,

$$x(p_2, h_{2real}) = 0.85$$

$$h_3 = h_f(p_2) = 180.078 \text{ kJ/kg (Point 3)}$$

$$s_3 = s_f(p_2) = 0.6122 \text{ kJ/kgK}$$

$$p_4 = p_{Deaerator} = p_{sat}(T_{Deaerator}) = 1.209 \text{ Bars}$$

$$h_{4Adiabatic}(p_4, s_3) = 180.174 \text{ kJ/kg}$$

Similar to calculation in point 2 but using the efficiency of pump,

$$h_{4real} = h_3 + \frac{h_{4Adiabatic} - h_3}{\eta_{pump}} = 180.206 \text{ kJ/kg (Point 4)}$$

$$h_5 = h_f(p_4) = 440.21 \text{ kJ/kg (Point 5)}$$

$$s_5 = s_f(p_4) = 1.363 \text{ kJ/kgK}$$

$$h_{6Adiabatic}(p_1, s_5) = 448.98 \text{ kJ/kg}$$

Similar to point 4,

$$h_{6real} = h_5 + \frac{h_{6Adiabatic} - h_5}{\eta_{pump}} = 451.90 \text{ kJ/kg (Point 6)}$$

$$h_{7Adiabatic} = h_{Adiabatic \text{ extraction}}(p_{deaerator}, s_1) = 2443.13 \text{ kJ/kg}$$

Similar to point 2,

$$h_{7real} = h_1 - \eta_{turbine}(h_1 - h_{7Adiabatic}) = 2535.65 \text{ kJ/kg (Point 7)}$$

Doing the mass flow enthalpy balance for the deaerator yields,

$$h_5 \dot{M} = h_7 \dot{m} + h_4 (\dot{M} - \dot{m})$$

Defining mass flow fraction alpha, and solving for alpha in the enthalpy balance equation yields,

$$\alpha = \frac{\dot{m}}{\dot{M}}$$

$$\alpha = \frac{h_5 - h_4}{h_7 - h_4} = 0.11$$

Defining enthalpy extraction fraction of expansion mu,

$$\mu = \frac{h_{7real} - h_{2real}}{h_1 - h_{2real}} = 0.275$$

Finding M dot from the equation for output Power,

$$P = \dot{M}(h_1 - h_2)\eta_{turbine}\eta_{generator}\eta_{mechanical}(1 - \alpha\mu)$$

Then,

$$\dot{M} = \frac{P * 1000}{(h_1 - h_2)\eta_{turbine}\eta_{generator}\eta_{mechanical}(1 - \alpha\mu)} = 10.766 \text{ kg/s}$$

The cycle thermodynamic efficiency,

$$\eta_{cycle} = \frac{\dot{W}}{Q_{supplied}} = \frac{P * 1000}{\dot{M}(h_1 - h_6)} = 39.811\%$$

And finally, water consumption from the cooling water loop at 1 atm pressure,

$$\dot{m}_w = \frac{Q_{supplied}}{L} = 8.371 \text{ kg/s}$$

2.3 Organic Rankine Cycle (ORC)

2.3.1 Principle of Organic Rankine Cycle

Like the Steam Rankine Cycle, the Organic Rankine cycle uses Organic working fluids instead of water, with a construction almost identical to the aforementioned cycle, the Organic Rankine cycle main advantage is the ability to use Organic working fluids which have low boiling point, making it suitable for small and medium power plants (up to 10 MW), with the possibility to use the low exhaust temperature as heating for a local district. It also eliminates the need to superheat the working fluid to avoid liquid in the turbine, this follows from the thermodynamic properties of the organic working fluids, making it almost always expand in such a way to have superheated vapor after expansion [12]. See Fig 11.

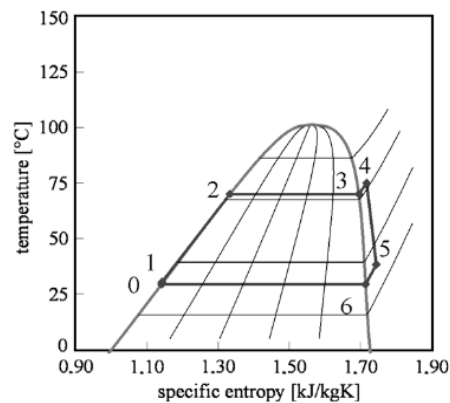


Fig 11. T-s diagram for fluid R134a [12].

Where,

0-1.... Feed pump compression

1-2.... Preheating (exhaust vapor of turbine is used to reheat part of our compressed fluid)

2-3.... Evaporation

3-4.... Superheating

4-5.... Expansion into lower pressure superheated vapor

5-6.... Desuperheating

6-0.... Condensation

2.3.2 Comparison of ORC & SRC

As the main difference between the two cycles is the working fluid, it is therefore useful to look at a table with the most common organic working fluids properties, comparing them with water in Table 1. [13].

Fluid	Formula/name	MW [kg/mol]	T_{crit} [°C]	p_{crit} [bar]	BP [°C]	E_{evap} [kJ/kg]
Water	H ₂ O	0.018	373.95	220.64	100.0	2257.5
Toluene	C ₇ H ₈	0.092	318.65	41.06	110.7	365.0
R245fa	C ₃ H ₃ F ₅	0.134	154.05	36.40	14.8	195.6
n-pentane	C ₅ H ₁₂	0.072	196.55	33.68	36.2	361.8
cyclopentane	C ₅ H ₁₀	0.070	238.55	45.10	49.4	391.7
Solkatherm	solkatherm	0.185	177.55	28.49	35.5	138.1
OMTS	MDM	0.237	290.98	14.15	152.7	153.0
HMDS	MM	0.162	245.51	19.51	100.4	195.8

Table 1. Working fluids common name, Chemical name, Molar Weight, Critical Temperature (critical point temperature), critical pressure, Boiling point, and Evaporation heat energy [13].

We can see in Table 1. All the fluids except the last two have lower boiling point than water, even for temperatures higher than 100 C, some working fluids provide higher gross cycle efficiency, making the Organic Rankine cycle a flexible system that can be adjusted to specific needs and requirements of its application [13]. See Fig 12.

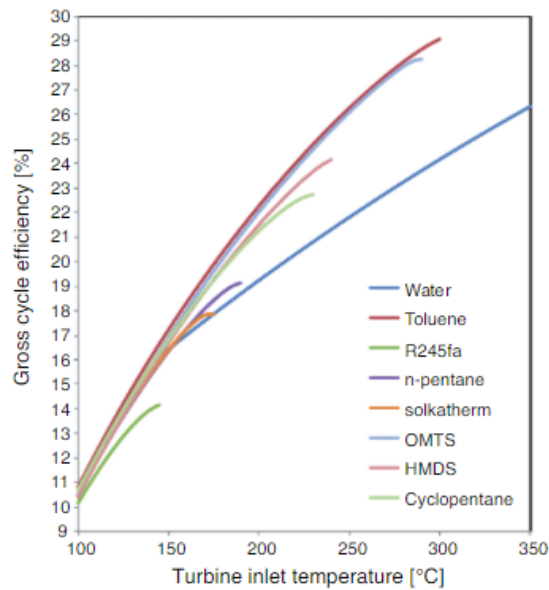


Fig 12. Working Fluids in Table 1. Gross Cycle Efficiency in percentage vs Turbine inlet Temperature [13].

2.3.3 Example

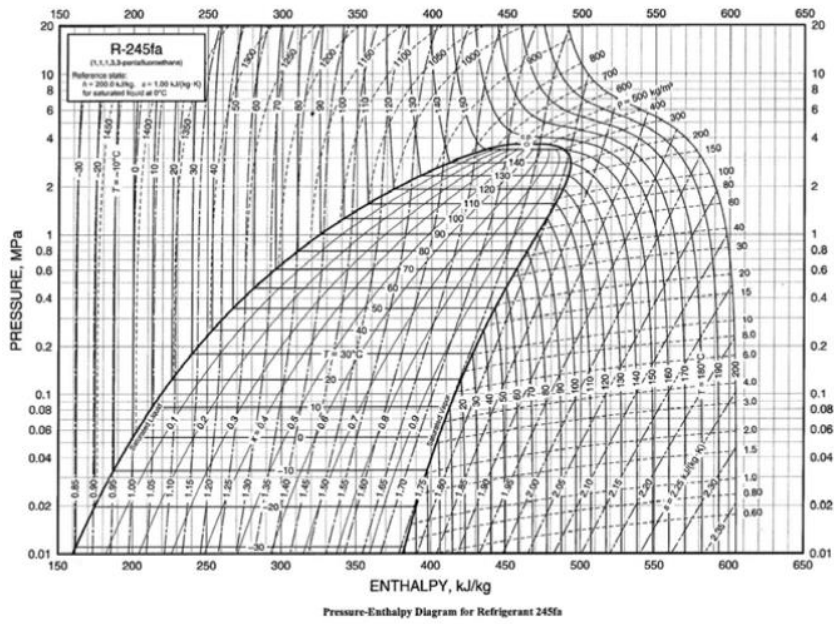


Fig 13. P-h Diagram of Working fluid R-245fa [14].

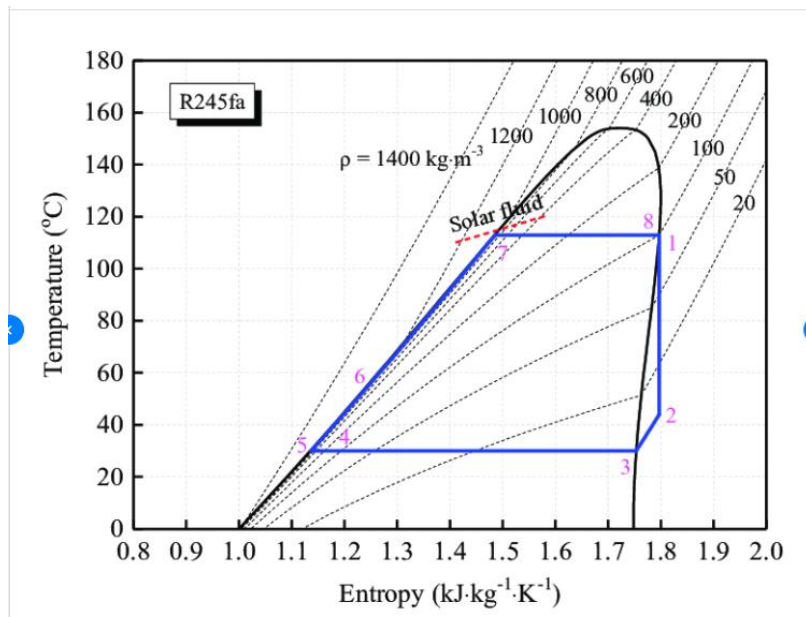


Fig 14. T-s Diagram of working fluid R-245fa [15].

Working Fluid: R245fa

Given Parameters:

$$p_1 = 30 \text{ Bars}, t_1 = 220 \text{ C}^\circ, t_{\text{Cond}} = 40 \text{ C}^\circ, \eta_{\text{Turbine}} = 0.9, \eta_{\text{Pump}} = 0.75, \eta_{\text{mechanical}} = 0.99, P = 12.5 \text{ MW}$$

Calculated Parameters:

$$h_1(p_1, t_1) = 606.60 \text{ kJ/kg (Point 1)}$$

$$s_1 = s_2 = 2.0564 \text{ kJ/kgK}$$

$$p_2(x = 0, t_{cond}) = 2.4941 \text{ Bars}$$

$$t_2(p_2, s_2) = 145.58 \text{ C}^\circ$$

$$h_{2Adiabatic}(p_2, s_2) = 542.17 \text{ kJ/kg}$$

$$h_{2real} = h_1 - \eta_{turbine}(h_1 - h_{2Adiabatic}) = 548.61 \text{ kJ/kg (Point 2)}$$

$$h_3(x = 0, p_2) = 252.84 \text{ kJ/kg (Point 3)}$$

$$s_3 = s_4(x = 0, p_2) = 1.1802 \text{ kJ/kgK}$$

$$h_{4Adiabatic}(s_4, p_1) = 254.95 \text{ kJ/kg}$$

$$h_{4real} = h_3 + \frac{h_{4Adiabatic} - h_3}{\eta_{pump}} = 255.66 \text{ kJ/kg (Point 4)}$$

$$q_{supplied} = h_1 - h_4 = 350.94 \text{ kJ/kg}$$

$$q_{rejected} = h_2 - h_3 = 295.78 \text{ kJ/kg}$$

$$\dot{M} = \frac{P * 1000}{(h_1 - h_2)\eta_{turbine}\eta_{generator}\eta_{mechanical}} =$$

$$\eta_{cycle} = 1 - \frac{q_{rejected}}{q_{supplied}} = 15.7 \%$$

And finally, water consumption from the latent heat of cooling water loop at 1 atm pressure,

$$\dot{m}_w = \frac{\dot{Q}_{supplied}}{L} = 29.708 \text{ kg/s}$$

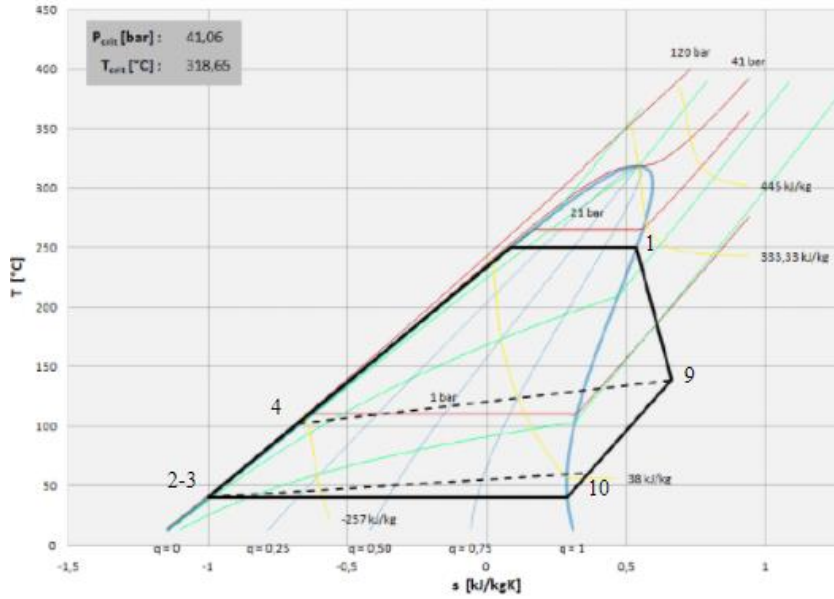


Figure 3 : T-s diagram of ORC with toluene

Fig 15. T-s Diagram of Working fluid Toluene [16].

Working Fluid: Toulene

Given Parameters:

$$p_1 = 10.514 \text{ Bars}, t_1 = t_g(p_1) = 220 \text{ C}^\circ, t_{cond} = 40 \text{ C}^\circ, \eta_{Turbine} = 0.9, \eta_{Pump} = 0.75, \eta_{mechanical} = 0.99, P = 12.5 \text{ MW}$$

Calculated Parameters:

$$h_1(p_1, x = 1) = 514.08 \text{ kJ/kg (Point 1)}$$

$$s_1(p_1, x = 1) = s_2 = 1.1018 \text{ kJ/kgK}$$

$$p_2(x = 0, t_{cond}) = 0.0783 \text{ Bars}$$

$$t_2(p_2, s_2) = 90.32 \text{ C}^\circ$$

$$h_{2Adiabatic}(p_2, s_2) = 336.25 \text{ kJ/kg}$$

$$h_{2real} = h_1 - \eta_{turbine}(h_1 - h_{2Adiabatic}) = 354.03 \text{ kJ/kg (Point 2)}$$

$$h_3(x = 0, p_2) = -130.98 \text{ kJ/kg (Point 3)}$$

$$s_3 = s_4(x = 0, p_2) = -0.3807 \text{ kJ/kgK}$$

$$h_{4Adiabatic}(s_4, p_1) = -131.39 \text{ kJ/kg}$$

$$h_{4real} = h_3 + \frac{h_{4Adiabatic} - h_3}{\eta_{pump}} = -130.98 \text{ kJ/kg (Point 4)}$$

$$q_{supplied} = h_1 - h_4 = 645.06 \text{ kJ/kg}$$

$$q_{rejected} = h_2 - h_3 = 486.65 \text{ kJ/kg}$$

$$\eta_{cycle} = 1 - \frac{q_{rejected}}{q_{supplied}} = 24.55\%$$

And finally, water consumption from the latent heat of cooling water loop at 1 atm pressure,

$$\dot{m}_w = \frac{Q_{supplied}}{L} = 17.01 \text{ kg/s}$$

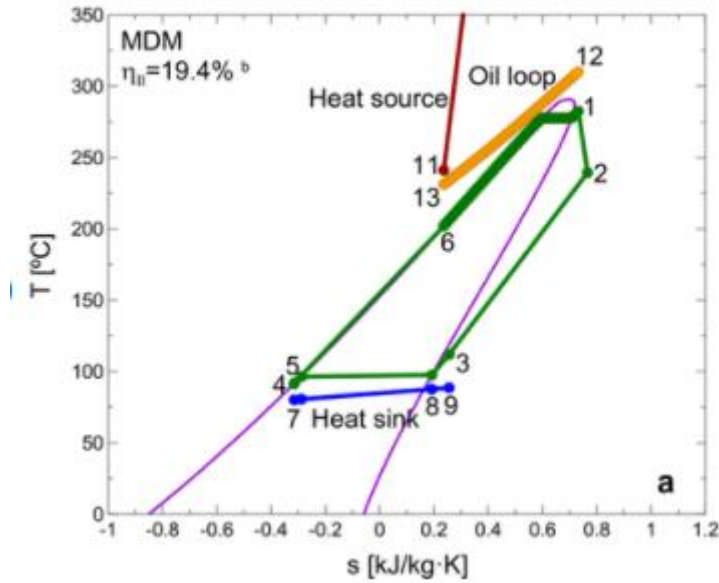


Fig 16. T-s Diagram of Working fluid OMTS (MDM) [17].

Working Fluid: OMTS (MDM)

Given Parameters:

$$p_1 = 4.4357 \text{ Bars}, t_1 = t_g(p_1) = 220 \text{ C}^\circ, t_{cond} = 40 \text{ C}^\circ, \eta_{Turbine} = 0.9, \eta_{Pump} = 0.75, \eta_{mechanical} = 0.99, P = 12.5 \text{ MW}$$

Calculated Parameters:

$$h_1(p_1, x = 1) = 265.70 \text{ kJ/kg (Point 1)}$$

$$s_1(p_1, x = 1) = s_2 = 0.5601 \text{ kJ/kgK}$$

$$p_2(x = 0, t_{cond}) = 0.01245 \text{ Bars}$$

$$t_2(p_2, s_2) = 163 \text{ C}^\circ$$

$$h_{2Adiabatic}(p_2, s_2) = 173.865 \text{ kJ/kg}$$

$$h_{2real} = h_1 - \eta_{turbine}(h_1 - h_{2Adiabatic}) = 183.049 \text{ kJ/kg (Point 2)}$$

$$h_3(x = 0, p_2) = -221.91 \text{ kJ/kg (Point 3)}$$

$$s_3 = s_4(x = 0, p_2) = -0.6030 \text{ kJ/kgK}$$

$$h_{4Adiabatic}(s_4, p_1) = -221.36 \text{ kJ/kg}$$

$$h_{4real} = h_3 + \frac{h_{4Adiabatic} - h_3}{\eta_{pump}} = -221.18 \text{ kJ/kg (Point 4)}$$

$$q_{supplied} = h_1 - h_4 = 486.88 \text{ kJ/kg}$$

$$q_{rejected} = h_2 - h_3 = 404.96 \text{ kJ/kg}$$

$$\eta_{cycle} = 1 - \frac{q_{rejected}}{q_{supplied}} = 16.82\%$$

And finally, water consumption from the latent heat of cooling water loop at 1 atm pressure,

$$\dot{m}_w = \frac{Q_{supplied}}{L} = 27.382 \text{ kg/s}$$

2.4 The Hygroscopic Cycle Technology (HCT)

2.4.1 Principle of HCT

Although the principle of hygroscopy was studied extensively in physics and chemistry, The hygroscopic cycle was developed in 2010 by the Spanish Engineer Francisco J. Rubio-Serrano [18]. It relies on the fact that some compounds called hygroscopic compounds(salts) have the tendency to absorb water in liquid or vapor from the atmosphere around it, although they vary in how much moisture they absorb, all hygroscopic compounds have a crucial property which allows this cycle to thrive over a normal Rankine cycle. Mainly the fact that when these salts (NaCl, LiBr, NaOH, etc.) mix with normal water, they increase the boiling point temperature in what is called Boiling Point Elevation, thus increasing the condensation temperature, while also decreasing the freezing point [19] [20].

That is the reason why these hygroscopic compounds have been in use since the 1950's in absorption cycles used for refrigeration [19]. This follows the idea to replace the Condenser in a traditional Rankine cycle, with an Absorber, using the hygroscopic

properties of salts to absorb and adsorb the exhaust steam from the turbine, resulting in a condensation of the exhaust steam [20].

2.4.2 Components of Hygroscopic Cycle

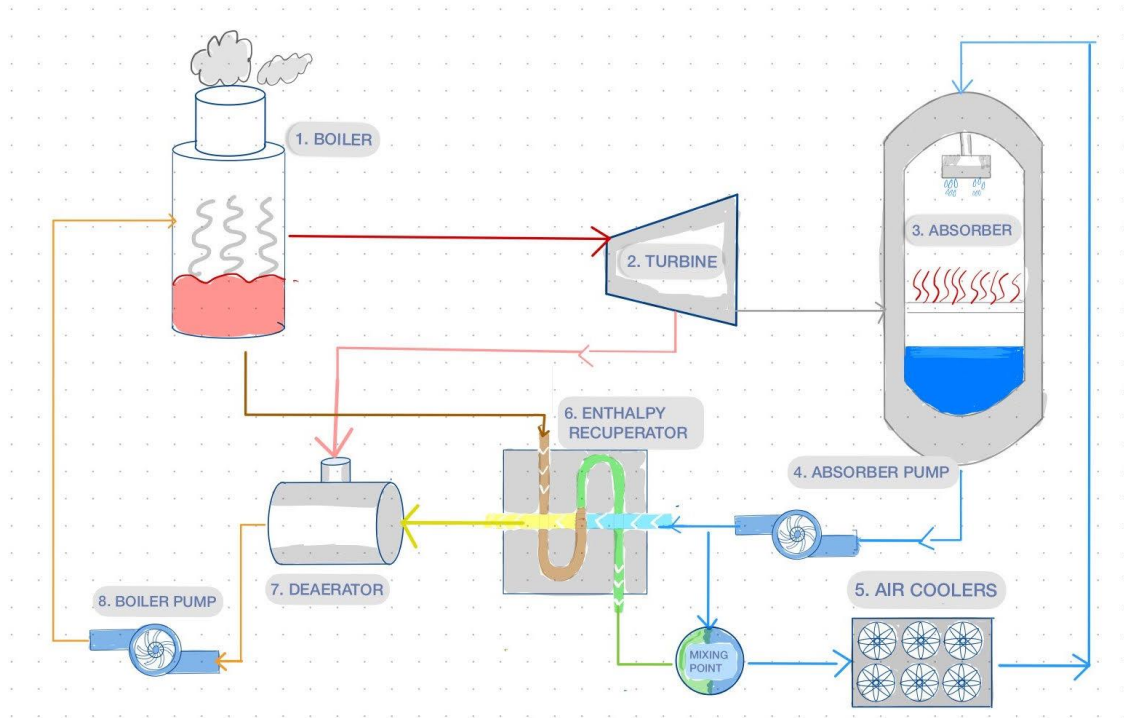


Fig 17. Power Plant based on the Hygroscopic Cycle.

Where:

- 1- Steam Boiler for heating the water to steam [20].
- 2- Steam turbine [20].
- 3- The Absorber: (main component of the cycle), here the hygroscopic compounds are added to condensate the exhaust steam [20].
- 4- Condensate Pump: to elevate the pressure of the solution, where part of the solution is going to the air coolers, to be used again in the absorber after cooling down, and another part is passed to the enthalpy recuperator, which is basically a heat exchanger, allowing the boiler blow down to be used for heating the solution before part of it is discharged and part is passed on to the air coolers [19].
- 5- Air coolers: here is where all the water savings comes in, where in this configuration of the hygroscopic cycle, no water is needed to cool down the condensate (unlike a traditional Rankine cycle, where water runs in pipes that go through the condenser and exchanges heat with the exhaust steam, causing it ultimately to condense) [20].

6- Enthalpy Recuperator: Each boiler has what's called a boiler blow down [21]. It is some part of the boiler water that is rejected, to control the parameters inside the boiler. (Some compounds that could cause corrosion, scales, and other potentially harmful components), but since this blowdown is rejected from the boiler, naturally it is at a very high temperature and carries energy with it, so in order not to lose efficiency, a heat exchanger is placed [20].

7- Deaerator: this acts as a filter, eliminating all the bubbles, microbubbles, and corrosive gasses, mainly oxygen and CO₂ from the dilute solution [19].

8- Feed Pump: this pump is used to pass the water on to the boiler, after increasing its pressure to the required operating pressure of the turbine [19].

2.4.3 Efficiency factors in HCT

Why is the hygroscopic cycle more efficient than a traditional Rankine cycle, based on the theory we have already discussed?

Since these hygroscopic compound elevates our boiling point and condensation point under the same pressure compared to normal water, this means in practical terms, that our cooling equipment need not work as much, hence the elimination of the need for water cooling towers and water cooling altogether, this also means that we are not limited by the ambient temperature or so called cold sink temperature for cooling just like in a traditional Rankine cycle, another equivalent practical advantage is that, for the same cooling temperature, the condensing pressure is much lower, allowing us to boost the turbine power output (increasing the area under a T-S diagram), without sinking the dryness fraction below 0,85 [18].

The efficiency of Hygroscopic Cycle depends on mainly two factors, one being the cooling temperature, and the concentration of salts in the solution, mainly LiBr (Lithium Bromide), which is the most popular compound used in absorption cycles in refrigeration due to its exceptional hygroscopic properties [20].

To date, there is only one operating 12.5 MW biomass power plant which has been operating since 1996 in the province of Cordoba, Spain, and in 2010, it implemented Hygroscopic Cycle Technology (HCT). There is also a test plant in Gijon, Spain, both use HCT which is built and owned by the company IMASA, at which the developer of

this technology works. We are going to show their data and experiment to see the comparison between it and a traditional Rankine cycle [18].

Since HCT has been implemented in the 12.5 MW biomass, the amount of water saved is 229,200 m³ per year, since all cooling is done by air coolers [18].

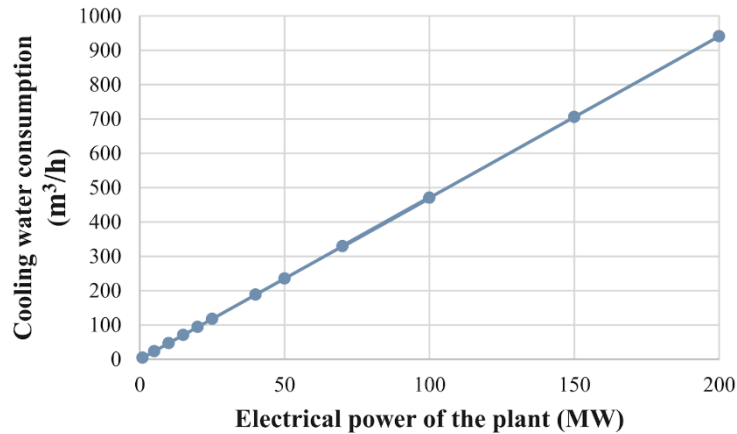


Fig 18. Required Cooling water as a function of Electrical power of the plant [18].

This is the main reason why HCT is considered an attractive option for Concentrated Solar Power CSP in Semi-Deserted areas with high amounts of annual solar energy, or any other type of power plant in an area where water is scarce [18].

The following two charts are from the test plant located in Gijon Spain [18].

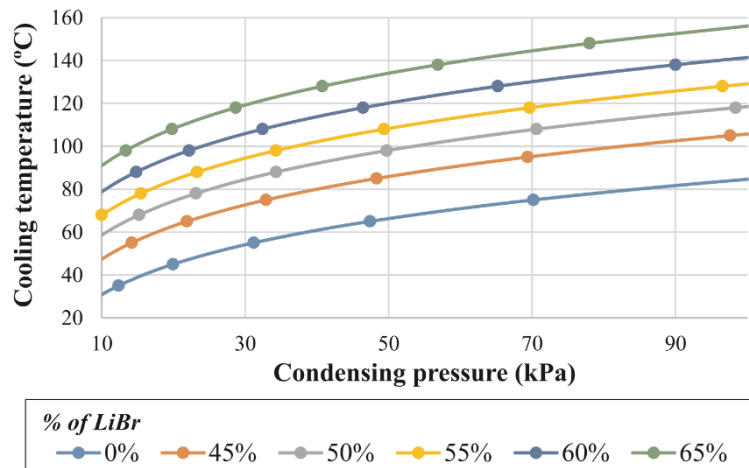


Fig 19. Cooling temperatures that are needed for condensing pressures between 10-90 kPa for different concentrations of Lithium Bromide in our solution [18].

Here is interesting to notice, as we increase the concentration, for the same pressure, we have more ceiling for cooling temperature, lowering our power input required for the cooling equipment. (See Figure. 20 below)

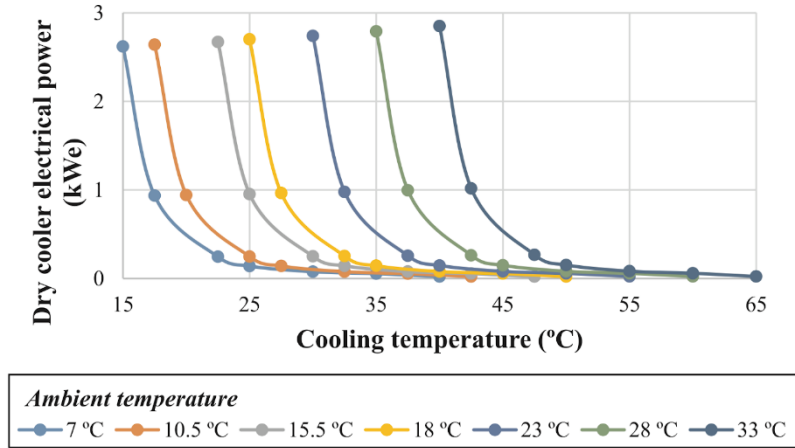


Fig 20. Required power of air coolers depending on cooling temperatures for different ambient temperatures [18].

Here is illustrated clearly what is called Cold Sink temperature, we can see that for each ambient temperature, we cannot go below it, but with HCT, we need not go that low near it because our condensing temperature for a given pressure is much higher than for a traditional Rankine cycle, this is further illustrated in next two figures (see Figure 21. And Figure 22. Below) [18].

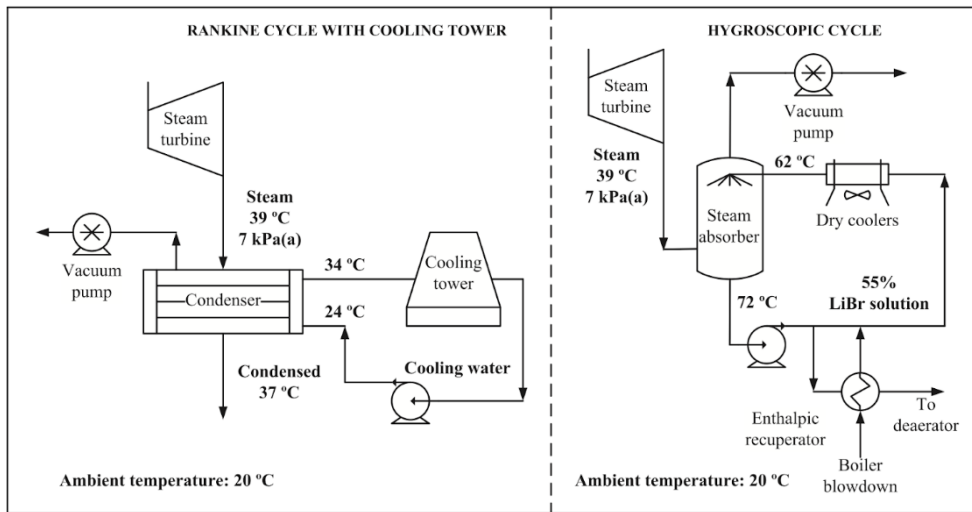


Fig 21. Rankine vs HCT for the same condensing pressure [22].

Notice the inlet temperature of water to the Condenser 24 C vs inlet temperature of LiBr solution to the Absorber 62 C for the same temperature and pressure of exhaust turbine steam 39 C at 7 kPa (a). This is one of two ways to take advantage of HCT. In this configuration, we have lower consumption of power for the air coolers, and in both configurations (look Figure 9.) 100% reduction in cooling water consumption.

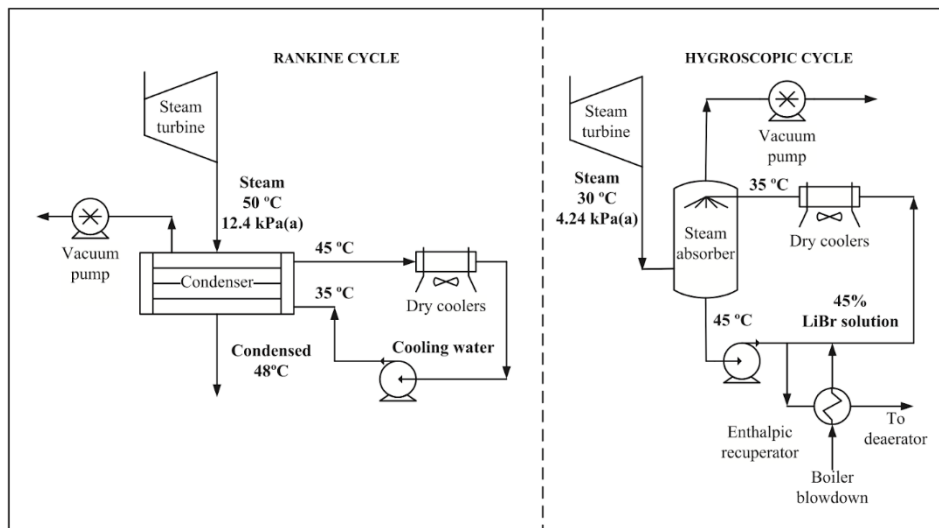


Fig 22. Lower exhaust turbine pressure and temperature for the same inlet cooling temperature [22].

The Second way HCT having an advantage over Rankine cycle can be seen in Figure 9., where we can lower the pressure much more and still have a dryness fraction over or equal to 0,85 [22].

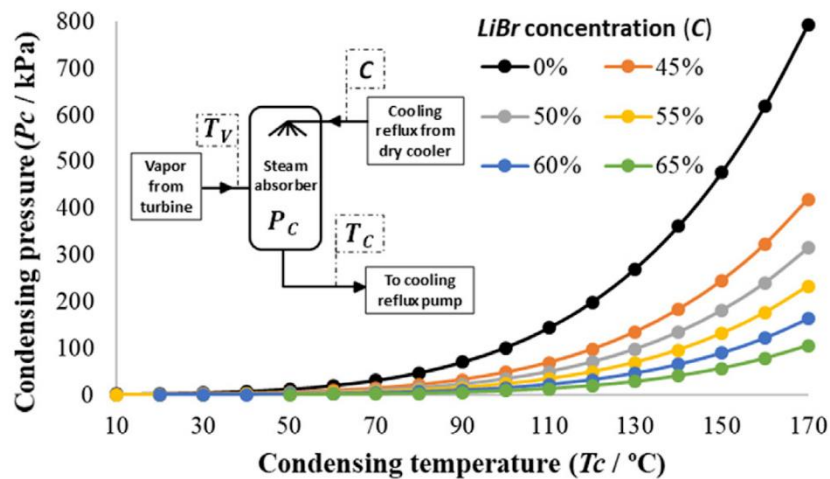


Fig 23. Condensing Pressure vs Condensing temperature for different concentrations of LiBr [22].

This graph clearly illustrates the effect of hygroscopic compounds on working temperatures and pressures. I mainly included this graph because it is the closest to a T-S diagram, perhaps a T-S diagram could be constructed and compared to a Rankine traditional cycle.

Mr. Serrano claims in his paper [22], that it was experimentally found that the electrical efficiency decreases when the concentration of LiBr is below 45% [22].

He does however present this data in Figure 11. Below for concentrations greater than 45% [22].

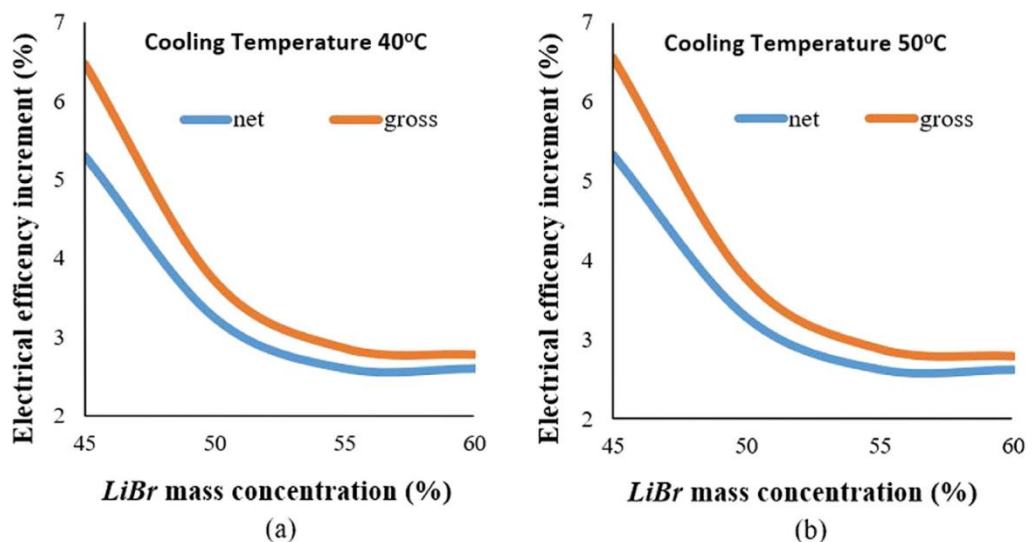


Fig 24. Increment in electrical efficiency for LiBr mass concentration from 45 to 60%[22]

One can see that after approximately 55%, the increment of electrical efficiency is almost constant, therefore a concentration of 45% is optimal for Hygroscopic solution using LiBr in HCT.

3 Processing data and methodology

When trying to compute fluid properties in a Hygroscopic cycle, one must first define which solution is being used, in our case, we will use the same fluid as in [22]. Namely Lithium Bromide (LiBr) water solution, mainly to compare the computed data with the data acquired through experimentation, this is important since, till the time of writing this dissertation paper, it was noticed that the only provider of experimental data is the same person who has patented the technology.

Another reason why choosing Lithium Bromide solution might be justified is that, it is widely used in refrigeration cycles, so that we can use existing tables for calculating fluid properties, with that being said, it was not an easy task, and it has proposed some limitations to our computations, mainly because these tables only go up to a certain temperature (about 225 C), in the experiments it was mentioned in [22]. that the temperature in a test plant at the turbine inlet was around 500 C.

After experimenting with different LiBr fluid properties tables, the main paper that was used is by J. Klomfar and corresponding author J. Patek from the Czech Academy of Sciences [23]. Mainly because it is one of the only tables that gives us the range of 225 C over full composition range of LiBr concentration [23]. Other tables either have a lower temperature range, or a limited concentration range of LiBr. It is important to note that such tables usually only give the fluid properties of a saturated fluid.

The computational model in [23]. uses tables of coefficients that are plugged in corresponding equations of each fluid property (*see Table 2.*). Hence, we have written a MATLAB code (*see Appendix A.*) based on such tables, this code works in the way of having Temperature and Weight Fraction as inputs, and specific heat capacity at constant pressure, specific enthalpy, specific entropy, density, and pressure as outputs, and all these properties correspond to a saturated fluid.

i	m_i	n_i	t_i	a_i
1	3	0	0	-2.41303×10^2
2	4	5	0	1.91750×10^7
3	4	6	0	-1.75521×10^8
4	8	3	0	3.25430×10^7
5	1	0	1	3.92571×10^2
6	1	2	1	-2.12626×10^3
7	4	6	1	1.85127×10^8
8	6	0	1	1.91216×10^3

$T_c=647.096$ K; RMS=2.1%.

Table 2. An example from J. Klomfar paper, Table of Coefficients and Exponent for the pressure function [23].

The computation will be in the fashion of defining all the properties at inlet and outlet at each equipment from 1-8, writing the mass flow balance for each equipment, then going further to calculate efficiency. For ease of reference for the following computation, it is helpful to bring back Fig. 17 because it demonstrates clearly, all the parts of the Hygroscopic cycle, detailed description of these points was mentioned in the section about Components of Hygroscopic Cycle. It is also worth remembering that the streams of fluid flow in Fig. 17 that are in Red, Pink and Gray are pure Water Steam, therefore for such parts of the cycle, the same tables are used as for the Steam Rankine Cycle example calculations.

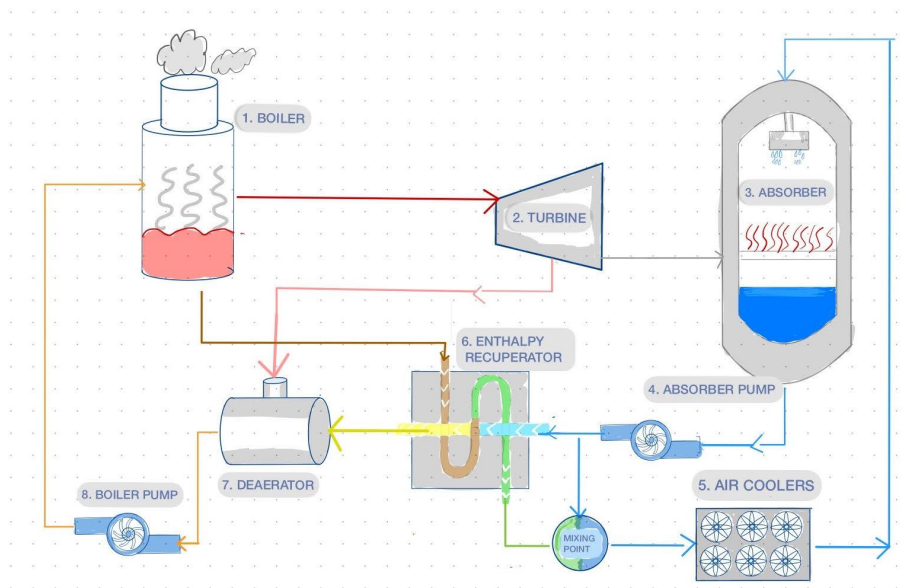


Fig 17. Power Plant based on the Hygroscopic Cycle..

Nomenclature

$x_{i n j}$	fluid property x , at equipment i , at inlet n , inlet number j
$x_{i o j}$	fluid property x , at equipment i , at outlet o , outlet number j
h	Specific enthalpy [kJ/kg]
s	Specific entropy [kJ/kg K]
p	Pressure [kPa]
w	Weight fraction of Lithium Bromide solution $w=0$ for pure steam
ρ	Density [kg/m ³]
v	Specific Volume [m ³ /kg]
c_p	Specific heat capacity at constant pressure [kJ/kg K]
\dot{m}	Normalized mass flow rate [kg/s]
Eq_i	Equipment number i
Source/Sink	Source of inlet, Sink of outlet, e.g., Inlet 1 source Eq_1 , Outlet 1 sink Eq_3 Inlet 1 comes from Equipment 1, and the output 1 goes to Equipment 3
t	t Temperature [C]
(x)	Adiabatic values of a fluid property x
$x_{(mp)o,nj}$	Fluid property x at mixing point, outlet, or inlet, number j

Table 3. Table of Nomenclature of processed data

Equipment 1: Boiler

The boiler is by far the most ambiguous part of this cycle, mainly because the author did not provide nearly any information of what goes on inside of it, all we know is that the mixture is brought to a boiling point, where water steam separates from the mixture, and a higher concentration boiler blowdown is rejected, the author did not provide pressure values and did not clarify the boiling and separation process.

And due to the limitation of our LiBr tables, we assume for our computation that the mixture is brought to a boiling point at 230 C and it continuously evaporates steam, increasing the concentration of the mixture and therefore increasing the boiling temperature, this continues till the steam comes to an equilibrium with the mixture at 250 C, then the steam is taken to a superheater where it is heated to 500 C before entering the turbine.

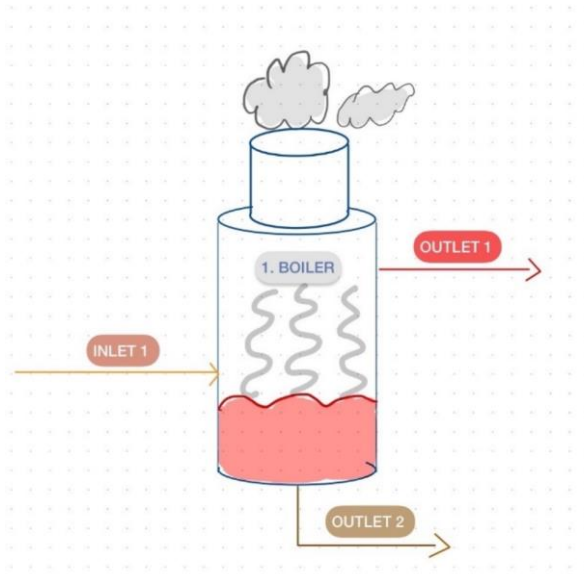


Fig 25. Equipment 1: Boiler diagram

Equipment 1 fluid properties

	Inlet 1	Outlet 1	Outlet 2
h [kJ/kg]	577.79	3471.4	569.93
t [C ⁰]	229.2	500	238.5
p [kPa]	1702	1700	1701
s [kJ/kg K]	1.5966	7.5117	1.5255
c _p [kJ/kg K]	2.811		2.6746
w [wt%]	40	*Pure steam*	45
\dot{m}	9 \dot{m}	\dot{m}	8 \dot{m}

Source/Sink	Eq8	Eq2	Eq6
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Table 4. Table of Values of Equipment 1,

(Note that h_{1o1} is the enthalpy after entering superheater at 238.5 C and exiting at 500 C)

Since the cycle objectively operates around the mass flow rate to the turbine, we will normalize all other flows as a multiple of it. For the boiler we write two equations, for pure water (with index w) and for the solution. A detailed solution is presented here for the first equipment, the rest only the necessary equations are written.

$$m_{i_{o1}}^i = \dot{m}$$

$$w_{1n1w} m_{i_{n1}}^i = w_{1o1w} m_{i_{o1}}^i + w_{1o2w} m_{i_{o2}}^i \quad (3.1)$$

$$w_w = 1 - w \quad (3.2)$$

Mass balance we get, $m_{i_{o2}}^i = m_{i_{n1}}^i - m_{i_{o1}}^i \quad (3.3)$

$$w_{1n1} m_{i_{n1}}^i = w_{1o2} m_{i_{o2}}^i \quad (3.4)$$

$$w_{1n1} m_{i_{n1}}^i = w_{1o2} (m_{i_{n1}}^i - m_{i_{o1}}^i) \quad (3.5)$$

$$\alpha_{1n} = \frac{m_{i_{n1}}^i}{m_{i_{o1}}^i} = \frac{w_{1o2}}{w_{1o2} - w_{1n1}} = 9 \Rightarrow m_{i_{n1}}^i = 9m_{i_{o1}}^i = 9\dot{m} \quad (3.6)$$

$$m_{i_{o2}}^i = m_{i_{n1}}^i - m_{i_{o1}}^i = 9m_{i_{o1}}^i - m_{i_{o1}}^i = 8m_{i_{o1}}^i = 8\dot{m} \quad (3.7)$$

$$\alpha_{1o} = \frac{m_{i_{o2}}^i}{m_{i_{o1}}^i} = 8 \quad (3.8)$$

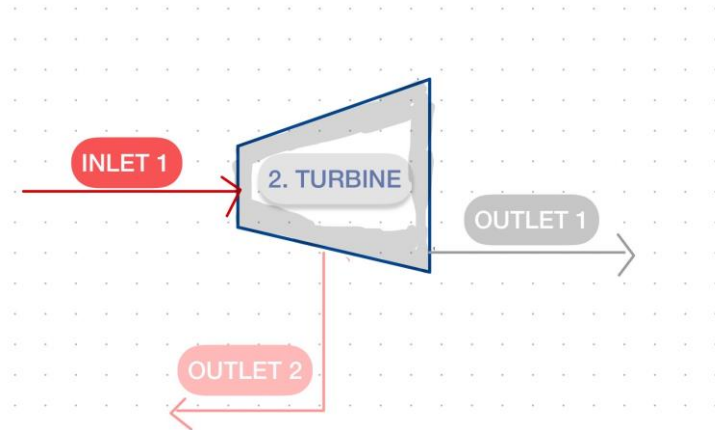


Fig 26. Equipment 2: Turbine diagram

Equipment 2 fluid properties

	Inlet 1	Outlet 1	Outlet 2
h [kJ/kg]	3471.4	2469.8 (2358.5)	2828.0 (2756.5)
t [C°]	500	42.59	120
p [kPa]	1700	8.4680	112.79
s [kJ/kg K]	7.5117	7.5117	7.5117
w [wt%]	*Pure steam*	*Pure steam*	*Pure steam*
Source/Sink	Eq1	Eq3	Eq7

Table 5. Table of Values of Equipment 2

h_{2o1} , h_{2o2} are calculated from their adiabatic values and the turbine efficiency respectively as,

$$h_{2o1} = h_{2n1} - \eta_{turbine}(h_{2n1} - h_{2o1Adiabatic}) \quad (3.9)$$

Similarly, for h_{2o2}

$$h_{2o2} = h_{2n1} - \eta_{turbine}(h_{2n1} - h_{2o2Adiabatic}) \quad (3.10)$$

Where $\eta_{turbine} = 0.90$

Defining enthalpy extraction fraction of expansion μ ,

$$m_{2n1} = \dot{m} \quad (3.11)$$

$$\dot{m} = m_{2o1} + m_{2o2} \quad (3.12)$$

$$\dot{m}h_{2n1} = (\dot{m} - m_{2o2})h_{2o1} + m_{2o2}h_{2o2} \quad (3.13)$$

$$\mu = \frac{h_{2o2} - h_{2o1}}{h_{2n1} - h_{2o1}} = 0.3577 = \frac{\dot{m}}{m_{2o2}} \quad (3.14)$$

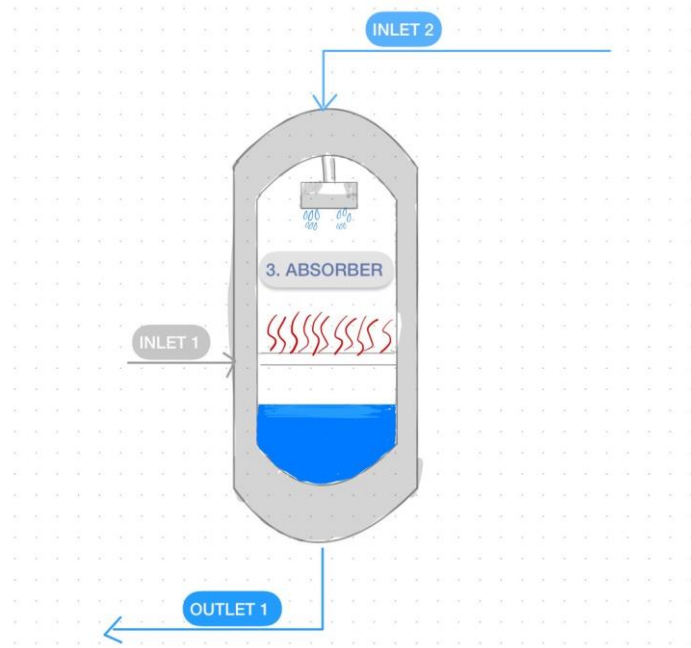


Fig 27. Equipment 3: Absorber diagram

Equipment 3 fluid properties

	Inlet 1	Inlet 2	Outlet 1
h [kJ/kg]	2469.8 (2358.5)	70.8449	123.2170
t [C°]	42.59	35	55
p [kPa]	8.4680	8.4680	8.4680
w [wt%]	*Pure steam*	40.83	40.1
\dot{m}	\dot{m}	44.81 \dot{m}	45.81 \dot{m}
Source/Sink	Eq1	Eq5	Eq7

Table 6. Table of Values of Equipment 3

$$\dot{m}_{3n1} = \dot{m} \quad (3.15)$$

$$\dot{m}_{3n1} + \dot{m}_{3n2} = \dot{m}_{3o1} \quad (3.16)$$

$$\dot{m}_{3n1}h_{2n1} + \dot{m}_{3n2}h_{3n2} = (\dot{m}_{3n1} + \dot{m}_{3n2}) h_{3o1} \quad (3.17)$$

$$\alpha_{3n} = \frac{\dot{m}_{3n2}}{\dot{m}_{3n1}} = \frac{h_{3o1} - h_{3n1}}{h_{3n2} - h_{3o1}} = 44.81 \quad (3.18)$$

$$\Rightarrow \dot{m}_{3n2} = 44.81\dot{m} \quad (3.19)$$

$$\dot{m}_{3o1} = (44.81 + 1)\dot{m} = 45.81\dot{m} \quad (3.20)$$

$$\alpha_{3o} = \frac{\dot{m}_{3o1}}{\dot{m}_{3n1}} = 45.81 \quad (3.21)$$

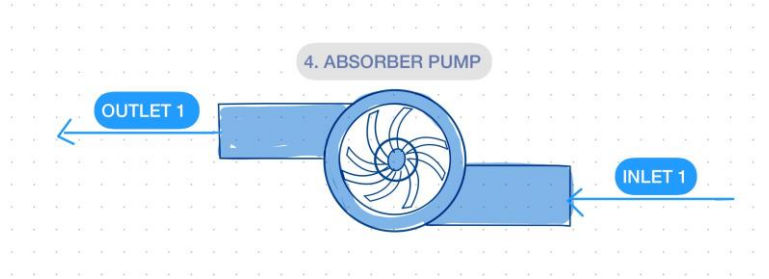


Fig 28. Equipment 4: Absorber pump diagram

Equipment 4 fluid properties

	Inlet 1	Outlet 1
h [kJ/kg]	123.2170	123.3182(123.23)
t [C°]	55	55
p [kPa]	8.4680	112.79
w [wt%]	40.1	40.1
ρ [kg/m ⁻³]	1373.7	1373.7
\dot{m}	45.81 \dot{m}	45.81 \dot{m}
Source/Sink	Eq3	Eq6, Eq5

Table 6. Table of Values of Equipment 3

$$h_{4o1 \text{ adiabatic}} = \frac{1}{\rho_{4n1}} (p_{4o1} - p_{4n1}) + h_{4n1} \quad (3.22)$$

$$h_{4o1} = \frac{h_{4o1 \text{ adiabatic}} - h_{4n1}}{\eta_{\text{pump}}} + h_{4n1} \quad (3.23)$$

$$\frac{1}{\rho_{4n1}} = v_{4n1}, \quad \eta_{\text{pump}} = 0.75$$

Finally,

$$h_{4o1} = \frac{v_{4n1}(p_{4o1} - p_{4n1})}{\eta_{\text{pump}}} + h_{4n1} = 123.3182 \text{ kJ/kg} \quad (3.24)$$

$$m_{4n1} = m_{4o1} = m_{3o1} = 45.81\dot{m} \quad (3.25)$$

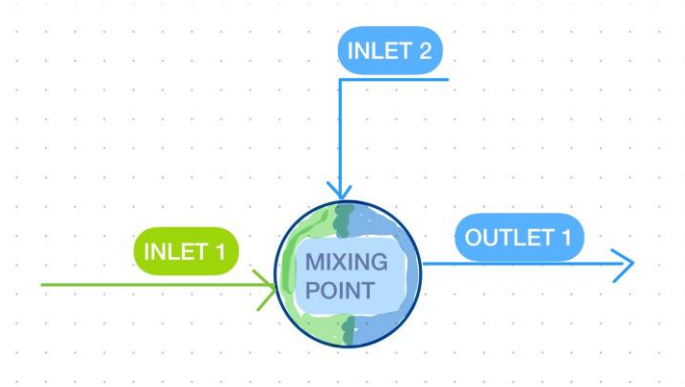


Fig 29. Mixing Point before entering air coolers

Mixing point fluid properties

	Inlet 1	Inlet 2	Outlet 1
h [kJ/kg]	401.1871	123.3182	173.3105
t [C ⁰]	171	55	72.57
p [kPa]	112.79	112.79	112.79
w [wt%]	45	40.1	40.83
m	8m	36.80m	44.81m
Source/Sink	Eq6	Eq4	Eq5

Table 7. Table of fluid properties values for mixing point

$$h_{(mp)o1} = \frac{\alpha_{1o}h_{6o2} + (\alpha_{3n} - \alpha_{1o})h_{4o1}}{\alpha_{3n}} = 173.3105 \text{ kJ/kg} \quad (3.26)$$

$$t_{(mp)n1} = t_{1o2} - \Delta t_{n2o2} = 171 \text{ C}^0 \quad (3.27)$$

$$t_{(mp)o1} = \frac{t_{3o1}(\alpha_{3n} - \alpha_{1o}) + t_{(mp)n1}\alpha_{1o}}{\alpha_{3n}} = 75.87\text{C}^0 \quad (3.28)$$

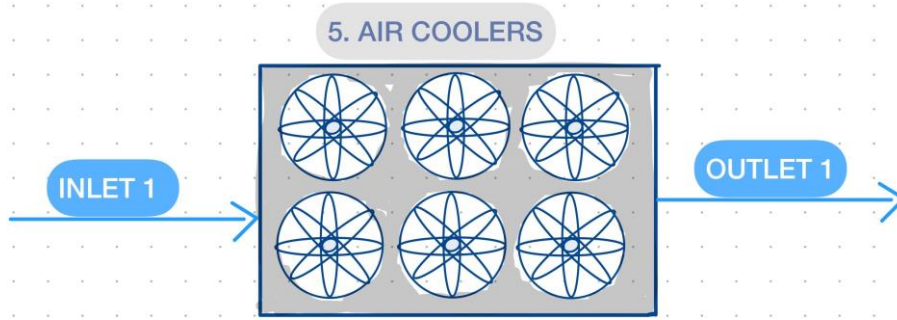


Fig 30. Equipment 5: Air Coolers

Equipment 5 fluid properties

	Inlet 1	Outlet 1
h [kJ/kg]	173.3105	70.8449
t [C ^o]	75.87	35
p [kPa]	112.79	112.79
w [wt%]	40.83	40.83
\dot{m}	44.81 \dot{m}	44.81 \dot{m}
Source/Sink	Eq4, Eq6	Eq3

Table 8. Table of Values of Equipment 5

$$t_{5n1} = t_{(mp)o1} \quad (3.29)$$

$$t_{5o1} = 35C^o \quad (\text{given parameter})$$

$$h_{5n1} = h_{(mp)o1} \quad (3.30)$$

$$h_{5o1} = h_{(mp)o1} - \frac{c_{p3n1} + c_{p7n1}}{2} (t_{5n1} - t_{5o1}) \quad (3.31)$$

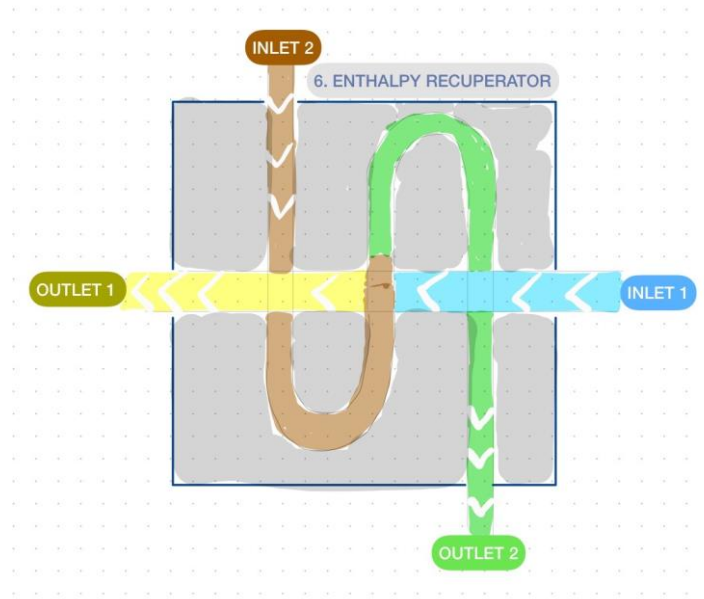


Fig 31. Equipment 6: Enthalpy Recuperator

Mixing point fluid properties

	Inlet 1	Inlet 2	Outlet 1	Outlet 2
h [kJ/kg]	123.3182	123.3182	273.3182	401.1871
t [C°]	55	238.5	120	171
p [kPa]	112.79	1700	112.79	1700
c_p [kJ/kg K]	2.4876	2.6746	2.5462	2.3254
w [wt%]	40.1	45	40.1	45
\dot{m}	$8.9516\dot{m}$	$8\dot{m}$	$8.9516\dot{m}$	$8\dot{m}$
Source/Sink	Eq3	Eq1	Eq7	Eq5

Table 9. Table of Values of Equipment 6

$$\alpha_{1o} = 8, \alpha_{1n} = 9$$

$$\Delta t_{6n1o1} = t_{7n1} - t_{4o1} \quad (3.32)$$

$$h_{6n1} = h_{4o1} \quad (3.33)$$

$$h_{6o1} = h_{6n1} + \Delta t_{6n1o1} \frac{c_{p6n1} + c_{p6o1}}{2} \quad (3.34)$$

$$\Delta h_{6n1o1} = \Delta h_{6n2o2} = h_{6o1} - h_{6n1} \quad (3.35)$$

$$h_{6n2} = h_{1o2} \quad (3.36)$$

$$h_{6o2} = \frac{\alpha_{1o} h_{6n2} - \alpha_{1n} \Delta h_{6n1o1}}{\alpha_{1o}} \quad (3.37)$$

$$\Delta t_{n2o2} = \frac{\alpha_{1n} \Delta h_{6n2o2}}{\frac{c_{p6n2} + c_{p6o2}}{2} \alpha_{1o}} \quad (3.38)$$

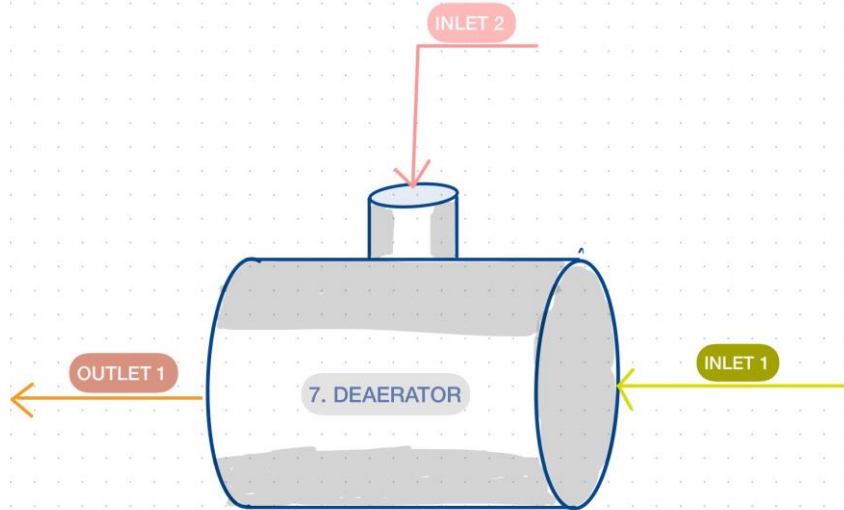


Fig 32. Equipment 7: Deaerator

Equipment 7 fluid properties

	Inlet 1	Inlet 2	Outlet 1
h [kJ/kg]	273.3182	2828.0 (2756.5)	287.0689
t [C°]	120	120	120
p [kPa]	112.79	112.79	112.79
c _p [kJ/kg K]	2.5462		2.5462
w [wt%]	40.1	*Pure steam*	40
m	8.9516m	0.0484m	9m
Source/Sink	Eq6	Eq2	Eq8

Table 10. Table of Values of Equipment 7

$$\alpha_{1n} = 9$$

$$m_{7n1} + m_{7n2} = m_{7o1} = 9\dot{m} \quad (3.39)$$

$$(m_{7o1} - m_{7n2})h_{7n1} + m_{7n2}h_{7n2} = m_{7o1}h_{7o1} \quad (3.40)$$

$$\alpha_{7n} = \frac{\alpha_{1n}(h_{7o1} - h_{7n1})}{h_{7n2} - h_{7n1}} = \frac{m_{7n2}}{\dot{m}} = 0.0484 \quad (3.41)$$

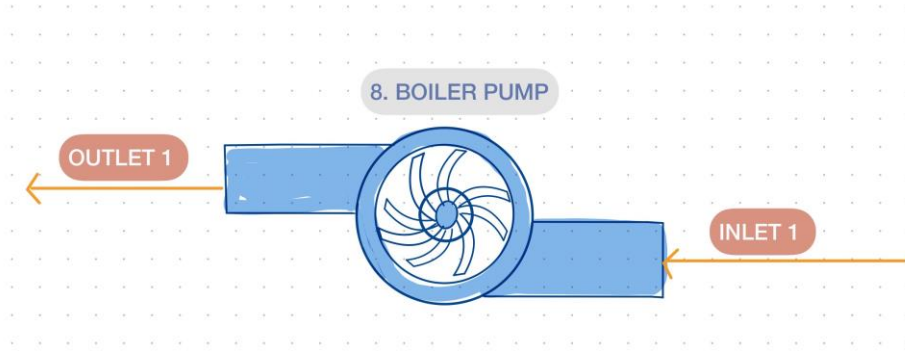


Fig 33. Equipment 8: Boiler pump diagram

Equipment 8 fluid properties

	Inlet 1	Outlet 1
h [kJ/kg]	287.0689	577.79
t [C ^o]	120	229.2
p [kPa]	112.79	1702
c_p [kJ/kg K]	2.5462	2.811
w [wt%]	40	40
ρ [kg/m ⁻³]	1326.6	1326.6
\dot{m}	$9\dot{m}$	$9\dot{m}$
Source/Sink	Eq7	Eq1

Table 11. Table of Values of Equipment 3

$$h_{8o1 \text{ adiabatic}} = \frac{1}{\rho_{8n1}} (p_{8o1} - p_{8n1}) + h_{8n1} \quad (3.42)$$

$$h_{8o1} = \frac{h_{8o1 \text{ adiabatic}} - h_{8n1}}{\eta_{pump}} + h_{8n1} \quad (3.43)$$

$$\frac{1}{\rho_{8n1}} = v_{8n1}, \quad \eta_{pump} = 0.75$$

Finally,

$$h_{8o1} = \frac{v_{8n1}(p_{8o1} - p_{8n1})}{\eta_{pump}} + h_{8n1} \quad (3.44)$$

Efficiency,

$$P = \dot{m}(h_{2n1} - h_{2o2})\eta_{turbine}\eta_{generator}\eta_{mechanical}(1 - \alpha_{7n\mu}) \quad (3.45)$$

$$P = 12500W, \eta_{turbine} = 0.9, \eta_{generator} = \eta_{mechanical} = 0.99$$

$$\dot{m} = \frac{P}{(h_{2n1} - h_{2o2})\eta_{turbine}\eta_{generator}\eta_{mechanical}(1 - \alpha_{7n\mu})} = 12.95 \text{ kg/s}$$

$$P_{absorber \ pump} = (h_{4o1} - h_{4n1}) * \dot{m} * (\alpha_{3n} + 1 - \alpha_{7n}) \quad (3.46)$$

$$P_{boiler \ pump} = (h_{8o1} - h_{8n1}) * \dot{m} * \alpha_{1n} \quad (3.47)$$

$$Q_{supplied} = ((h_{1o2} - h_{1n1})\dot{m} \alpha_{1n}) + (h_{1o1} - h_{1n1w})\dot{m} = 71244 \text{ kW} \quad (3.48)$$

Where, h_{1n1w} is pure water enthalpy at p_{1n1}, t_{7o1}

$$Q_{released} = (h_{5n1} - h_{5o1})\dot{m} \alpha_{3n} = 59271 \text{ kW} \quad (3.49)$$

$$\eta_{cycle} = \frac{Q_{supplied} - Q_{released}}{Q_{supplied}} = 16.81\% \quad (3.50)$$

4 Discussions and results

4.1 discussion of the processed data

	<i>SRC</i>	<i>ORC</i> (<i>MDM</i>)	<i>ORC</i> (<i>R245fa</i>)	<i>ORC</i> (<i>Toluene</i>)	<i>HCT</i> (<i>40% LiBr</i>)
Power Output [MW]	12.5	12.5	12.5	12.5	12.5
Turbine inlet pressure [kPa]	8500	443.5	3000	1051	1700
Turbine Outlet pressure [kPa]	4.5	1.245	249.4	7.838	8.4680
Turbine inlet temperature [C]	490	220	220	220	500
Turbine outlet temperature [C]	43	40	40	40	42.59
Cooling Water consumption [m ³ /day]	723.3	2365.8	2566.8	1470.3	*ZERO*
Efficiency [%]	39.811	16.82	15.71	24.55	16.81

Table 12. Comparing values for different computed power cycles.

It is important to note that the parameters used for the other cycles are recommended values and are far from optimized, as this would require a lot of further investigation that are suitable for further development perhaps continued in a future master thesis. Although the efficiency obtained is reasonable, we were limited by the available tables of Lithium Bromide. We have assumed the behavior of an ideal fluid when calculating fluid properties that were not given by LiBr tables (i.e., not on the saturation line), and since we were working with limited available data, some assumptions were made especially in the boiler, there is room for improvement, especially if we were able to use higher temperatures and higher concentrations which is where this cycle is supposed to operate optimally.

4.2 Advantages & disadvantages of HCT over SRC & ORC

As we have seen in the experimental data, and in our calculations, the Hygroscopic cycle offers the advantage of eliminating the need for water towers, thus saving 80% of a traditional powerplant water consumption, it is not only suitable for areas with lack of water, but also in areas with hot climate, due to the boiling point elevation as we have seen, and it promises higher efficiency than a Rankine cycle based powerplant or an Organic Rankine Cycle powerplant. Although in our calculations it is higher than some Organic Rankine cycles but not all, and it is not higher than a Steam Rankine cycle, but our model is far from being optimized.

Another advantage of the Hygroscopic Technology is that it has nontoxic waste, unlike some of the Organic compounds used in the Organic Rankine cycle, with the current direction of emphasis on environmental footprint, this comes as an attractive option.

The Hygroscopic Technology is allegedly also compatible with current steam powerplant which operates based on the SRC, this means that the cost to transition to a Hygroscopic powerplant is relatively inexpensive, because only a few equipment needs to be integrated.

One of the disadvantages of the HCT is the dependence of efficiency on the concentration of the salts, this might be a disadvantage directly or indirectly, directly because such levels need to be continuously monitored and adjusted, indirectly because it can make it increasingly difficult to ensure the quality of pure steam entering the turbine, otherwise we risk corrosion and shortened lifetime of our turbine and consequently our powerplant.

This leads me to my next point, which is, there are almost no information on the process that goes inside the boiler, making it difficult to know how much solution is needed to evaporate a target amount of pure steam, also dealing with such high concentrations (up to 60%) of salts might propose a challenge to the equipment in terms of maintenance and longevity of the equipment when dealing with corrosion prone environment.

Another point is that although this cycle promises such high performance over SRC and ORC, it is still for relatively small to medium powerplants 12.5MW in our case, although this is not a disadvantage per se, but rather an unknown whether such a cycle is scalable or not both economically and in terms of the MW capacity of the cycle.

5 Conclusion

The Hygroscopic cycle seems to have a lot of potential, especially when small powerplants are needed in areas with lack of water and hot climate, it is seemingly environmentally friendly, from the point of water consumption and nontoxic waste, but there are a lot of unknowns at this point of time that needs to be explored further and developed, so that we might get close or confirm the experimental data, if such a thing can be done, the Hygroscopic cycle might be a great solution in a world of increasing demands on electricity and where water is more and more scarce.

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