THE ORGANIC RANKINE CYCLE: THERMODYNAMICS, APPLICATIONS AND OPTIMIZATION

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Contents

Summary

New energy conversion technologies are required in order to insure the production of electricity without generating environmental pollution. Among them, low-grade heat sources or low power heat sources present an important potential of development.

The Organic Rankine Cycle (ORC) is a well known technology since the early 1980's. A large amount of ORC power plants have been built, mainly for geothermal, waste heat recovery and combined heat and power applications. This technology shows a number of advantages over the traditional steam Rankine cycle that make it more profitable for power plants with a limited electrical output power (typically lower than 1 MWe), despite a lower efficiency. The optimization of the ORC is quite different from that of the steam cycle, mainly because of the heat source temperature limitation, and because there is usually no constraint regarding the vapor quality at the end of the expansion.

This chapter presents an overview of the current state of the art in the ORC technology and exposes the main target applications. The modeling of such a cycle is described and issues such as fluid selection, optimization or control of the cycle are thoroughly reviewed.

1. Introduction

Over the last century, the world economical growth has accelerated dramatically. The industrial development, the increasing number of vehicles on the road and the multiplication of energy-consuming domestic equipments have caused an important growth of the energy demand.

This demand has been mostly covered by a massive consumption of fossil fuels, causing many serious environmental problems, such as global warming or atmospheric pollution.

An important number of new solutions have been proposed to generate electricity from alternative heat source, such as low-temperature or low-power heat sources. Among the proposed solutions, the Organic Rankine Cycle (ORC) system is the most widely used. This system involves the same components as in a conventional steam power plant (a boiler, a work-producing expansion device, a condenser and a pump). However, the working fluid is an organic component characterized by a lower ebullition temperature than water and allowing power generation from low heat source temperatures.

The success of the ORC technology can be partly explained by its modular feature: a similar ORC system can be used, with little modifications, in conjunction with various heat sources. This success was reinforced by the high technological maturity of most of its components, due to their extensive use in refrigeration applications. Moreover, unlike with conventional power cycles, local and small scale power generation is made possible by this technology.

Today, Organic Rankine Cycles are commercially available in the MW power range. However very few solutions are actually suitable for the kW scale.

This chapter presents an overview of the current state of the art in the ORC technology and exposes the main target applications. The modeling of such a cycle is described and issues such as fluid selection, optimization or control of the cycle are thoroughly reviewed.

2. Overview of Traditional Vapor Cycle Systems

The traditional vapor cycle system uses water as working fluid. It was first developed as a "Steam Engine", which is an open cycle: the water is compressed, vaporized, expanded and then rejected to the atmosphere.

Nowadays, closed-loop cycles are preferred, such as the Clausius-Rankine cycle, presented in Figure 1. In this cycle, the liquid water (point 6) is pressurized (6-1), then heated up to the evaporation temperature (1-2), vaporized (2-3), and superheated (3-4).

The mechanical work is produced in the turbine by expanding the high pressure vapor down to the condensing pressure. The cycle is finally closed by re-condensing the low pressure vapor (5-6).
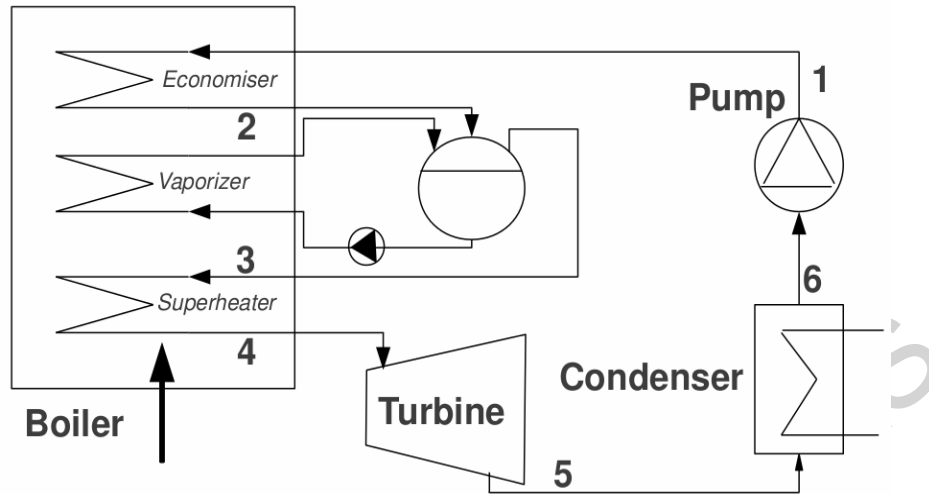

Figure 1. Working principle of a Clausius-Rankine cycle

This cycle can be presented in various thermodynamic diagrams, such *p-h, T-s* or *p-v*. In this chapter, the *T-s* diagram will be used since it is the one that represents best the irreversibilities in the heat exchangers and in the turbine (Figure 2). The different phases of the cycle can be described as follows:

• 6-1: Compression of the liquid in the pump. Points 1 and 6 almost coincide on the *T-s* diagram: if the fluid is not compressible and if the pump is isentropic, there is no increase in entropy and the temperature remains constant.

• 1-2: Liquid preheating. In the ideal cycle, this transformation is isobaric. Temperature and entropy are increased

• 2-3: Vaporization: The liquid has reached its saturation temperature and start boiling. The temperature is constant, the entropy increases.

• 3-4: Superheating: The vapor is superheated in the boiler. The temperature and the entropy increase.

• 4-5: Expansion: In the ideal cycle, the expansion is isentropic (i.e. the line 4-5 is vertical). In the real cycle irreversibilities are generated and increase the entropy.

• 5-6: Condensation: at the end of the expansion, the vapor starts condensing, until there is only liquid remaining. The temperature is constant, the entropy decreases.

In general, the compression of a liquid consumes much less energy than that of a gas. In the Rankine Cycle, the pump consumption is therefore much smaller than the electricity generated by the turbine and the net power generation is positive. The efficiency of the cycle is given by the net output power divided by the heat flow provided in the boiler:

For the basic cycle presented in Figure 1 and 2, the efficiency is typically close to 30 %
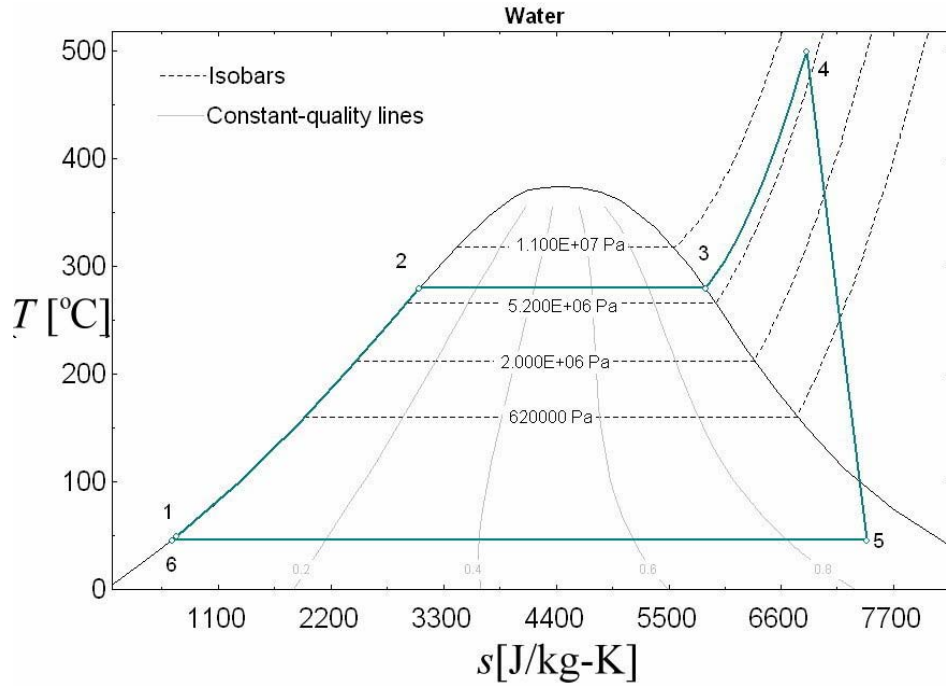
Figure 2. *T-s* diagram of the Rankine Cycle

2.1. Limitations and Optimization

The work and the efficiency of the ideal Rankine cycle can be assimilated to the work and the efficiency of an equivalent Carnot cycle working between the mean hot temperature (i.e. in the boiler) and the condensing temperature, as indicated in Figure 3 (Moran, 2004).
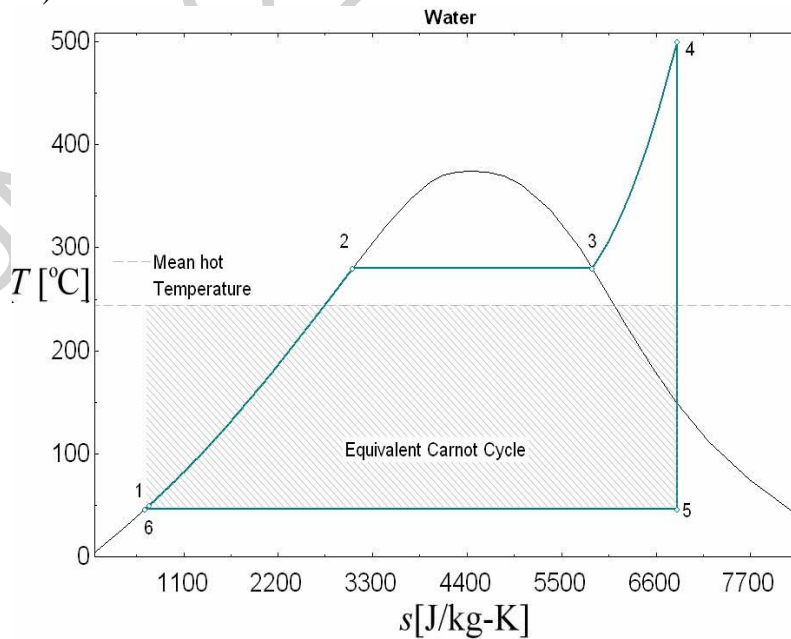

Figure 3. Equivalent Carnot cycle of a Rankine cycle

The efficiency of both cycles can be expressed by the Carnot efficiency: $\eta = 1 - \dfrac{\overline{T_c}}{\overline{T_h}}$

Where $\overline{T_h}$ is the mean heat source temperature and $\overline{T_c}$ the condensing temperature (in degrees Kelvin).

This efficiency increases when the heat source temperature increases: the evaporating curve (1-2-3-4 in Figure 3) rises, the mean hot temperature is increases so is the cycle efficiency.

In summary, increasing the efficiency of the basic ideal Rankine can be achieved in two ways: increasing the mean hot temperature, or decreasing the condensing temperature.

The condensing temperature (or pressure) is limited by the temperature of the heat sink: if a cooling tower is used, this temperature is roughly the ambient air temperature. There is therefore little space for optimization on this parameter.

Two degrees of freedom are available when trying to increase the mean hot temperature: the evaporating temperature (T_2 or T_3), and turbine inlet temperature (TIT = T_4).

For the cycle presented in Figure 3, increasing the superheating by 1K increases the efficiency by 0.041%, while increasing the evaporation temperature by 1K increases the efficiency by 0.085%. The main goal is therefore to maximize the evaporating temperature/pressure first and not the turbine inlet temperature.

However these temperatures cannot be indefinitely increased because of the following limitations:
- The TIT is limited by the turbine design: high temperatures can decrease the blade resistance and the longevity of turbine. It is typically comprised between 500 and 600°C. Current research aims at elevating this maximum temperature, among others by the use of high temperature-resistant alloys (Leyzerovich, 2008).
- The presence of liquid droplets in the steam can corrode the turbine blades and decrease its lifetime. Therefore, the vapor quality (the mass of liquid divided by the total mass of steam and liquid) is always maintained higher than 0.9.

These limitations have practical implications on the maximum evaporating temperature, as illustrated in Figure 4:
- If the maximum turbine inlet temperature is fixed (here at 500°C), and if the evaporating temperature is set to 235°C, cycle 2 is obtained (dashed lines). The cycle efficiency is 30.1% and the vapor quality at the end of the expansion is 0.95, which is acceptable (> 0.9).
- For the same maximum TIT, if the evaporating temperature is set to 345°C, cycle 1 is obtained (plain lines). The cycle efficiency is increased up to 34.4% because the Mean Hot temperature is increased. However, the vapor quality at the end of the expansion is 0.84, which is not acceptable for the turbine blades.
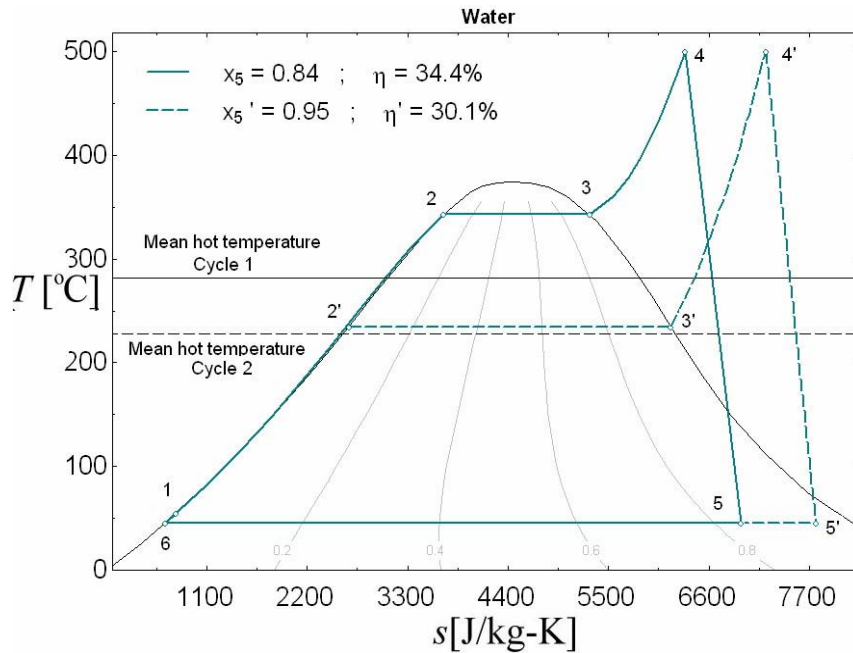
Figure 4. Effect of the practical constraints on the cycle performance

Therefore, the optimization consists in setting the TIT to its maximum value, and increase the evaporating temperature until reaching the vapor quality limit of 0.9.

2.2. Improvement of the Rankine Cycle

2.2.1. Reheating

In order to get rid of the minimum vapor quality issue at high pressure, a good solution consists in splitting the expansion in two, and introducing a reheating between both turbine stages (Figure 5). The high pressure, high temperature vapor is first expanded in the high pressure (HP) turbine (4 – 4a); The medium-pressure vapor is then returned to the boiler where it is reheated (4a – 4b); The high temperature, medium pressure vapor is finally expanded in the low pressure (LP) turbine (4b – 5).
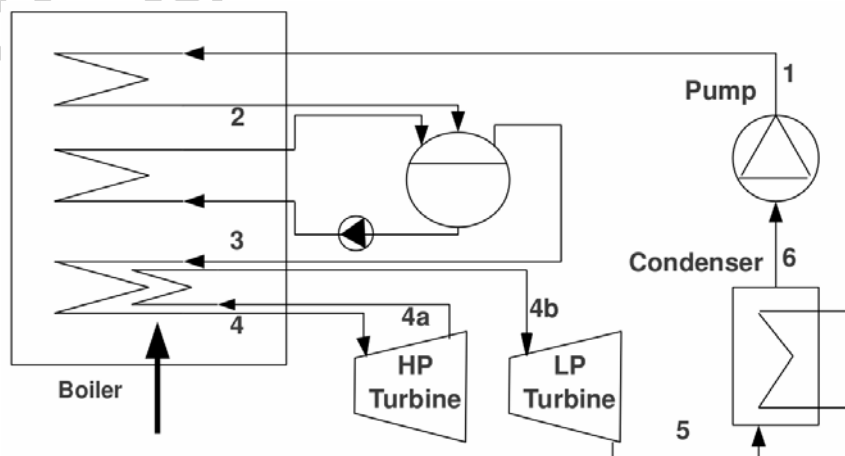

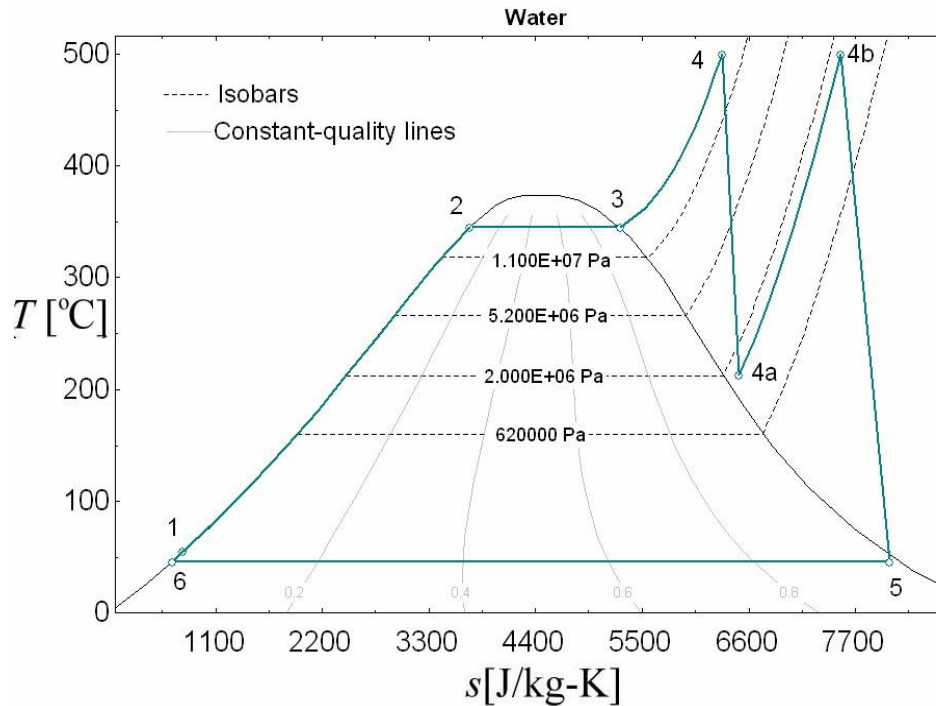Figure 5. Working principle of a Rankine cycle with reheating

Figure 6. *T-s* diagram of a Rankine cycle with reheating

Figure 6 shows on the *T-s* diagram that point 5 is displaced towards the vapor zone, decreasing in the same process its liquid droplets content. The evaporating temperature can therefore be set to a much higher value than in the basic cycle, and the efficiency is improved by a few percents.

It is possible to introduce more than one reheating, but this solution is not common because of the complexification of the system, leading to high investment costs.

The selection of the intermediate pressure (p_{4a} or p_{4b}) results of an optimization that can easily be performed numerically using a thermodynamic model of the cycle.

2.2.2. Regeneration

One difference between the ideal Rankine cycle and the Carnot cycle – and thus a source of irreversibility – is the feed water heating phase: In the Rankine cycle, the curve 1-2 is not vertical, which generates irreversibilities in the boiler. This irreversibility could be eliminated by using the heat of the steam during the expansion to preheat the liquid.

In practice, this can be achieved by extracting (or bleeding) a fraction of the steam between certain stages of the axial turbine. Figure 7 shows the working principle of the system with one bleed point, used to preheat the feed water in an open feed water heater: a fraction of the steam flow rate is bled at point 4b, at an intermediate pressure between the evaporating and the condensing pressure. The steam is directed to a direct contact heat exchanger where it is mixed with the pressurized feed water. This mixing allows heating the liquid up to its saturation temperature at the bleeding pressure. This is illustrated in the diagram of Figure 8: the mixing between states 1 and 4a heat the

feed water up to state point 1a. The saturated liquid is then pressurized to state 1b and directed towards the boiler.
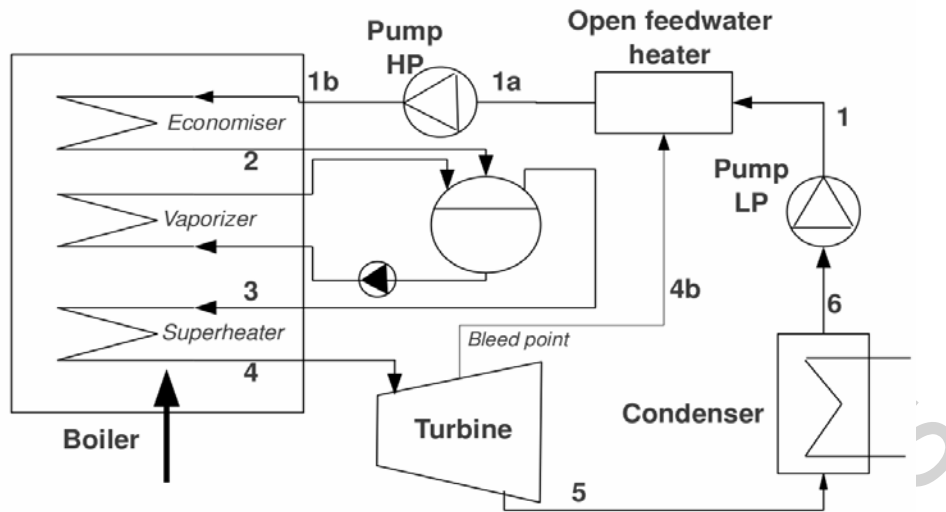
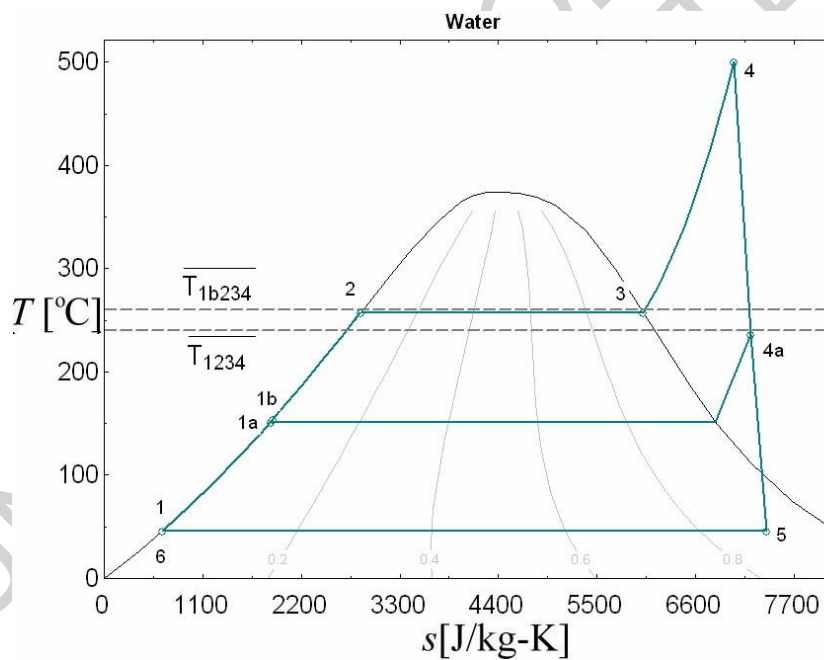Figure 7. Working principle of a Rankine cycle with one feedwater heater

Figure 8. *T-s* diagram of a Rankine cycle with one feedwater heater

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Biographical Sketches

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Sylvain Quoilin obtained a Master in Mechanical/Energy Engineering at the University of Liege in 2007. He is also the holder of a PhD in the field of small-scale Organic Rankine Cycles applied to solar and waste heat recovery applications. His research focuses more specifically on the following aspects: Fluid selection, components static modeling, optimization, dynamic modeling and regulation of the system, experimental studies. He gained a strong international experience by spending 2 months in the US (Massachusetts Institutes of Technology), 6 months in Germany (TU München), And one month in South Africa (Solar Turbine Group, NGO), visiting research groups working on the same topic. He is the author

and co-author of more than 30 journal and conference papers dealing with the ORC technology and he is the main inventor of one patent dealing with the dynamic control of waste heat recovery ORC systems.