Iğdır Üniversitesi Fen Bilimleri Enstitüsü Dergisi, 14(2), 848-864, 2024
Journal of the Institute of Science and Technology, 14(2), 848-864, 2024

ISSN: 2146-0574, eISSN: 2536-4618

Mechanical Engineering **DOI: 10.21597/jist.1442608**

Review Article

Received: 25.02.2024 **Accepted: 20.03.2024**

To Cite: Tiltay, C. (2024). Organic Rankine Cycle (ORC) Systems: A fundamental Overview of Small-scale Applications Fuelled by Low-grade Heat Sources. *Journal of the Institute of Science and Technology*, 14(2), 848-864.

Organic Rankine Cycle (ORC) Systems: A fundamental Overview of Small-scale Applications Fuelled by Low-grade Heat Sources

Celal TİLTAY

Highlights:

- Operational principles, fields of applications, and current market adaptability of the ORC technology are analysed
- Selection and design criteria for the components of small-scale ORC systems are thoroughly evaluated
- ORC performance enhancement methods and other ORC-alternative thermodynamic cycles for low-grade heat sources are highlighted

Keywords:

- ORC
- Low-grade Heat
- Small-scale ORC
- Combine Heat and Power
- Micro Generation

ABSTRACT:

Environmental issues shift energy production from conventional methods to new and more efficient alternatives. One of these alternatives is the use of organic Rankine cycles (ORC) in low-grade heat sources to generate both heat and power at small scales. Among different technologies available for this purpose, ORC-based systems seem to be the most suitable and promising option due to their simplicity and versatility. Thus, such systems have been investigated intensively. However, current studies often focus on only one aspect of these systems due to the massive research scale in this field. Therefore, this study aims to provide a fundamental and holistic overview to evaluate ORC-based low-heat sourced and small-scale applications from multiple perspectives. As a result, the basic operating principles and application areas of ORCs, selection and design criteria of their working fluids and all other system components, methods of improving their performance, and other thermodynamic cycles that can be ORC alternatives are examined in detail. The results of this study show that ORC applications can enable small-scale combined heat and power generation, while geothermal and solar energy sources have the potential to scale the size of such applications up to kW capacities. The results also showed that dry & isentropic fluids and vane & scroll expanders are the most suitable refrigerant and expander types, respectively, for small-scale ORC applications. Furthermore, the implications of all findings are critically discussed.

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INTRODUCTION

Due to the increasing level of environmental concerns and high energy consumption in all sectors, there has been a significant interest in generating energy from alternative sources, such as low-grade heat sources (30-150 ºC) (Hung et al., 2010). With this regard, one of the most crucial technologies to be employed in such applications is the Organic Rankine Cycle (ORC) (Quoilin et al., 2011), (Peris et al., 2015). In general, an ORC has the same operation principle as the conventional Rankine Cycles (RC) and consists of the same components. However, the working fluid used in an ORC is an organic refrigerant which has a lower boiling temperature than water used in RC) (Quoilin et al., 2010). This distinguishes ORCs from Rankine Cycles and enables the technology to generate energy from many different sources, including low-grade ones, and to reduce the size of energy generation to small-scales (Quoilin et al., 2011).

The adaptation into the different heat sources allows the ORC systems to be implemented into renewable energy sources. In particular, solar, geothermal, and biomass are the suitable candidates whose heat outputs are usually categorised as low-grade (Udeh et al., 2021), (Cioccolanti et al., 2017). In addition, affordable investment and maintenance, less complexity, and good market availability are the other favourable features of the ORC systems that make local and small-scale combined heat and power generation possible (Qiu, 2012). As a result, the ORC systems have been studied in a number of previous work. For example, (Quoilin et al., 2013) have conducted a techno-economic review on the feasible ORC systems and examined the viability of the technology. Some other authors focused on the more detailed technical parameters, such as refrigerant selection. With this regard, (Tchanche et al., 2009) have conducted one of the most comprehensive study on different working fluids for ORC systems and concluded that R134a refrigerant is the most suitable fluid for small-scale applications. Other authors, such as (Saleh et al., 2007), (Babatunde and Sunday, 2018), and (Bao & Zhao, 2013) have also recommended fluids that can be used in ORC systems. In addition, the utilization of the ORC systems in residential applications has gained considerable attention recently. In this concept, (Pereira et al., 2018) have investigated the ORC systems for co-generative applications and revealed the state-of-the-art current challenges in these implementations. Further, performance enhancements in the ORC have been studied extensively. In a leading study, (Shengjun et al., 2011) have studied possible ways of improving the cycle efficiency and shown that subcritical cycles yield the highest thermal and exergy efficiencies. Also, other pioneering studies can be found in (Lemort et al., 2009) for ORC's expander modelling, in (Lecompte et al., 2015) for waste heat recovery applications, in (Tchanche et al., 2009) for various other applications, and in (Tchanche et al., 2014) for different heat sources.

This paper, on the other hand, aims to present a fundamental holistic overview of ORC systems fuelled by low-grade heat sources in order to overcome the main impediments to their widespread use in small-scale applications. Therefore, design and operation principles, fields of implementation (in light of renewable energy and combined heat & power systems), and thermodynamic properties of the technology are examined, together with an in-depth analysis of the inherent challenges such as working fluid selection and expansion device issues. The novelty of the study is that it presents a holistic fundamental synopsis of all main areas of the technology, unlike the previous studies that addressed particular subjects. Hence, a fundamental overview of all aspects is summarised in this paper.

This article is organized as follows; first, the operation principles of a fundamental ORCs is presented. Then, the main field of the ORC implementation and market adaptability of the technology are provided. In addition, parameters when selecting a working fluid are evaluated. Also, details on expander selection are given. Besides, other components of a basic ORC system, including heat exchangers and pumps, are assessed. Additionally, performance enhancement methods are investigated. Further, other alternative thermodynamic cycles for low-grade heat sources are underlined. Finally, an overall conclusion and the major outcomes of the study are provided.

Operational Principles

An ORC conceptually has the same working principle and components as a conventional steam Rankine cycle, but it uses an organic refrigerant to make the cycle adaptable for various temperatures. A basic ORC consists of four main components (an evaporator, a condenser, an expander, and a pump) circulating a low-boiling refrigerant in a closed circuit. The functionality of an ORC system strongly depends on the correlations between these components, and the working fluid circulating in the cycle (He et al., 2017).

In Figure 1, a typical configuration of these components are illustrated, together with the T-s diagram (temperature (T) and (s) entropy) of an organic dry fluid (n-Pentane) as an example. As seen from Figure 1, the refrigerant is first heated and evaporated at high pressure with the input heat source in stage (1-2). Second, the produced vapour is expanded to the lower pressure, where the mechanical work is produced in stage (2-3). The refrigerant is then condensed in stage (3-4), where the available heat is extracted from the process. Finally, the working fluid is pumped to its original high pressure as a liquid in stage (4-1) and the cycle is repeated (Lecompte et al., 2015). As a result of this operation principle, an ORC unit can provide both thermal and electrical energies from condenser and expander units, respectively (Pereira et al., 2018).

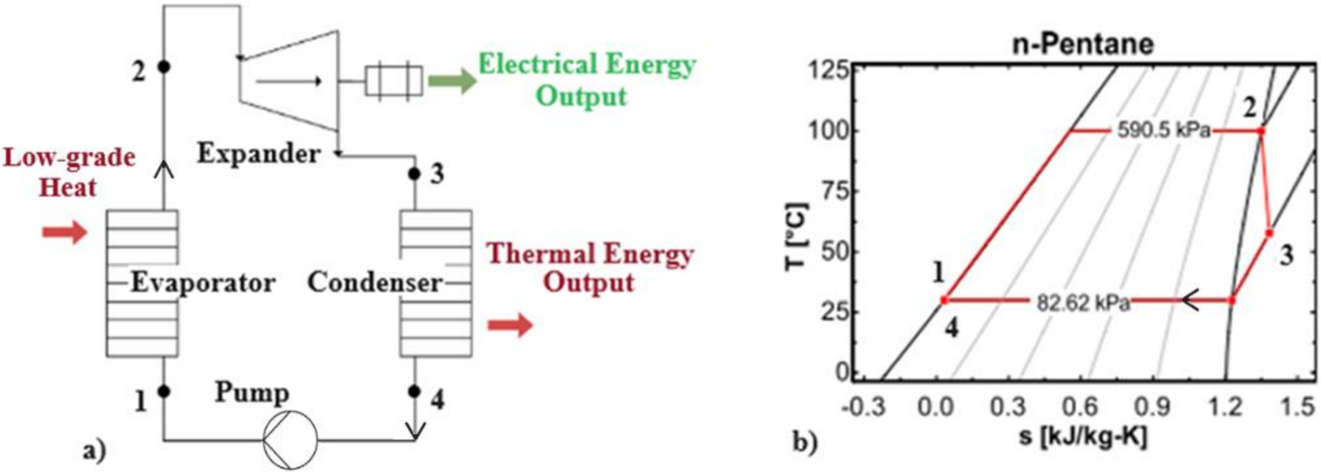

Figure 1. A typical ORC: a) Schematic representation, b) T-s diagram of n-Pentane refrigerant (Tourkov & Schaefer, 2015)

Applications and Market Adaptability

The modularity of the components, adaptability to the various temperature ranges, and non-complex operational characteristics enable the ORC to be used in various applications. These could be in power plants as a bottoming or topping cycles, or as a combined heat and power system with different heat sources such as solar, geothermal, biomass and fossil fuels, or residual heat streams (Vélez et al., 2012; Pereira et al., 2018).

Figure 2 summarises these feasible configurations of the ORC systems. As seen, the low-grade heat source of the ORC systems can be generated from renewables directly, through heat recovery processes indirectly, from the industrial processes in the form of waste heat, and other technologies such as district power cycles.

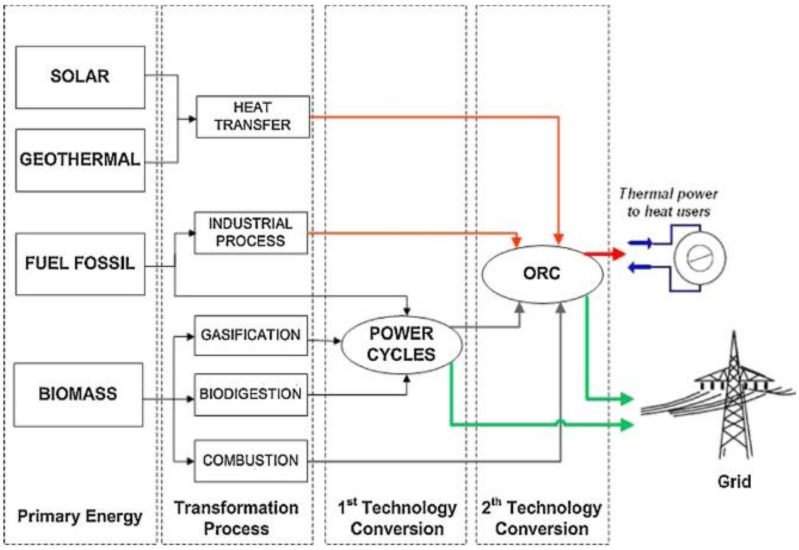

Figure 2. Possible ORC applications based on distinctive energy sources (Vélez et al., 2012)

The capacity of the ORCs can range significantly (from few kW to MW units) depending on the heat source and the purpose of the utilization. For instance, geothermal and biomass ORC applications have a heat source temperature range of 80-300 ºC and can provide energy in large capacities (up to 20 MW of electricity equivalent). In addition, solar ORC applications have a significant potential to bring down the average size of the ORC systems into kW units, making the technology smaller enough for residential applications (Tartière & Astolfi, 2017).

Working Fluid Selection

The working fluid in an ORC plays a pivotal role in system performance and economy. It has a direct impact on the sizing of the system components, designing of the expander, and the system stability. According to (Chen et al., 2010), the saturation vapour curve is one of the most essential characteristics for categorizing the different working fluids in an ORC system. For example, on a temperature-entropy diagram (T-s), fluids can be assigned to three categories; wet fluids having a negative slope, isentropic fluids having an almost vertical slope, and dry fluids having a positive slope (Qiu, 2012; Bao & Zhao, 2013). The characteristics of these categories are visualised on a typical thermodynamic T-s diagram in Figure 3.

As seen from Figure 3, wet fluids (a), such as ammonia, have a negative slope ($dT/ds<0$) in the T-s diagram, and the expansion process takes place inside the two-phase region (Vapour + Liquid). For wet fluids, this means that the outlet stream of the expander, may contain saturated liquid, which can cause severe damage to the expander (e.g. turbine blades) and reduce the thermodynamic efficiency of the turbine (Herath et al., 2020). To avoid this two-phase region, wet fluids used in ORC can be superheated before the expansion to satisfy the minimum dryness fraction of the outlet fluid, which should be around 85% (Saleh et al., 2007). However, adding another component to superheat the vapour can increase the total system cost significantly. These two disadvantages therefore limit the widespread use of wet fluids in ORC systems.

Also, due to having a nearly vertical ($dT/ds\approx0$) and a positive ($dT/ds>0$) slope in the T-s diagram, neither isentropic fluids nor dry fluids pose a threat to the expander (Saleh et al., 2007). For isentropic fluids, this is because the working fluid is expanded along a vertical line (see Figure 3-b), which means that the saturated vapour entering the expander device remains saturated throughout the expansion process and leaves the device without condensing (Chen et al., 2010). For dry fluids, similarly, there is no threat of harming the expander as the outlet stream is guaranteed to be in the superheated region (beyond the saturation curve, see Figure 3-c). Thus, isentropic and dry fluids do not require superheating and become ideal options for ORC applications. However, it should be noted that “very dry” fluids may leave the expander in the form of superheated vapour with high temperatures, which is usually regarded as ‘waste heat’ and requires an additional cooling load in the system to increase the cycle efficiency (Chen et al., 2010; Bao & Zhao, 2013), (Herath et al., 2020). Also, this additional cooling component may lead the system to be more complex and less cost-effective (Herath et al., 2020), (Saleh et al., 2007). Nevertheless, to cogenerate heat and power from the low-grade heat sources, (Lecompte et al., 2015) and (Tourkov & Schaefer, 2015) noted that dry fluids can be safely used for ORC systems in low-grade heat sources, as superheated steam can be used to generate useful heat for space heating or domestic hot water supply instead of being treated as 'waste heat'. Finally, both fluid categories (isentropic and dry) have many candidates, such as R218, R134a, HCF134a, R600a, R245fa etc. that are suitable to be employed in low-grade heat source ORC systems (Qiu, 2012), (Freeman et al., 2017). Further, other significant thermodynamic and physical properties need to be considered when selecting a working fluid for an ORC are summarised as follows:

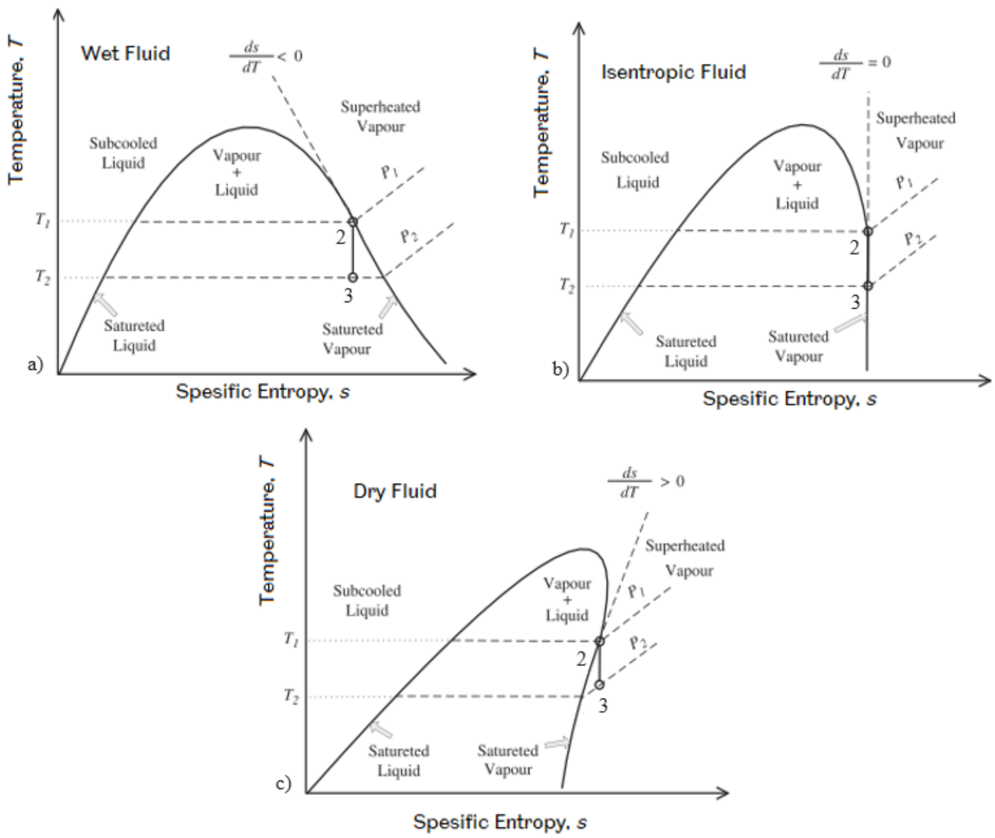

Figure 3. A typical T-s diagram for: a) wet, b) isentropic, and c) dry fluids (Bao & Zhao, 2013)

•**Critical temperature:** The critical point of a working fluid is the highest point of the saturation curve on a T-s diagram, and refers to the optimal operation temperatures of the fluid forms (liquid or vapour). Although it is acknowledged that employing a fluid with high critical temperature results in higher performances, the critical temperature of the working fluids employed in the ORC system (designed for low-temperature heat sources) should not be too high (Vetter et al., 2013). This is particularly important as the source temperature could not be sufficiently high to change the liquid form (from liquid to vapour) if the critical temperature point of the liquid is too high. Therefore, the working fluids with low critical temperatures are preferable for the ORC system fuelled by low-grade heat sources (Vetter et al., 2013), (Marion et al., 2012).

However, if the critical temperature of the fluid is excessively low, it could be difficult to condense the vapour exiting the expander. The fluids like methane, for example, whose critical temperature is below 27 °C, may not even reject heat during the condensation (Marion et al., 2012).

•**Boiling temperature:** To be easily handled at ambient environment, the expected boiling temperature for the working fluid is between 0-100 ºC (Qiu, 2012). Authors in (Guo et al., 2011) showed that the boiling temperature and critical point of the organic fluids increase simultaneously. The same authors also stated that a critical point lower than 200 ºC may lead the boiling temperature of the same fluid to be lower than 100 ºC, which is an achievable temperature that low-grade heat sources can provide.

•**Latent heat, molecular weight, density and specific heat:** According to the pioneering literature, working fluids with high latent heat, greater molecular weight, high density, and low specific heat are preferable. These properties allow the evaporator to absorb more energy from the heat source, reducing the size of the system (i.e. the pump, the evaporator and the condenser components). These characteristics also decrease the flow rate in the cycle which reduces the pump's energy consumption (Vélez et al., 2012).

•**Pressure:** According to (Moradi & Cioccolanti, 2024), fluids that require high operating pressure in a cycle make the process more efficient. However, it should be noted that such refrigerants tend to increase equipment costs and complexity of the system and affect the reliability (safety) of the system (Chen et al., 2010).

•**Enthalpy and conductivity:** A high enthalpy drop during the expansion is desirable for an ideal ORC working fluid (Qiu, 2012). This is best explained with the first law of thermodynamics where a high difference between the inlet and the outlet temperatures of the expander results in more power output for the system. Similarly, high conductivity is required for an ideal working fluid as it maximizes the heat transfer coefficient in the heat exchangers (Pourpasha et al., 2020).

•**Availability and cost:** Although the working fluid constitutes a small proportion of the entire investment cost in an ORC system, an ideal working fluid should be low-cost and abundant (Quoilin et al., 2013).

•**Safety:** A safe working fluid is often considered to be non-corrosive, non-toxic and non-flammable (Qiu, 2012; Bao & Zhao, 2013).

•**Environmental aspects**: The ozone depletion potential (ODP), global warming potential (GWP), and atmospheric lifetime (ALT) are the environmental indicators when a working fluid is considered (Bruno et al., 2008), (Chen et al., 2010). Accordingly, while R-11, R-12, R-113, R-114, and R-115 have already been phased out (Tchanche et al., 2009), some other fluids such as R- 21, R-22, R-123, R-124, R-141b and R-142b are set to be banned by 2030 (Pourpasha et al., 2020).

Expander Selection

Similar to the selection of the working fluid, there is a strong correlation between the performance of an ORC system and the type of expansion device used in the system. When an ORC system is designed, not all types of expanders are applicable for the imposed operating conditions, hence they may need further designs or modifications (Quoilin et al., 2010), (Bao & Zhao, 2013). The selection of the expander strongly depends on the size and operation conditions of the system, the selected working fluid, and the cost of the device itself (Saghlatoun et al., 2014).

The expanders that are suitable for an ORC are generically categorized into two types: velocity type and volume type (also called turbo-expanders and positive-displacement expanders) (Quoilin, 2011). However, the velocity-type expanders are less appropriate for small-scale ORC applications run by low-grade heat sources. This is because they have high rotational speeds, low-pressure ratios, and high flow rates, which make them suitable for power outputs higher than 50 kW (Qiu et al., 2011; Quoilin et al., 2013). Further, velocity-type expanders cannot tolerate the hazardous effects of the two-phase condition such as damages to the turbine blades (Saghlatoun et al., 2014). Therefore, this section only gives further details on volume-type expanders suitable for small-scale applications (below 50 kW power outputs).

The volume-type expanders can be categorised as; screw expanders, reciprocal piston expanders, scroll expanders, and vane expanders (Imran et al., 2016). The following paragraphs present further details on efficiency, working temperatures and pressures, rotational speed, lubrication requirement, rated power output, availability, cost, reliability, leakage, and frictional losses of each type of expander as follows:

Screw expanders best suit the geothermal and waste heat recovery ORC applications. They have the widest range of capacity over other types with the reported power outputs varying in between 1.5 kW and 1MW (Smith et al., 2009). The approximate operation temperature and pressure of the screw expanders are 190 ºC and 1.6 MPa, respectively (Lemort et al., 2009). In addition, the two-phase region entrance of the fluid is not a threat for these expanders due to the design of the device (Imran et al., 2016). Figure 4 shows the layout of a typical twin screw expander. The simple structure, a medium level of noise and cost, and a moderate efficiency (range from 20% to 70%) are other typical characteristics of the screw expanders (Dumont et al., 2018). However, it has to be noted that screw expanders are usually suggested for systems which have capacities higher than 10 kW due to increase levels of leakage losses in low capacities (Imran et al., 2016).

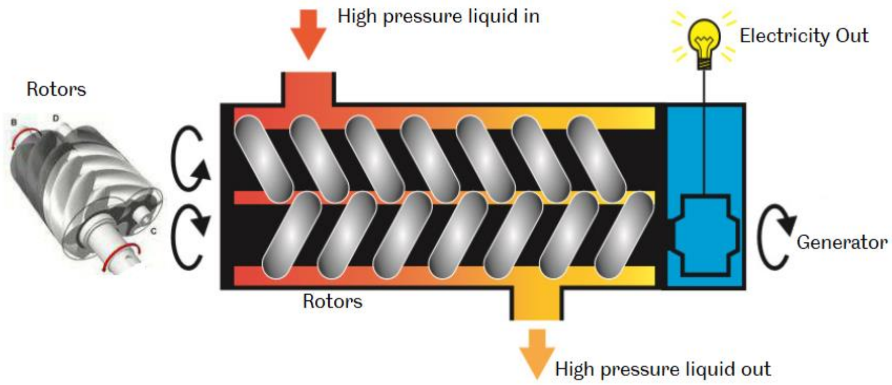

Figure 4. A typical twin screw expander (Smith et al., 2009)

Reciprocal piston expanders have mainly been used to generate heat and power simultaneously in waste-heat recovery processes from internal combustion engines (Glavatskaya et al., 2012). Although the highest recorded isentropic efficiency is 76%, their efficiency is usually lower than 50% (Saghlatoun et al., 2014). These expanders are known for their stability in high operational temperatures (380-560 ºC) and pressures (9 MPa) (Dumont et al., 2018). Also, they are not affected by two-phase zone operations, as the device does not have a turbine design that would be affected by unsaturated vapour conditions. However, reliability is one of the main issues of piston expanders as they have many moving parts that need to work together. Hence, the leakage issues can be encountered in these moving parts (Dumont et al., 2018). Furthermore, the need for frequent lubrication and high operating and maintenance costs are other major obstacles that must be overcome before this type can be widely used in small-scale applications.

Scroll expanders have the most complicated geometry (see Figure 5) compared to the other positive displacement expanders. However, they have attracted increasing interest in recent years as they are utilised to generate power outputs lower than 10 kW, making them suitable for small-scale applications such as the residential sector (Ziviani et al., 2015). This interest is also because they are cost-effective and do not require valves that increase the noise during the operation and reduce the durability of the device. Another major advantage of this type is that scroll expanders are able to start operating under any load profile without requiring a starting component or device. Further, they have a relatively better isentropic efficiency over screw & piston expanders and can operate under high temperature & pressure conditions (180 ºC and 8.2 MPa, respectively) (Zywica et al., 2016). Scroll expanders have been used with different working fluids, and tested for small scale ORC systems fuelled by low-grade heat sources in various studies in literature, (Lemort et al., 2009), (Quoilin et al., 2010). However, it should be recognised that these expanders must be properly lubricated, otherwise, the corrosion issues may occur.

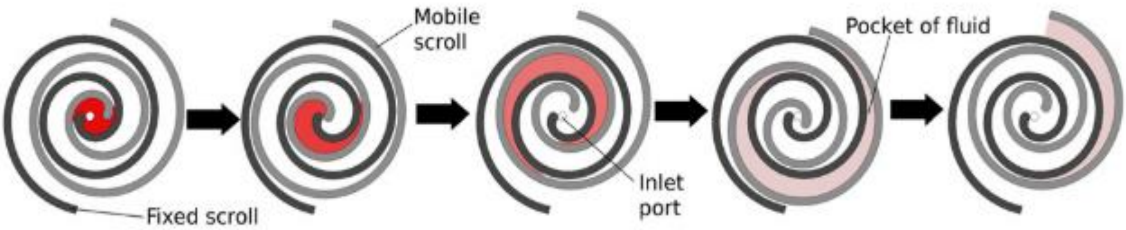

Figure 5. Working principle of a scroll expander (Quoilin et al., 2013)

Vane expanders have a high volumetric efficiency and torque output, and have a relatively simpler structure (shown in Figure 6), compared to other volumetric expanders (Kolasinski, 2019).

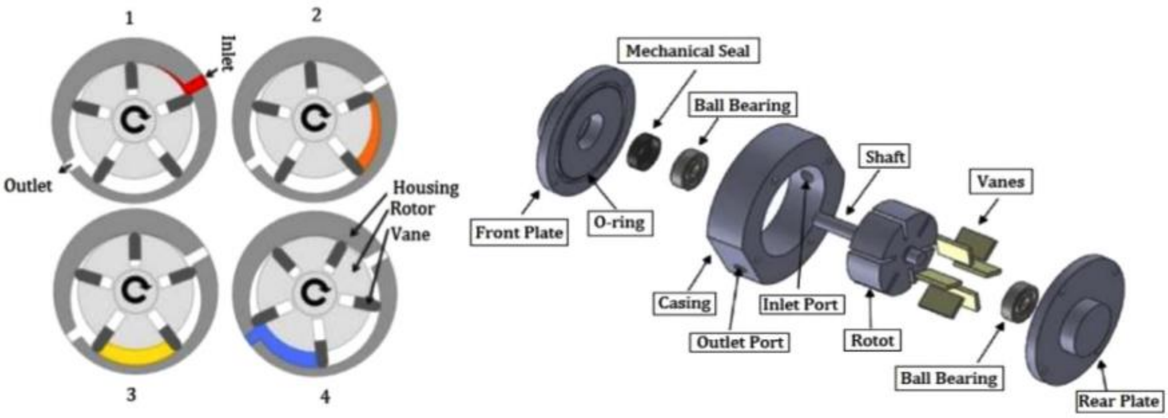

Figure 6. Working principle and components of a typical vane expander (Imran et al., 2016)

Vane expanders also have other advantages such as, high tolerance to the various vapour qualities, self-starting under different loads (similar to scroll type), and lower operation costs (Kolasiński et al., 2017). In addition, they have been tested in distinctive operational conditions under different ORCs, where minimum maintenance and lubrication requirements were observed (Imran et al., 2016). Also, they have low rotational speeds (1500–3000 rpm), hence can be directly attached to a generator without requiring a gearbox (Bao & Zhao, 2013). Low noise and vibration levels, durability in high temperatures and pressures (150 ºC and 8MPa), and cost-effectiveness are the main characteristics of this expander type (Kolasiński et al., 2017). The power output of a vane expander can reach up to 10 kW. However, they are generally applicable for systems requiring power capacities up to 2 kW, making them particularly suitable for small-scale ORC applications (Saghlatoun et al., 2014). On the other hand, lubrication is still a main requirement to be considered (Dumont et al., 2018).

Overall, scroll and vane expanders are highly likely to be reasonable choices for small-scale ORC applications. They provide an acceptable level of performance over other types, together with the ability to work under high temperature and pressure operational conditions. Simple design, affordable cost, low level of noise, and high reliability are the other characteristics that make these expanders suitable for small-scale ORC applications.

Heat Exchanger and Pump Selection

Heat exchangers can be used both as evaporators and condensers in an ORC. They are responsible for a considerable percentage of the total cycle cost. In order to correctly size and optimize a heat exchanger, two key characteristics of the heat transfer have to be considered: efficiency and pressure drop. Taking these into account, it is agreed that plate heat exchangers are preferred in small-scale systems as they provide higher efficiencies (due to the larger heat transfer surface areas), while the tube & shell exchangers are usually used in large-scale systems (Li et al., 2011).

When a heat exchanger is used as an evaporator, the available heat can be transferred from the source to the evaporator in two different ways: direct evaporation or through a heat transfer loop. In the first method, the heat is directly transferred to the working fluid from the source, hence the exchanger acts as an evaporator to the system (Quoilin et al., 2013). Although this method is more efficient and conceptually has a simpler design, it involves two important challenges. First, in order to avoid corrosion and fouling concerns, direct evaporators must withstand high temperatures, which increases the overall cost of the unit due to the use of related materials (Sharabi et al., 2008). In addition, at high temperatures, deterioration of the organic fluid may occur, especially when the temperature of its maximum chemical stability is reached (Quoilin, 2011). Second, the controllability and stability of the ORC are hardly achievable with the direct evaporators due to the phase change process of the fluid occurring during the evaporation (Vittorini et al., 2018). In the second method (for transferring the heat), an intermediate heat transfer loop is used between the heat source and the evaporator, usually using a thermal storage unit. This heat transfer loop makes the ORC operation smoother by damping the fast variations of the heat sources (e.g. to control the superheating in a solar ORC system) (Pourpasha et al., 2020). Therefore, the heat transfer loop method is usually seen as the most suitable method for the small-scale ORC installations. In addition, the same principles are applied when a exchanger is used as a condenser. However, pressure drops should be limited to not allow the exchangers to interfere with the process (Li et al., 2011).

Pumps are the final components of a typical ORC system which need to be given a particular attention. There are three significant characteristics when selecting a pump. The first one is controllability as the pumps are utilized to control the variable mass-flow rates of the working fluid in the cycle. Hence, they should be applicable to different control methods, such as differential controllers, PID, etc (Klimaszewski et al., 2020).

The second parameter is efficiency. In general, the power consumption of a pump in a Rankine cycle is quite low compared to the power output of the total cycle, hence when designing such systems, the pump's power consumption is usually neglected (Wang et al., 2020). In an ORC, however, the pump's power consumption can significantly reduce the overall efficiency of the cycle (Benato et al., 2019). The ratio between the power consumption of a pump and the expander power output is called the

Back Work Ratio (BWR) and increases significantly when the evaporation temperature increases, thereby reducing the overall cycle efficiency (Wang et al., 2020).

The tightness is the final factor that needs to be considered. As aforementioned, the organic fluids can be flammable, toxic, and have high global warming potential, hence the selected pump must be resistant to these parameters (Klimaszewski et al., 2020).

ORC Performance Enhancement

The efficiency of an ORC can be enhanced through two different methods. The first method is to integrate an internal heat exchanger (IHE) into the cycle, after the expander (shown in Figure 7). As previously mentioned, some fluids may still have a considerable amount of heat after the expansion process, which is usually considered as 'waste heat'. Thus, the IHE implementation aims to reuse this heat in the cycle and therefore increase the ORC efficiency. (Chen et al., 2010) stated in their studies that an IHE can be placed into a basic ORC system to capture this 'waste heat' for preheating the working fluid before sent to the evaporator. Preheating the working fluid essentially increases the thermal efficiency of the system. This is because with the same amount of heat source input (even if lower), a high power output from the cycle can be achieved as the refrigerant will already have been heated to some extent by the waste heat. Therefore, employing an IHE in an ORC system could be highly versatile for low-grade heat source applications (Lecompte et al., 2015). In order to implement an IHE, however, the working fluid should leave the expander without undergoing condensation. This is only possible if the dry or isentropic fluids are used (Yari, 2009). In the case of dry fluids, in particular, the temperature of the vapour exiting the expander is substantially higher than the condensation temperature, which makes the dry fluids particularly suitable for IHC-ORC systems (Tchanche et al., 2014). Figure 7 demonstrates the configuration of an IHE-ORC system and the T-s diagram of its working fluid which is the same fluid (n-Pentane) employed in Figure 1, hence the performance improvement of the system can be observed by comparing the T-s diagrams of these two figures.

In addition, there are two more modifications based on the first method (utilizing the waste heat from the expander): the open-feed organic fluid heater (OFOH) and the closed-feed organic fluid heater (CFOH). In these modifications, instead of an IHE, feed-liquid heaters are employed to preheat the working fluid. However, their designs are more complex and complicated compared to the IHE-ORC modification. Figure 8 shows the schematic diagrams of these modifications, where more details can be found in the previous studies conducted by (Lecompte et al., 2015; Tourkov & Schaefer, 2015).

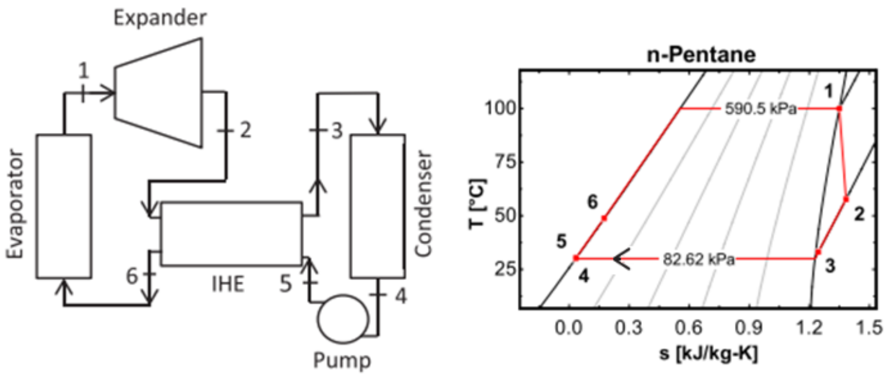

Figure 7. A typical IHE-ORC configuration and its T-s diagram (Tourkov & Schaefer, 2015)

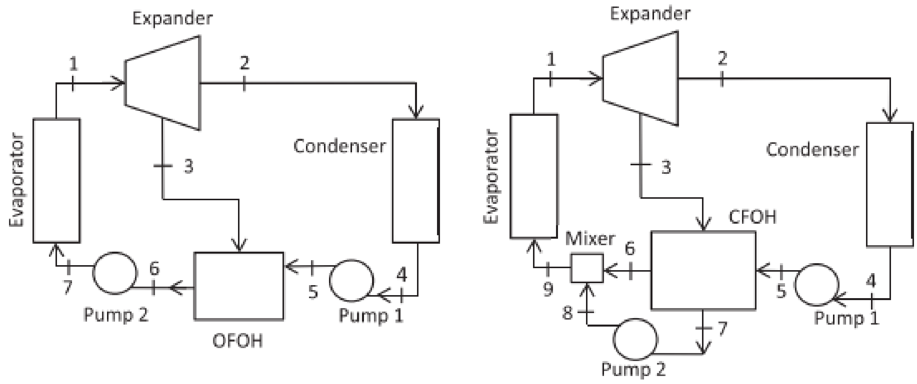

Figure 8. Typical OFOH-ORC (left) and CFOH-ORC (right) configurations (Tourkov & Schaefer, 2015)

The second method to increase the performance of a basic ORC is to superheat the vapour (before sending it to the expander) to operate the working fluid above its critical point in a T-s diagram (Moradi & Cioccolanti, 2024). In this method, however, thermo-physical features of the working fluid considerably influence the performance of the cycle. Therefore, fluids with high density and enthalpy are desirable. Among different categories (see section for the working fluid selection), wet fluids are more suitable to be superheated because superheating increases the temperature of saturated vapour, resulting in an increase in the power output of the cycle. This can be seen in Figure 9 (e.g. in wet fluids $h_1{}^{'}-h_2{}^{'}>h_1-h_2$). Likewise, it is also seen that the superheating process does not affect the performance of isentropic fluids (Hung et al., 2010) while it has a negative effect on the performance of the dry fluids.

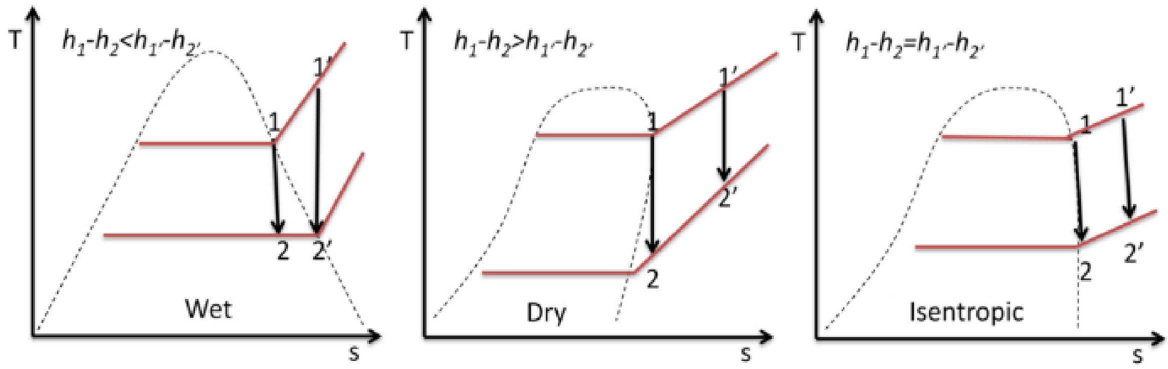

Figure 9. The effect of superheating on the fluid categories (Tchanche et al., 2014)

Other Alternative Cycles for Low-Grade Heat Conversion

Apart from the ORC, there are other thermodynamic cycles available in the literature that have been proposed and studied to convert low-grade heat sources into electricity or usable heat. The operational principles of these cycles are the same and based on a typical Rankine cycle. These alternative cycles, in a broad view, are the Kalina cycle, Goswami cycle, trilateral flash cycle, and supercritical Rankine cycle (SRC). Among these, the Kalina cycle is generally claimed to generate 15–50% more power output compared to the ORC (Shankar & Srinivas, 2016). However, (Tchanche et al., 2014) and (Chen et al., 2010) proved in their studies that the calculated performance difference between these two cycles is actually around 3% in favour of the Kalina cycle. In addition to this low actual performance difference, the Kalina cycle has a much more complex design due to the mixed working fluid process. This requires more maintenance and is less cost-effective compared to the ORCs (Pinto & Mady, 2020). As a result, for small-scale applications fuelled by low-grade heat sources, ORC outperforms the Kalina cycle.

Also, the Goswami cycle has a mixed working fluid process, which again makes the system more complex compared to the ORC (Demirkaya et al., 2018). Nevertheless, the main challenge with this cycle is the technical barriers, meaning that the power output and the cooling capacity in the cycle do not reach at their maximum level simultaneously. This leads the designers to set additional operating condition. Further details on this process can be found in (Vijayaraghavan & Goswami, 2006; Tchanche et al., 2014).

Different from the first two cycles, the trilateral cycle is based on a single working fluid process and is essentially a modified type of a basic Rankine cycle aiming to increase the efficiency of the cycle (Zamfirescu & Dincer, 2008). However, the availability of expanders, which must operate in two-phase regions and at high isentropic efficiencies, is still a major issue. According to (DiPippo, 2007), for example, an isentropic efficiency of 90% is required for these expanders to be economically viable in small-scale geothermal plants.

Among all ORC-alternative cycles, the SRC is the most favourable type to be compared with an ORC for small-scale applications. The principle of an SRC is based on compressing the working fluid (which has a relatively low critical temperature and pressure) directly to its supercritical pressure and heating this high-pressure fluid to its supercritical state before expansion process (Chen et al., 2010). Due to this low critical point and high pressurization, a better thermal performance over the ORCs is obtained from the low-grade heat sources, and this makes supercritical cycles suitable to be operated even in very low temperatures (Turchi et al., 2013).

Although there are very few different organic and inorganic fluids that have been tested and proposed as working fluids for the supercritical Rankine cycle (e.g. R123, R290, R-245fa, etc.), carbon dioxide (CO_2) is the most widely used working fluid for these cycles (Chen et al., 2006). This is because the CO_2 provides better power performance, heat transfer, and pressure drop features than the other available fluids. Also, it has a low critical point of 31 ºC, making the supercritical point easily achievable (Chen et al., 2010). Furthermore, low acquisition cost is another favourable characteristic of the CO_2 to be used in SRCs. The T-s diagram of a typical SCR cycle using the CO_2 as the working fluid is shown in Figure 10.

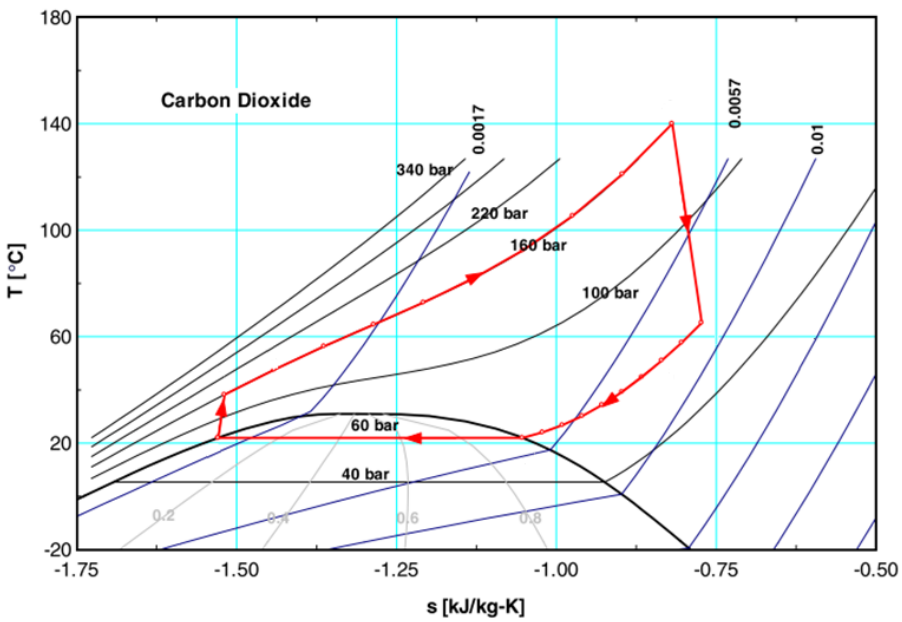

Figure 10. T-s Diagram of a typical SCR cycle using the CO_2 as the working fluid (Chen et al., 2010)

Nevertheless, employing the supercritical Rankine cycles in small-scale co-generative applications is quite challenging. This is first because the cycle operation at high pressures requires larger and more durable components, making the total system less cost-effective. In addition, high pressurization (e.g. 60–160 bar), creates safety concerns that require severe regulations (Yamaguchi et al., 2006). Moreover, the low critical temperature point (31 ºC) of the CO_2 can actually be disadvantageous for the condensation process. This is particularly quite a challenge because the working fluid (CO_2) has to be condensed, preferably around 20 ºC (see Figure 10) in the condensation process to close the cycle. However, for a system designed to provide both heat and power, this low condensing temperature may not guarantee to provide sufficient heat in many applications, such as domestic hot water or space heating.

CONCLUSION

In this study, a fundamental overview of the small-scale ORC applications fuelled by low-grade heat sources was presented. The examined aspects in this paper were the working principles of the ORCs, the main fields of their applications, selection criteria of their working fluid and expander, methods to increase their efficiency, and other alternative thermodynamic cycles that can possibly be used instead of the ORCs. The main conclusions drawn from these subjects are summarised as follows;

• ORC utilization can make the small-scale combined heat and power provision possible. This is because of the organic fluids that can be operated in various temperatures and low-grade heat sources (e.g. renewable sources).

• Solar, geothermal, biomass, and waste heat recovery are the primary fields of the current ORC applications. Geothermal or solar-integrated ORC applications, in particular, are promising technologies to scale down the size of ORCs into kW capacities. Thus, there has been an intensive interest in utilizing such combinations in residential applications to generate both thermal and electrical energy.

• Working fluid and expander selection are two main research areas for ORCs and have therefore been widely investigated in recent years.

• Dry and isentropic fluids are the most suitable refrigerant types for ORC applications due to their thermo-physical properties. However, more experimental research is needed to test these fluids in the future for widespread utilization.

• Volume-type expanders are more suitable than velocity-type expanders for ORC application that operate with low-grade heat sources. Moreover, scroll and vane expanders outperform other volumetric-type expanders in small scale applications. However, there is still a large research gap in the literature highlighting suitable expanders for ORC systems. Therefore, future studies should focus on this gap with an emphasis on leakage issues in existing expander types.

• The performance of an ORC can considerably be enhanced by implementing an internal heat exchanger to reuse the waste heat of the expander in the cycle.

• Although there are some other cycles (principally other modified Rankine cycles) suitable for low-grade heat conversion, ORC is still seen as the most favourable option for such applications due to its versatility, simple design, and low cost.

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