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Chapter

Recent Developments of Combined Heat Pump and Organic Rankine Cycle Energy Systems for Buildings

Wahiba Yaïci, Evgueniy Entchev, Pouyan Talebizadeh Sardari and Michela Longo

Abstract

To develop efficient and lower emission heating and cooling systems, this book chapter focuses on interests for the innovative combination of a heat pump (HP) and organic Rankine cycle (ORC) for building applications. In this state-of-the-art survey, the potentials and advantages of combined HP-ORC systems have been investigated and discussed. Past works have examined various combinations, comprising indirectly-combined as series and parallel, directly-combined units, as well as reversible combination configurations. Following describing such arrangements, their performance is discussed. Considerations for optimising the overall architecture of these combined energy systems are pinpointed using these same sources, taking into account heat source and sink selection, expander/compressor units, selection of working fluids, control strategies, operating temperatures, thermal energy storage and managing more variable seasonal temperatures. Furthermore, experimental works present further functional problems and matters needing additional research, and assist to emphasise experimental techniques that can be utilised in this field of research. Finally, from the studies surveyed, some areas for future research were recommended.

Keywords: heat pump (HP), organic Rankine cycle (ORC), energy systems, microgeneration, design, performance, buildings

1. Introduction

Overall, the building sector represents approximately 40% of the final utilisation of energy and 36% of greenhouse gas emissions (GHGs). For meeting international emission targets, there is a requirement for more efficient heating and cooling systems in order to decrease electricity needs while improving system efficiency as well as reliability; this is because such systems involve over 80% of residential heating usage in many nations and particularly in countries with colder climates, such as Canada [1]. In buildings, advanced heating and cooling as well as micro cogeneration technologies can possibly decrease electricity and fossil fuel-derived use via increased usage of renewable energy sources, thermal storage, micro-cogeneration as well as systems with better efficient energy

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systems. A thermodynamic-based, the organic Rankine cycle (ORC) is a remarkable system that is appropriate for recovering low-temperature heat from different heat sources, such as solar, geothermal or low-grade thermal power sources for cogenerating heat and power. Of noteworthy relevance is its combination with a heat pump (HP), with which it would be able to more efficiently supply heating and cooling, hence decreasing electricity utilisation and generation of pollutant emissions.

The thermodynamic organic Rankine cycle is characterised as a heat and power generation system, which is labelled as organic due to it utilising working fluids and this allows it to operate at lower temperatures, contrastingly to a steam Rankine cycle. This is positive as it allows the system temperatures to operate efficiently at lower temperatures, in this manner permitting small-scale applications.

Figure 1 illustrates a typical ORC cycle. Advanced ORC systems can include additional units such as a regenerator and recuperator units or extra heat exchanger/turbine, alike to the assortment of potential components for a steam Rankine cycle. In the evaporator, the heat source transfers heat to the working fluid, which evaporates and is hereafter pressurised. This working fluid is sent to the expander turbine unit, which converts the high-pressure gas to low pressure gas, converting the work produced from this process into electricity. The low-pressure gas moves in the condenser, where surplus heat that has not been converted into electricity is dissipated to the heat sink, raising the heat sink fluid's temperature and condensing the working fluid. Ultimately, a pump is employed for circulating the working fluid flow in the cycle [2, 3]. Detailed principle and research works on ORC systems can be found in [2–13].

A heat pump has an analogous working operation to that of an organic Rankine cycle, apart from it just supplying heat rather of generating of power. As its name indicates, it finally receives heat from the heat source and transfers it to the heat sink, reducing the heat source temperature and rising the heat sink temperature. A compressor and a throttling valve are employed for aiding drive and enhance the performance of this process, which can, on typical manner, operate in either heating or cooling mode, and this is where both the heat sink and source are interchanged. The efficiency of the heat pump is denoted by its coefficient of performance (COP), defined as the ratio of total heat delivered by the heat pump to the amount of electricity needed to drive the heat pump. An example of the unit's schematic is illustrated in **Figure 2**. One main benefit of this cycle is that it is able of removing heat from a heat source versus a temperature gradient with a

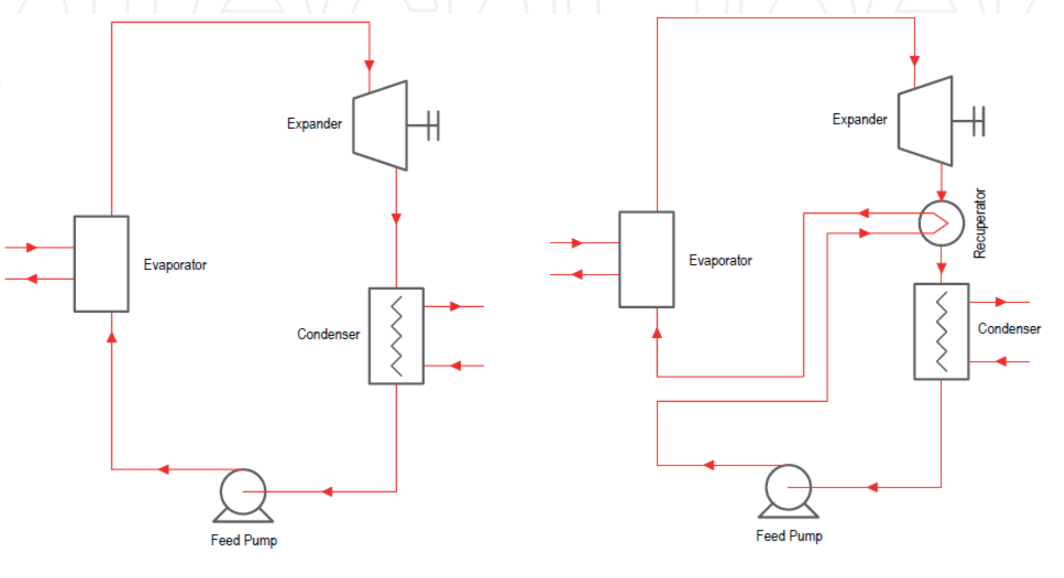

Figure 1.
Schematic of the ORC without (left) and with (right) recuperator.

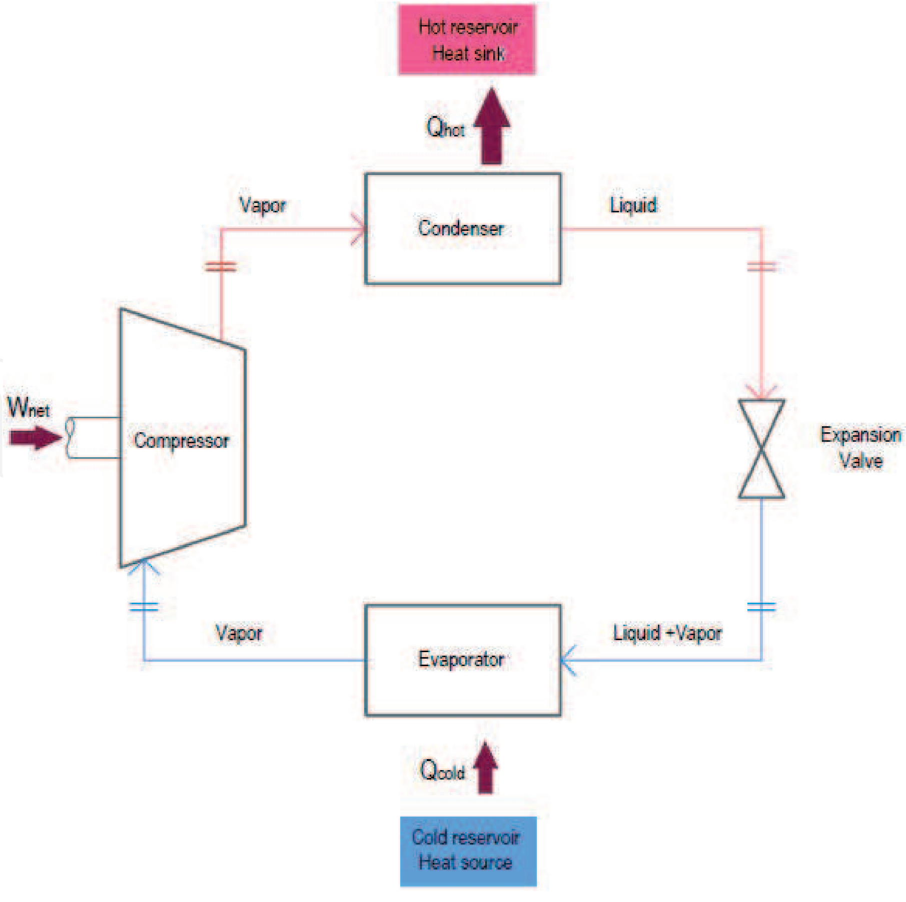

Figure 2.
Schematic of the heat pump.

higher temperature heat sink, enabling it to get to low-grade thermal sources. A heat pump has an assortment of arrangements and configurations available, which are generally out of the extent of this review. Representative heat pumps viable, incorporate a water-source heat pump (WSHP) that include a liquid–liquid heat exchange; a ground-source heat pump (GSHP), and this is fundamentally the same as a WSHP, however the heat source liquid works as a heat transfer medium for geothermal heat exchangers as well; and an air-source heat pump (ASHP) that includes an air-liquid heat exchange, with performance reliant on ambient conditions. Moreover, there are likewise absorption as well as adsorption heat pumps that have increasingly complex operation. For the motivations behind this paper, the heat pump arrangements are optional as against to the overall system arrangement, and thus, will not be examined thoroughly. More detailed works regarding HP systems can be found in [14–29].

In spite of there being various surveys addressing HPs and an ORC system's design, performance and technical-economic studies, there are a lack of reviews focusing on the combined HP and ORC systems for low-temperature and micro co/trigeneration applications. Hence, building upon and extending the work of the authors [30], the aim of the present chapter is to focus on innovative combination of a HP and ORC for use in buildings. In this overview survey, the potentials and advantages of combined HP-ORC systems will be investigated and discussed.

With these objectives in mind, the remaining part of this chapter is organised as follows: Section 2 presents the various types of combined HP-ORC systems. Section 3 elaborates the performance results from the configurations in the Section 2. Section 4 presents further thorough on specific design considerations for optimisation purposes found in important works reviewed. Section 5 summarises findings from experimental works. This later section is followed by a brief overview of some studies on the economic analysis. Finally, the main conclusions drawn in this chapter are provided in the last section.

2. Types of combined HP-ORC systems

This section will give a general overview of various types of HP-ORC systems and discuss related works that are resulted with them.

2.1 Indirectly-combined HP-ORC system

These systems are not directly combined, which will be considered later on in this section. Series HP-ORC systems have comparable connection points to those that are directly coupled, however they have an intermediary loop or device amidst them. An illustration here is a gas-engine HP-ORC system that recuperates engine exhaust gas as ORC heat source as shown in **Figure 3** [31].

Parallel systems provide additional versatility in their mode of operation and are appropriate to be better for regions that have varying temperatures during the year. A study by Li et al. [32] summarised a parallel system to deliver heat, heat storage, as well as power for continental climates, with this system displayed in **Figure 4**. This system included an ORC and a HP, both being able to generate sufficient heat for one household during the cold, or heating, season. A ground heat exchanger (GHE) was utilised as thermal storage, being the hot source that the HP extracted heat from during the heating season and the ORC recharged it during the non-heating season annually.

2.2 Directly-combined HP-ORC system

As the name implies, a directly-combined HP-ORC system is one where the same process unit is shared by the separate HP and ORC units. From the literature, which has been analysed, this takes the arrangement of either a common heat exchanger

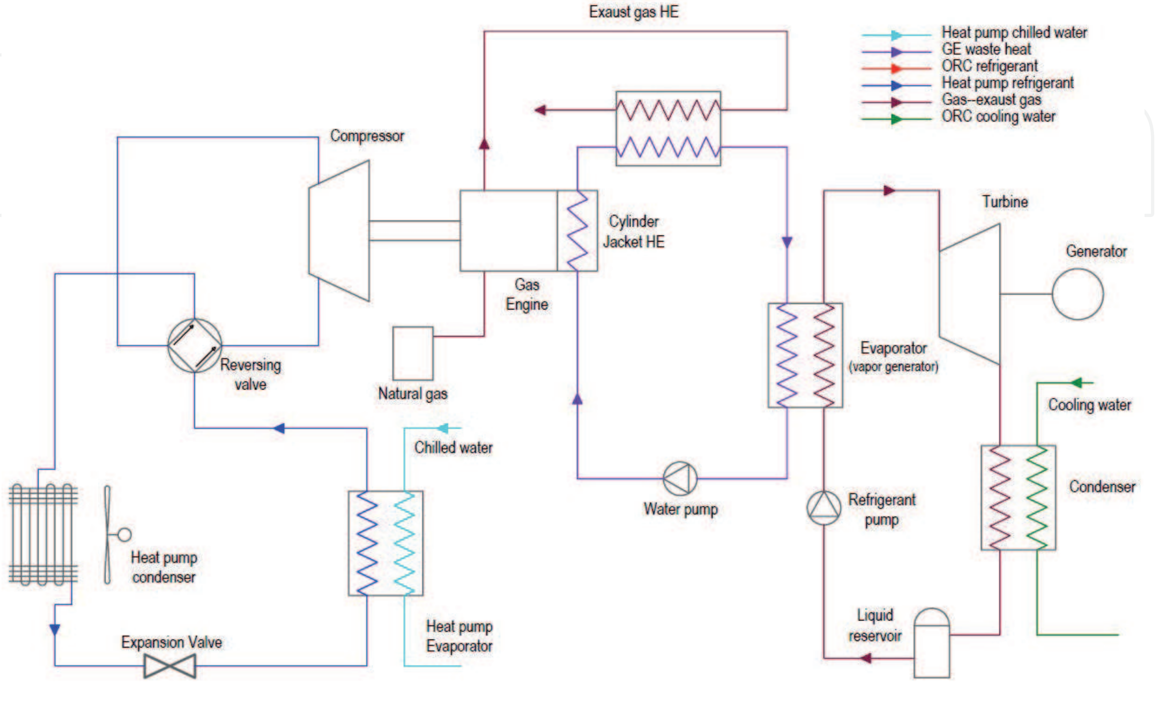

Figure 3.
Schematic of the gas engine HP with the ORC unit for exhaust waste heat recovery.

where the process streams transfer energy to each other with one being the heat source and the other functioning as the sink or serving of a coupled expander/compressor set. An example of the joint heat exchanger HP-ORC system, which was examined by Yu et al. [33] is illustrated in **Figure 5**. The HP unit shares its condenser that is also the evaporator of the ORC. This design is particular where the recovery of waste heat from another process takes place from the ORC evaporator to the HP evaporator and in return to the preheater.

The design from Roumpedakis' work [34] takes a very analogous technique, although this system is constructed in the opposed manner, with the sorption HP obtaining waste heat from the ORC and the heat source for the ORC being a solar thermal loop that underwent assessment for Amsterdam and Athens.

A comparable system to the previously mentioned one can be found in a work by Bellos and Tzivanidis [35], where the absorption HP with working fluid LiBr-H_2O, harnesses the rejected heat from the ORC that is solar parabolic trough thermal based.

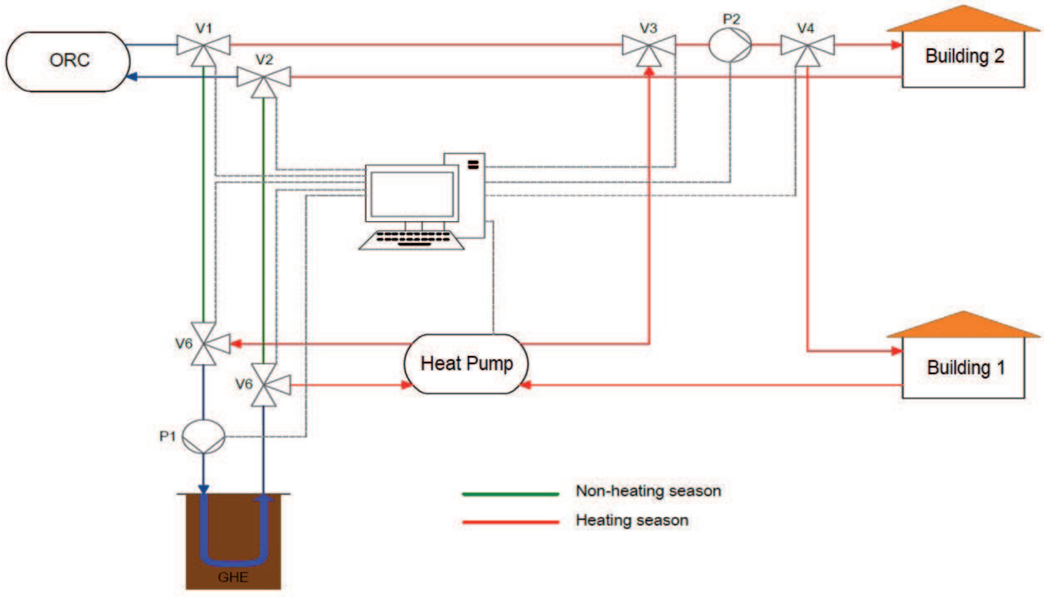

Figure 4.
Schematic of the parallel HP-ORC system with GHE thermal storage capability.

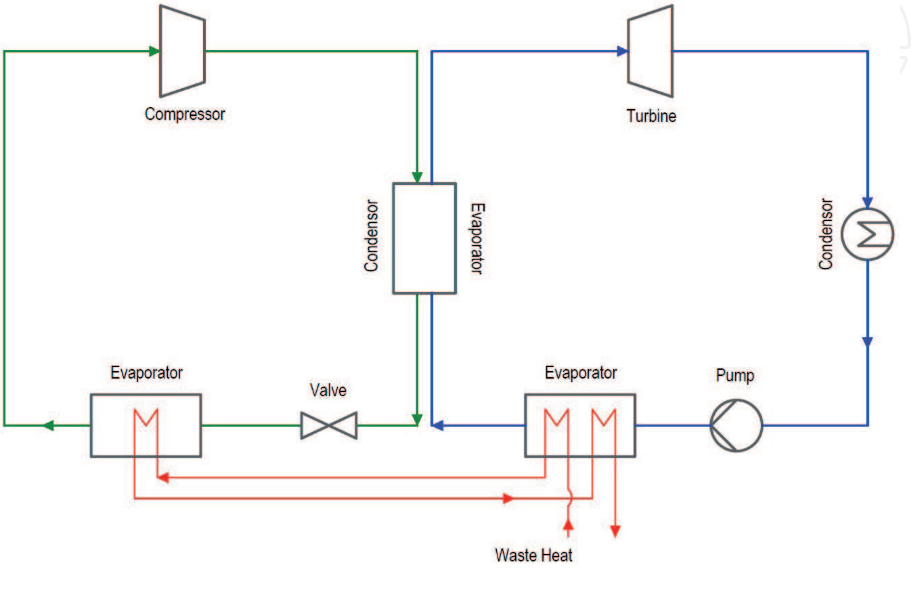

Figure 5.
Schematic of the directly-combined HP-ORC system: (left) HP; (right): ORC.

Research carried out by Mounier [36] investigated ORC-driven HPs also, with the compressor-turbine unit (CTU) being directly coupled the same as the heat exchanger unit shared between the HP and ORC, as depicted in **Figure 6**. A research paper by Collings et al. [37] assessed an identical system with a combined compressor-turbine unit and an air-source HP.

2.3 Reversible HP-ORC systems

A reversible HP-ORC unit is almost identical essentially to a parallel HP-ORC system, where there is versatility within the operations in what is currently performing, except that it integrates the ORC and HP into one unit that can operate in ORC or HP mode conditional on the need. This unit is favourable as it enables for the re-usage of components between these two modes. It additionally makes the use of thermal storage easier, wherein surplus heat can be utilised by the ORC mode to generate electricity with the residual heat stored for use by the HP to utilise for heating as well as hot water intents.

A reversible HP-ORC system has been suggested by Dumont et al. [38, 39], where the reversible unit is coupled with a solar absorber on one side, with the GHE and thermal storage tank being on the other, for ORC mode and HP mode separately. It is interesting to note that this unit moves heat one way, i.e. from the solar-based absorbers toward thermal energy storage. This system additionally gives the ability to the solar powered absorber to directly heat the thermal storage tank in the event that it can give an adequately high temperature; if not, the HP can extract more when necessary.

The system developed by Schimpf and Span [40, 41] was more basic, with the reversible unit being attached to a GHE on one side, and either the solar absorber array or storage tank on the other part as shown in **Figure 7**. In this system, the reversible unit alters path contingent upon the situation, with the ORC mode

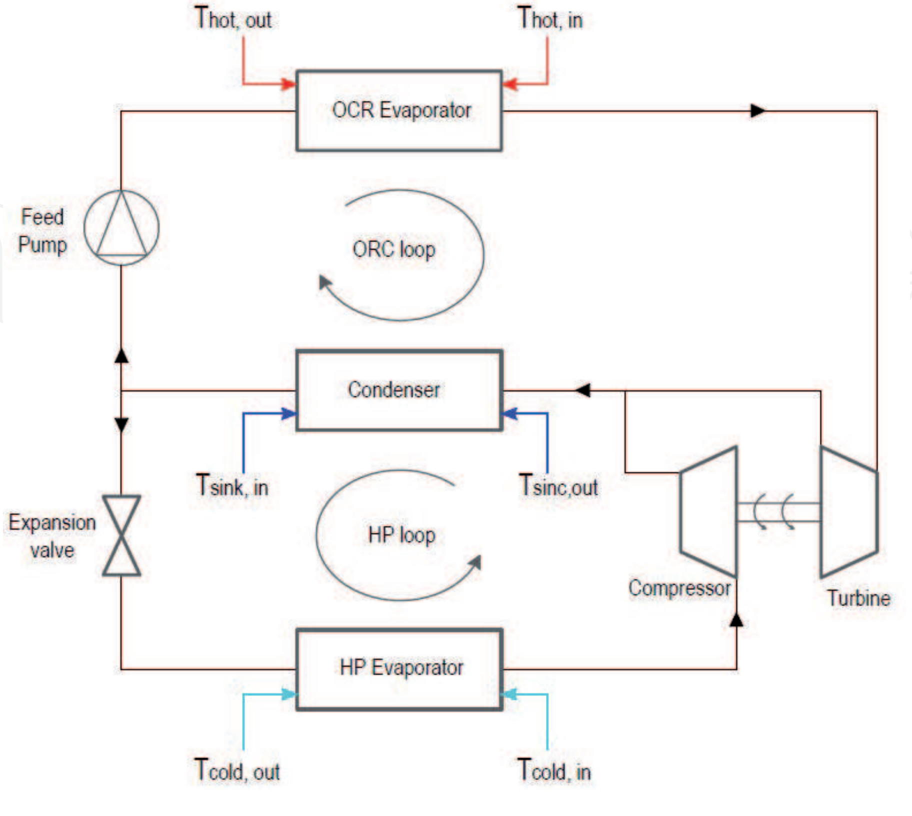

Figure 6.
Schematic of the directly-combined HP-ORC system showing the connected compressor-turbine unit.

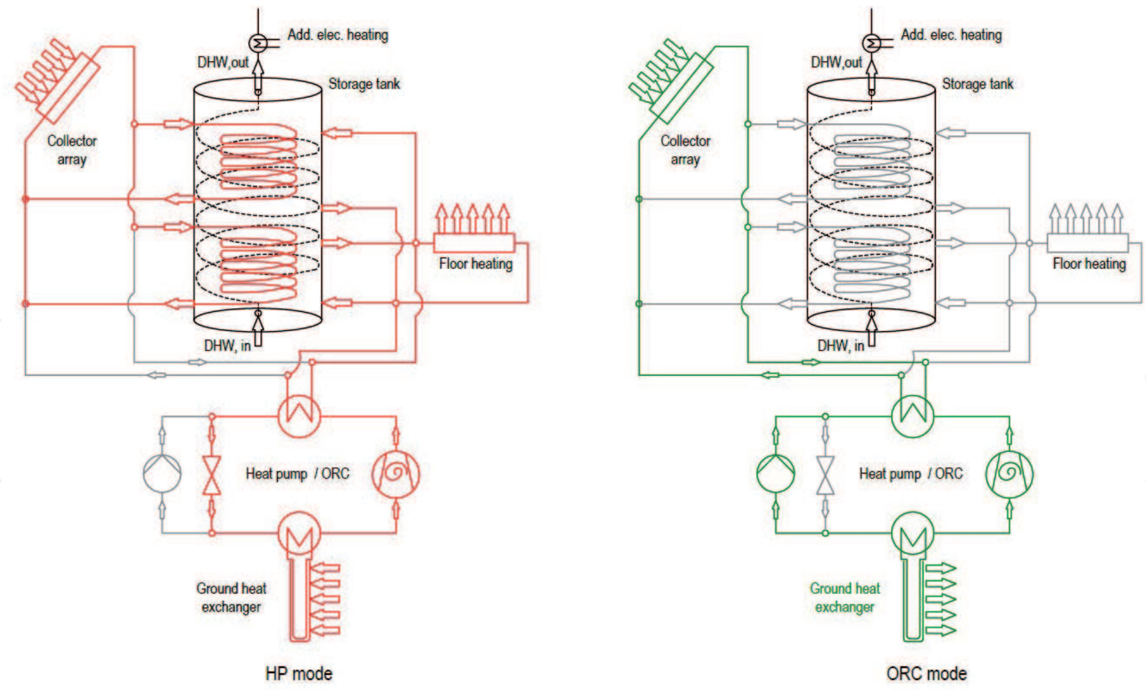

Figure 7.
Schematic of the reversible HP-ORC system: (left) the HP mode; (right) the ORC mode.

transferring heat from the solar panels to the GHE and the HP mode moving heat from the heat exchanger to the storage tank, and in this circumstance, the collector panels directly heat the storage tank too.

3. Performance

The following section on how the combined HP-ORC systems perform will go into details about configurations and results including challenges from the same literature.

3.1 Key performance metrics

As seen before, the essential duty of HP and ORC systems are to supply heating and power, with a considerable lot of these systems moreover performing capacities to supply cooling and domestic hot water. Accordingly, the main performance metrics referenced in these works are aimed at accomplishing these purposes.

There are two central methods to move toward such examination; a subsystem perspective wherein the emphasis is on the effectiveness of the HP-ORC itself, or a whole system point of view where the system's whole capacity is investigated, habitually on a yearly basis. The last is explicitly common when the heat sources and heat loads are transient. The main key performance metrics are listed in **Table 1**. Note that this list is not inclusive, rather only typical of commonly applied metrics.

3.2 Indirectly-combined series

As stated earlier on, these varieties of systems are usually designed for facilitating a greater level of heat recovery from the system, reusing waste heat via processes. In that capacity, numerous systems applicable in this class are from industrial processes and not appropriate for residential utilisation. However, one possibly applicable investigation by Liu et al. [31] recommended a gas engine-powered HP and ORC

Metric name	Typical units	Equations
Subsystem metrics		
Coefficient of performance (heat pump only)	—	For heating: $COP = \frac{Q_H}{W}$ For cooling: $COP = \frac{Q_C}{W}$ Q_H = *heating provided* Q_C = *cooling provided* W_c = *work consumed*
Thermal efficiency	%	For heat pump: $n_{th} = \frac{COP}{COP_{rev}}$, $COP_{rev} = \frac{T_H}{T_H - T_C}$ For ORC: $n_{th} = \frac{W_p}{Q_H}$ W_p = *work produced* T_C = *heat sink temperature* T_H = *heat source temperature*
Exergetic efficiency	%	$n_{ex} = \frac{n_{th}}{n_{max}}$, $n_{max} = 1 - \frac{T_C}{T_H}$
Output power	kW	Derivative of work produced by the organic Rankine cycle
Output heat	kW	Derivative of heating provided by subsystem (HP or ORC) to house or another subsystem
Output cooling	kW	Derivative of cooling provided by subsystem (HP or ORC) to house or another subsystem
Whole system metrics		
Net electricity consumption	kWh	Integral of net power generated by system *Net power* = $P_{ORC} - P_{con}$ P_{ORC} = *Power generated by ORC* P_{con} = *Power consumed (pumps, etc.)*
Total heating provided	kWh	Integral of heat power output
Total cooling provided	kWh	Integral of cooling power output
Reduction in electricity	%	Difference between electricity consumed by one system compared to a baseline, divided by the baseline's electricity usage
Reduction in emissions	%	Difference between emissions generated by one system compared to a baseline, divided by the baseline's emissions
Total waste heat recovery	%	*Waste heat recovery* = $\frac{Q_C}{Q_H}$

Table 1.
Key performance metrics of HP-ORC systems.

system utilising this waste heat recover as shown in **Figure 3**. Experiments of this system confirmed a waste heat recovery of over 55% and have possibility for residential buildings. The total cooling capacity ran between 25 and 48 kW, expanding with higher gas engine speeds. This setup found that as the water delta temperature varied in the range 11.8–24°C, the heat pump COP expanded, however the COP likewise reduced with higher gas engine speeds, running from an estimation of 6.5–10.

3.3 Indirectly-combined parallel

There are a number of potential designs within the parallel type of HP-ORC systems. Most of these arrangements have the HP and ORC units totally apart from each other, which do provide advantages, however, will not be explored here as they can be treated as separate, unintegrated systems. One exclusive parallel design is where the HP and ORC share the GHE, as represented in **Figure 4** [32]. This ORC, utilising R123 as a working fluid and a flowrate of 0.15 kg/s, generated around 2.1–2.2 kW of electricity, rejecting around 19.4–20 kW for heating. The heat pump supplied 24.9–28.7 kW of heating, spending approximately 6.9 kW.

This seems to present further advantages compared to the ground-source HP on its own, through a reduction in power consumption per unit heating area of $2.2 * 10^3$–$3.3 * 10^2$ kWh/m^2, with the ORC unit supplying 55.6% of the total heating capacity, and balancing for 78.5% of the power utilisation of the HP. This more energetically efficient system was capable to better swap a standard GSHP system, decreasing the operation time of the GSHP whilst preserving sustainable ground temperatures that operate in colder climates.

3.4 Directly-combined series

For the directly-combined series systems, one study observed that for a system of this type to be beneficial, some prerequisites are required [33].

a. the evaporation temperature of the ORC is set correctly;

b. the working fluid of the ORC has a slight ratio of latent to sensible heat;

c. poor thermal match between working fluid and waste heat for standalone ORC; and

d. the COP of the HP is adequate.

Applying these settings and the optimisation of heat exchanger temperatures, their model, with an ORC evaporation temperature of 120°C, an ORC condensation temperature to the HP of 45°C and a HP condensation temperature of 130°C, stated an expansion in net power yield and level of waste heat recuperated by 9.37 and 12.04% individually, when utilising n-Hexane as the HP refrigerant and R600a as the ORC working fluid [33]. This system had a power production of 400–800 kW when considering the working liquid, with waste heat recovery between 7000 and 9000 kW.

In a same system utilising solar-based parabolic trough thermal as a heat source for an ORC unit associated with an absorption HP, the greatest power generation attained was 152.1 kW, with the most extreme cooling generation of 465.2 kW. This was accomplished with a working pair of LiBr-H2O, water/steam as the refrigerant, the heat pump absorber and condensers working at 50°C to provide usable heat in a sensible temperature level for space heating or DHW (domestic hot water) outputs, and the HP evaporator working at 10°C to feed the cooling load. The simulation outcomes recommend that the optimal arrangement has an ORC working fluid of toluene, giving heat source temperatures near the extent of 300°C, a greatest exergetic efficiency of 24.66%, pressure ratio of 0.7605 and a heat rejection temperature of 113.7°C [35].

3.5 Reversible HP-ORC

A few investigations have called attention to focal points to exploiting this system, explicitly because it enables for efficient operation in both cold and hot climate conditions using the reversible unit and thermal storage, just as diminishing initial capital expenses. One investigation found that contrasted with the HP solar thermal system employed as the reference, reversing the HP into an ORC unit had the option to diminish the net power request of the system by 2–10% [40]. In this configuration, the ORC mode, with a working fluid of R134a, produced a daily total of between 43.3 and 145.6 kWh, contingent upon the area for solar thermal and the collector type. This production and resulting decrease in net power demand was assisted with the heat pump power between 4.58 and 6.49 kW, 0.2 m^3 of hot temperature water demanded at 45°C and a tank volume of 0.9 m^3.

Another model made established that in winter, the HP spent 17.28 kWh every day with a daily mean COP of 4.1 and throughout the summer, the ORC mode had a 5.5% effectiveness, attaining a peak power of 3.28 kW and producing 23.9 kWh the day it was attained [41]. The main difficulty of this recommended system is choosing the optimal control strategy and decreasing the overall thermal system loss as there are a couple of loss situations that ought to be reduced properly. For example, if the solar loop fluid temperature is adequately high, it may not be viable transmitting it for heating in the mid-year/summer since the house and storage are both at temperature; here the energy will rather be lost. As will be discussed in a later section, trial results founded on these designs are encouraging, showing results fundamentally the same as those simulated.

4. Design optimisation

This section will focus on elements for optimising the overall architecture of the combined energy systems, by considering HP and ORC components design, selection of working fluids, control strategies, and operating temperatures, and managing more variable seasonal temperatures.

4.1 HP and ORC components

From the viewpoint of single system and component optimisation, there are many studies concentrating on HPs or ORCs individually. Because of this, the extent of this section will be abridged where doable to considering studies including components in the framework of integrating HP and ORC into an overall combined heating and power system and their design concerns.

4.1.1 Heat source and sink selections

The differences in temperature between each evaporator and condenser pair is a significant factor in the effectiveness of individual cycle, because it closely influences the notional maximum generation and COP. Unfavourably, because of the residential or commercial uses of these systems, the accessible heat sources are habitually lower temperature, which implies the conceivable temperature differential and resulting effectiveness will be smaller. There are different strategies to enhance or optimise the temperature differential, which will be examined beneath.

For a HP-ORC system combined at the HP condenser/ORC evaporator, a work process to upgrade the working states of the combined system was made [33]. This involves adjusting the combined heat exchanger temperature and computing the other temperatures appropriately, diminishing the combined heat exchanger temperature until it either supplies an acceptable measure of waste heat recovery and expands the net power, or it is anything, but a productive combination. This operating procedure proposes that a diminished coupled heat exchanger temperature will expand the quantity of waste heat recuperated yet may not definitely influence the net power yield, which relies to a great extent upon the optimal or accessible heat source temperature.

For the directly combined compressor-turbine unit design, one investigation established that at a hot source temperature of 120°C, the maximum COP attainable by the comparing HP-ORC framework is 1.66 [36]. For a hot source temperature of 180°C, the maximum COP attainable by this HP-ORC system is above 1.8, with exergetic efficiencies in surplus of half. This analogous analysis concluded that overall, the conventional sorption systems, for example, the single effect absorption HP, function well at low heat source temperatures under 120°C, whereas these rotor-coupled HP-ORC systems function greater at temperatures above 150°C.

While working fluids will be examined in a subsequent section, it should be pointed out that a working fluid that is chosen for use in the system should correspond to the mix of source and sink temperatures for better thermodynamic maximisation.

The configuration of the system will aid prescribe the temperature differentials. Specifically, appropriate choice of a heat source is significant for defining the conceivable temperature from it, and will probably guide the configuration of the whole system.

Combustion heat sources, for example, natural gas and diesel, can normally give greater temperatures yet at a greater emissions generation. These emissions can be mostly decreased by the utilisation of biomass as alternative fuel. Moreover, the dissipated heat from these heat sources can be partly recuperated for additional heat transfer uses in the system as displayed in **Figure 3** [31]. Thus, to the exhaust heat from conventional fuels, the exhaust heat from a SOFC (solid-state fuel cell) can be recovered in a HP-ORC system as the heat source as was investigated in [42] and shown in **Figure 8**, giving a 3–25% increase on exergy efficiency contrasted with the SOFC power cycle, as it stood alone.

One possibility to additionally decrease emissions in comparison with combustion heat sources is via further renewable sources. Aside from electrically heated heat sources, which would have the option to consume power from the grid or nearby sustainable sources, both geothermal and solar thermal alternatives are frequently utilised to give a heat source, taking into consideration a progressively environmental activity, generally speaking. There are a broad assortment of designs

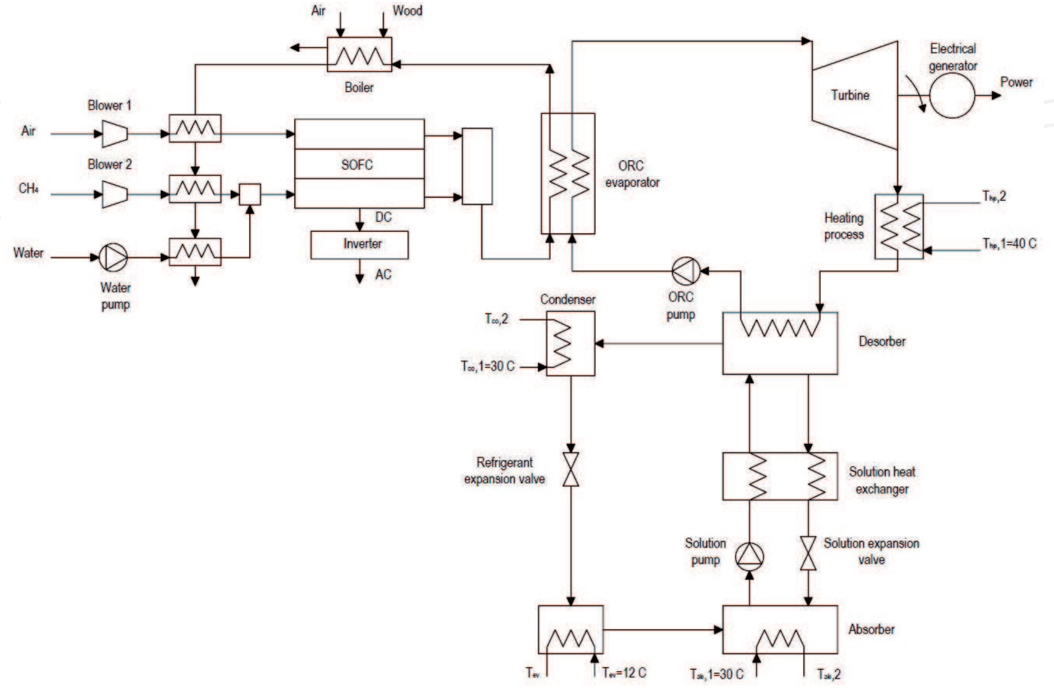

Figure 8.
Schematic flow diagram of the ORC-driven absorption HP system driven by waste heat from a SOFC.

for both of these sources that will not be explored in the extent of this chapter, yet they eventually both go about as heat exchangers for a working fluid experiencing their piping components. Aside from the regional climate conditions, the design of the solar thermal collectors for the most part prescribe the workable heat transfer attainable, with further complex as well as costly designs allowing higher temperatures. Because of their sporadic use, solar thermal heat sources frequently involve an intermediate storage between the ORC-HP circuit and the solar thermal collectors, and may additionally require optional heating capacities if the collectors are not adequate for dependable function.

A number of systems exploit certain type of thermal storage, both for keeping up a required heat source or sink temperature and for saving surplus heat. One basic design is a liquid storage tank, which, if the liquid is water, has the additional capacity of giving residential hot water also. An investigation operating a reversible HP-ORC unit found that for sizing this water storage tank, with estimated temperature extends between 100 and 150°C, the required ORC power yield and release period had a definite correlation to the size of the storage tank, with the temperature differential, examined above, showing an opposite relationship to it [43].

Small-scale geothermal units have, for the most part, lower temperatures by design and as such needs a HP to separate the accessible heat. Even though it is accessible dependably, over the lifespan of the system, the quality of this source will deteriorate when more heat is removed. So to alleviate this, it is conceivable to utilise a geothermal heat exchanger as a type of thermal storage in situations when the heat at a given instant is not essential for the consumer, recovering the geothermal heat source.

This technique was applied in the parallel framework displayed in **Figure 4**. It was discovered that the recharging of the GHE by the ORC had advantages for the complete system when contrasted with just the ground-source HP system, keeping up a greater yearly average COP of 3.8 instead of 3.7 to 3.2 in 20 years, and an 85% decrease in total power utilisation [32].

4.1.2 Expander/compressor units

One exclusive arrangement of this component can be found when the compressor of a HP and expander of the ORC are directly combined across their rotor. This provides the capacity of directly exploiting the mechanical energy from the ORC to power the compression in the HP, in spite of the fact that it represents some mechanical difficulties and involves the two units to be working reliably and dependably for suitable application. The most crucial piece of this system is the turbomachinery, which, in one investigation, when it was enhanced, provided efficiencies in surplus of 60%, a 20 point effectiveness increase compared to their proof of concept, featuring the requirement to limit fluid leak and turbomachinery tip clearances during the fabrication [36]. This work proposed a maximum HP COP amount of 1.8, and for the overall system, had 40 kW heating capacity.

Another research utilising this expander/compressor unit found that a fuel-to-usable heat efficiency of 136–160% was attained, with the HP COP extending between 2.8 and 5.5 [37]. This investigation employed the condensers of the two units to heat up water for household hot water production with the ASHP rising the water temperature to 25–30°C, the ORC condenser expanding it up to 55°C and the exhaust gas from the natural gas combustion for the ORC evaporator expanding the water temperature to 60°C for use in DHW application. The natural gas combustion rises the ORC working fluid evaporator inlet temperature to 200°C, bringing about an ORC efficiency of 20%. Below 5°C of ambient air temperature, 3.9 kW of energy was generated from the ORC to water, compared with 3.8 kW heating from the HP.

Another type of this unit is as a reversible compressor/expander that can work in either way in the event that HP mode or ORC mode is required. One investigation that attempted an assortment of small-scale below <5 kW, expander/compressor units observed that while the biggest isentropic effectiveness was 81% for the scroll expander, compared with 53% for the piston and screw expanders, the mechanical restrictions and working settings are essential for choosing the unit suitable for the application, and recommended an approach to precisely design these units [44].

4.2 Working fluids selection

The selection of working fluid is significant as it directly influences the thermal specificities of the system, for this reason it should be appropriately matched with the required functioning conditions. Working fluid selection is reliant to system arrangement and component sizing. One investigation testing the impact of working fluid choice on system performance concluded that working fluids that have greater decomposition temperatures have greater fuel-to-heat efficiency [45]. Frequently, there are a few working fluids that are comparable in performance of the system. Thus, some compromises must be done. For a rotor-coupled HP-ORC system, one research found that R134a and R152a were the ideal working fluids for this system, with the best selection eventually being an accommodation between the COP and capital expenses [46].

So also, the trade-off can likewise be between technical and environmental performances. Similar to the case with testing of a gas-driven HP-ORC system, where R123 generated the greatest thermal efficiency and energy efficiency of 11.84 and 54.24%, while trials of R245fa generated lower amounts of 11.42 and 52.25%, however showed lower ozone depletion potential and global warming potentials [31]. It ought to be noticed that the combination of an ejector directly into the evaporator subsystem of an ORC can possibly enhance the thermal performance of the whole system via efficient usage of working fluid phase transition [47–50]. In reference to **Figure 9**, a model of this system found that the cycle can generate 10.78% more power, and recover 19.04% more heat from the system [50].

Mixtures of working fluids can enhance the heat transfer capacities in a heat exchanger by giving a more thermodynamically efficient temperature glide, which can possibly intensify the power production and heat recuperation just as at the same time decreasing expenses and ecological effect contrasted with the more costly or higher effect fluid in the mixture.

There are various advantages to working with zeotropic mixtures of dry and wet working fluids, as exhibited by a modelling and simulation study by Zhu et al. [51], which found that inside an ORC combined with an ejector and HP, these blends brought about a higher temperature glide, power effectiveness, cooling efficiency and coefficient of performance. Worth mentioning was R141b/R134a (55:45), R123/R152a (85:15), and R141b/R152a (80:20), which had, individually, the most power yield with a greatest power efficiency of 6%, the most elevated cooling impact with a maximum cooling efficiency of 20.3%, and the maximum COP of 1.18. An evaluation of the relative net power output of working fluid mixtures compared to the best pure working fluids indicate that the potential rise from simulation, ranges from 2.56 to 13.6% relying upon the approach utilised to evaluate this enhancement [52]. It likewise proposed that if big heat exchangers are possible to utilise, the benefits of mixtures will be further obvious.

Furthermore, mixtures of working fluids can be finely adjusted in climate-reliant systems in accordance on the optimal composition for thermodynamic enhancement. A new investigation of an ORC system with composition modification capabilities, has demonstrated this capacity to upgrade the working fluid

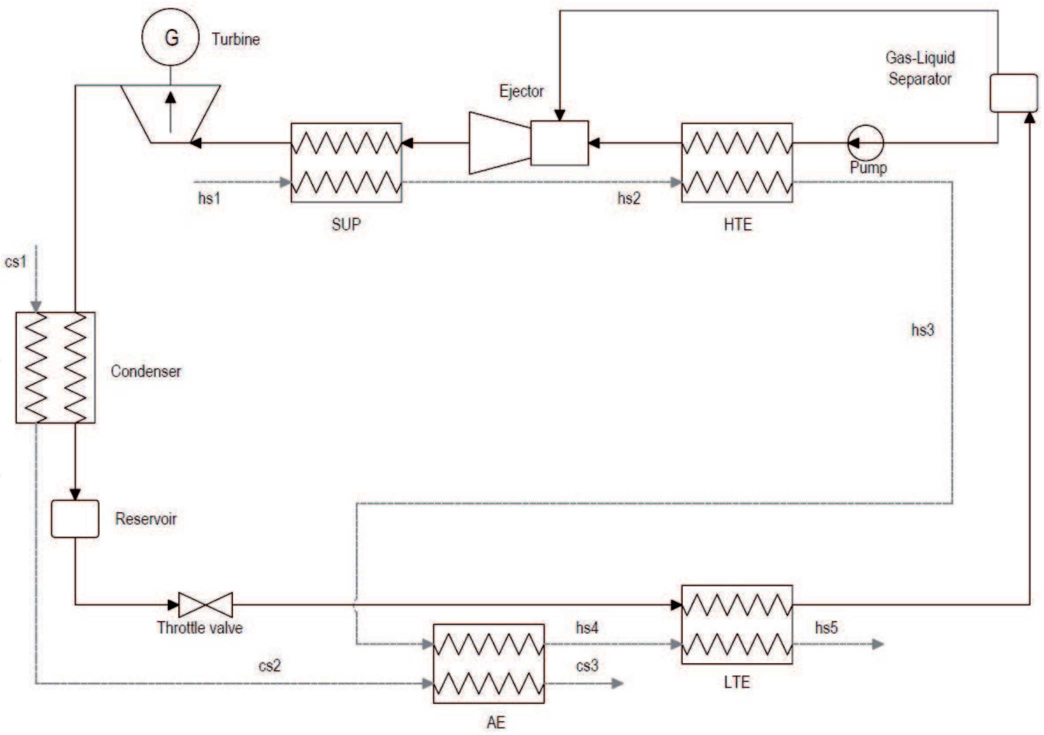

Figure 9.
Schematic diagram of the coupled ejector-ORC. AE: adaptive heat exchanger; HTE: high-temperature heat exchanger; G: generator; LTE: low-temperature heat exchanger; SUP: super heater; cs: cold source; hs: hot source.

composition to best suit surrounding conditions, providing an improvement of yearly mean thermal efficiency by up to 23% over a traditional ORC at a heat source temperature of 100°C, at a rise in capital expense of under 7%, overall proposing smaller reimbursement terms, particularly in areas with big air temperature variances amongst winter and summer periods [53].

4.3 Control strategies

A rise in convolution of a combined HP-ORC system usually requires several control strategies for appropriate arrangements of components and process flow layouts. These control strategies are commonly consisted of thermocouples to check temperature at various positions in the system, choosing the optimal arrangement of components dependent on thermal accessibility and requirement. A variant of this can be found in the system delineated by **Figure 4** where a duty cycle controls when it is supposed the heating season, when the red stream channels are operating, and non-heating season, when the green streams are effective [32]. During the heating season, a basic differential temperature controller controls if the HP is on or off.

Additionally, the reversible system displayed in **Figure 7** exploit both solar thermal and ground-source HP to expand the temperature in a thermal storage tank. When the tank attains the required temperature, any surplus solar energy is transferred to the now-reversed HP in ORC mode, where it is consumed to generate power and recharge the GHE [40, 41].

While most of systems considered apply the temperature monitoring to recommend system arrangement and system conditions, there are a couple of works that have tried further innovative control. One investigation examined the impacts strategies, for example, better space heating control when household hot water is needed, and the monitoring of occupancy to adjust set point temperatures, and at last established that they are typically beneficial in enhancing thermal and electrical efficiency and decreasing friction on the system parts [54]. Nonetheless, it was noticed that their exploitation and viability do differ dependent on the building properties and occupant schedule, and ought to be applied with attention.

4.4 Variable weather effect

Generally, if there were slight difference in the climate, both hot and cold temperature areas would not have to modify their combined HP-ORC system arrangement to satisfy seasonal conditions. Unfavourably, in areas that have much changeable climate (akin to a lot of Canada with cold winters and hot summers), seasonal prerequisites would prescribe an adjustment in the HP-ORC system arrangement. Therefore, both reversible HP-ORC systems and parallel units are best alternatives, as the adaptability in these systems take into account such modifications in arrangement, permitting indoor temperatures to be kept up during the time using thermal storage, and efficient component control.

A significant number of these suitable systems have been surveyed in earlier sections, and will just be referenced here. One investigation demonstrated an ORC with working fluid composition fine-tuning, permitting the system to be thermodynamically maximised dependent on the climate conditions [53]. A parallel system was simulated in an area of China with comparable temperature variances to Ottawa, Ontario, and incorporated a GHE thermal storage with recharging options within its design and control strategy [32]. At last, while not expressly examined in reference to big temperature differences, the investigations with reversible HP-ORC systems address the theme indirectly by their but milder areas of Belgium, Denmark and Germany [40, 41, 43, 44].

5. Experimental studies

Most of the studies done on these combined systems has been carried out by means of modelling and simulation studies, for example, the ones considered previously. Generally, this is reasonable, since several of the separable components in the system have validated models to achieve a specific understanding of the system dynamics and viability. There have been a couple of proofs of concept for a combined HP-ORC framework. While it will not be provided directly, individual ORC systems have been investigated, validating past thermodynamic models compared to experimental ORC designs, for example, ORCmKit [55]. As ORC units are generally more recent products, the assembling and configuration needed will probably bring about some intrinsic changeability in results compared with the models.

Even though the performance outcomes were before addressed in detail, the work with gas engine-driven HP and ORC heat recovery unit introduced results on a test rig of the system, developing their thermodynamic model of the ORC. For their model of the system, it was discovered that the greatest uncertainty for cooling capacity was 1.23%, gas engine energy consumption was 0.57%, waste heat was 2.11%, COP was 3.42% and primary energy ratio was 3.56%, all proposing a reasonably high conformity amongst their created models and the experimental design [31].

For the heat exchanger-combined HP-ORC unit, when contrasting their created models to proof of concept experimental data, it was revealed that their proof of concept has a 30% lesser COP and 43% greater expenses than simulated, recommending that these losses are because of not well optimised compressor-turbine units and heat exchangers and can be improved essentially with appropriate refinement [36].

While there have been works trying separate HPs or ORCs, an investigation regarding the reversible HP-ORC system analysed the experimental results from the system, finding different items for real application. This study established that a cycle efficiency of 4.2% is accomplished in ORC mode, from condensation and evaporation temperatures of 25 and 88°C individually, and a COP of 3.1 being acquired in HP mode from condensation and evaporation temperatures of 61 and 21°C, respectively, demonstrating the viability of the concept [38]. One cause of efficiency loss happened at the expander/compressor, as it was not at first geometrically intended for reversible application.

5.1 Experimental works

For model development, validation, and component optimisation intents, it is usual to disconnect or apart a subsystem to assist assessing its viability and use in the more extensive system preceding any bigger scale testing or demonstration. For these separate subsystems, a method known as the reconciliation method can be applied, which means to characterise the most likely physical condition of a system and modify every estimation as much as possible through information on its precision, duplications, restrictions, and solving mass and energy balances [56]. Through this investigation, exploiting this technique when implemented to a reversible HP-ORC unit allowed further effective data collection and validation, decreasing the error for example in the situation of the pinch-point calculation of an evaporator where its normalised root mean square deviation was diminished from 14.3 to 4.1%.

5.2 Deployment

There has not been any associated cases of combined HP-ORC systems beyond experimental facilities for building applications. From the past modelling and experimental studies, in any case, there are an assortment of concerns that ought to be examined once demonstrating these systems. One of these matters is the relative novelty of ORC units. These units, particularly at a small and micro scale, have not generally been adequately optimised in real-word use, and there is lesser data about it contrasted with well-known systems. This will in general reduce the possible advantages and rise the whole prices owing partly to maintenance necessities.

Another huge concern is the territorial changeability in climate and temperature. All together for the system to be monetarily attainable, the system must be designed explicitly to suit the consumers' requirements and application needs, which will be affected by the climate existent in the area. There must likewise be an assurance on what capacities are wanted such as cooling, heating, domestic hot water, power, which will likewise determine the arrangement of the system.

6. Economic analysis

For economic analysis, the basic equations applied are from standard engineering economics. In HP-ORC projects, the determinants of costs comprise of the underlying initial capital for buying the equipment, the net power consumption, and the related overhaul and maintenance necessities. In light of these qualities, the yearly prices and savings can be viably decided, taking into consideration an income investigation, evaluation of return and also financial viability. These outcomes enable for concrete examination between potential systems.

Numerous numbers of the studies realised concerning combined HP-ORC systems cover some financial analysis. Because of the intermittence of heating and power options, it is challenging to assess undeviating comparison amongst them accordingly. In contrast with a solar thermal system combined to a ground-source HP, one investigation assessed that the adaptation of this system to a reversible HP-ORC system for a residential system would just cost roughly $600, and following 20 years of utilisation, presents benefits of $230 in Ankara and $110 in Denver [40, 41]. Moreover, it recommends that the primary factors for the viability of the system is the area, especially continental climate conditions, in addition to components and pump expenses, including the working fluid choosing, which the pump is suitable with.

In view of the reversible HP-ORC system designed by Dumont [44], a comparison of this system with the further developed HP and PV (photovoltaic) system, indicates that as a whole, the reversible system is fewer beneficial, albeit an expansion in heat demand for heating or domestic hot water, can possibly enhance the effectiveness of the reversible system over that of the HP and PV system. This is additionally established from temperature areas with similarly lower temperature deviations. It prescribes further investigation is performed to ascertain where the reversible system is financially cost-effective against the HP and PV system.

A few models created stated reimbursement periods for the PV-ORC-chiller arrangement to be 9 years and 6.5 years in Amsterdam and Athens, individually, and for the combined solar thermal-ORC-chiller to have return periods of 16.5 years and 49.5 years in similar areas [34]. Likewise, the greatest exergy efficiency of the described models, for the PV case is 2 and 6%, while the solar thermal case is 18 and 37%. The sensitivity analysis performed with this work disclosed that if the energy cost increments by 10%, this could decrease the return period by 12.9–15.0%, besides it is the best effective factor on the financial outputs.

As shown, a portion of the bigger causes of changeability amongst implementation, are the power costs and areas climate, with the whole system design at last being advised by these factors. Improvement of this system will have some effect on the financial viability, albeit such changes ought to be assessed cautiously to guarantee some profit is achieved.

7. Conclusions

In conclusion, the combination of HP and ORC units have capacity for more energy efficiencies in various configurations. This survey provided some innovative concepts and designs to aid in prospect modelling and experimental studies alike, pinpointing various issues of more research. The main matters are:

- For improving the heat transfer, it is suggested to emphasise on maximising the temperature differential between the heat source and heat sink, similarly as suitably adjusting a working fluid for the specified ranges. Accounts of the heat sources and comprising thermal storage will be truly efficient at enhancing performance.

- Designs should be selected reliant on what is required from the system. Changing climate conditions across countries such as Canada, would need a variety of heating, cooling, and domestic hot water demands, in addition to being a concern itself for selection of heat sources, such as from air-source and ground-source HPs to solar thermal collectors. Some designs and arrangements are just applicable for particular ambient air temperatures or weather conditions, despite an improved optimal result is to have better control or resilience in the system if there is difference in seasons.

- Other optimisation approaches, for instance, advanced control strategies and individual component optimisation, will have certain influence on upgrading the system, regardless of the way this will not be as substantial as the alternatives above and should be counted economically as it might not be feasible to implement for every circumstance.

- On defining the weather conditions, the area is fundamental at choosing energy costs. Both of these two elements are maybe the greatest influence on assessing economic feasibility and hence, cannot be neglected.

For future research, it is recommended to:

- Compare a variety of advanced designs to a standard to understand which systems are optimal for a range of cold climate (such as Canadian) areas.

- Explore the combination of these systems in newly designs and configurations to realise alternative solutions for utilisation options, for instance, for isolated locations.

- Evaluate relevant combined HP-ORC systems and their readiness for bench testing, demonstration, deployment, in addition to developing models founded on them to well design the systems.

- Perform a sensitivity analysis to understand main drivers in the design and optimisation of systems of interest.

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Conflict of interest

The authors declare no conflict of interest.

Nomenclature

AE	adaptive heat exchanger
ASHP	air-source heat pump
COP	coefficient of performance
COP_{rev}	coefficient of performance of reversible HP
CTU	compressor-turbine unit
G	generator
GHE	ground heat exchanger
GHGs	greenhouse gas emissions
GSHP	ground-source heat pump
HP	heat pump

HTE	high-temperature heat exchanger
hs	hot source
LTE	low-temperature heat exchanger
ORC	organic Rankine cycle
PV	photovoltaic
Q_C	cooling provided
Q_H	heating provided
R123	2,2-dichloro-1,1,1-trifluoroethane ($C_2HCl_2F_3$)
R134	1,1,1,2-tetrafluoroethane (CF_3CH_2F)
R141b	1,1-dichloro-1-fluoroethane ($C_2H_3Cl_2F$)
R245fa	1,1,1,3,3-pentafluoropropane is a hydrofluorocarbon ($C_3H_3F_5$)
R152a	1,1-difluoroethane ($C_2H_4F_2$)
R600	n-Butane (C_4H_{10})
SOFC	solid-state fuel cell
SUP	super heater
T_C	heat sink temperature
T_H	heat source temperature
WSHP	water-source heat pump
W_C	work consumed
W_p	work produced

Greek letters

n_{ex}	exergetic efficiency
n_{th}	thermal efficiency

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