energies

MDPI

Review

# Expander Technologies for Automotive Engine Organic Rankine Cycle Applications

**Fuhaid Alshammari [1], Apostolos Karvountzis-Kontakiotis [1], Apostolos Pesyridis [1,2,*] and Muhammad Usman [1]**

1 Centre of Advanced Powertrain and Fuels (CAPF), Department of Mechanical, Aerospace and Civil Engineering, Brunel University London, Middlesex UB8 3PH, UK; Fuhaid.Alshammari@brunel.ac.uk (F.A.); A.Karvountzis@brunel.ac.uk (A.K.-K.); usman7@live.com (M.U.)
2 Metapulsion Engineering Ltd., Northwood, Middlesex HA6, UK
* Correspondence: a.pesyridis@brunel.ac.uk

Received: 13 June 2018; Accepted: 17 July 2018; Published: 20 July 2018

**Abstract:** The strive towards ever increasing automotive engine efficiencies for both diesel and gasoline engines has in recent years been forced by ever-stringent emissions regulations, as well as the introduction of fuel consumption regulations. The untapped availability of waste heat in the internal combustion engine (ICE) exhaust and coolant systems has become a very attractive focus of research attention by industry and academia alike. Even state of the art diesel engines operating at their optimum lose approximately 50% of their fuel energy in the form of heat. As a result, waste heat recovery (WHR) systems have gained popularity as they can deliver a reduction in fuel consumption and associated $CO_2$ emissions. Of these, the Organic Rankine Cycle (ORC) is a well matured waste heat recovery technology that can be applied in vehicle powertrains, mainly due to the low additional exhaust backpressure on the engine and the potential opportunity to utilize various engine waste heat sources. ORCs have attracted high interest again recently but without commercial exploitation as of today due to the significant on-cost they represent to the engine and vehicle. In ORCs, expansion machines are the interface where useable power production takes place; therefore, selection of the expander technology is directly related to the thermal efficiency of the system. Moreover, the cost of the expander-generator units accounts for the largest proportion of the total cost. Therefore, selection of the most appropriate expander is of great importance at the early stage of any automotive powertrain project. This study aims to review the relevant research studies for expansion machines in ORC-ICE applications, analyzing the effects of specific speed on expander selection, exploring the operational characteristics of each expander to further assist in the selection of the most appropriate expander, and comparing the costs of various expanders based on publically available data and correlations.

**Keywords:** expander selection; Organic Rankine Cycle; waste heat recovery; positive displacement expander; turbo expander; techno-economic analysis

## 1. Introduction

The transportation sector is responsible for one third of the global $CO_2$ emissions and approximately 14% of overall greenhouse gas emissions [1]. Light commercial and heavy duty vehicles emissions are responsible for one third of this percentage, although these vehicles account for only 5% of the vehicles in the EU [2]. EU legislation sets mandatory emission reduction targets, as the fleet average to be achieved by all new passenger cars is 95 g $CO_2$/km by 2021, reduced from 130 g $CO_2$/km in 2015 [3]. Regarding heavy duty diesel (HDD) engines, a 9% reduction in $CO_2$ is required by 2017 compared to 2010 [4]. Fuel consumption is not only limited by emission standards, but also related with a higher operating cost for heavy duty vehicles, which can reach up to 40% [5] as fossil fuel prices fluctuate. For all these reasons, the early pivotal demand to employ waste heat recovery systems such as ORCs to achieve lower pollutant greenhouse gas (GHG) emissions and fuel consumption will become more intensive in the near future [6].

Waste heat recovery in internal combustion engines lies in the range of a low to high grade heat rate, depending on the engine operating conditions and the heat sources. The main heat sources where the fuel energy is wasted are the exhaust gases, the cooling systems [7], and the relatively smaller amounts available from the EGR system [8]. Exhaust gases account for most of the wasted heat, burnt at a range of 33% to 40% [9,10]. Up to 33% of this wasted heat can be recovered and converted into useful work [11]. On the other hand, wasted heat through engine coolant accounts for up to 30% [12]. Due to the lower coolant temperature, the recovery potential of coolant energy is much lower than that of exhaust gas energy [13]. The remaining wasted heat dissipates through other engine components such as exhaust gas recirculation (EGR) and charge air cooler (CAC) systems. The efficiency of the heat source is a trade-off between its quantity (energy contained in the heat source) and quality (temperature range of the heat source) [7]. In some applications, the wasted heat has a high temperature but lower exhaust gas mass. This leads to less waste heat loss as a percentage of fuel input [14]. Nonetheless, Dolz et al. [15] stated that Rankine Cycle efficiency depends on the heat source temperature. According to Nadaf and Gangavati [16], exhaust gas presents a high quality and quantity, engine coolant presents a high quality but low quantity, and the EGR system presents a low quality but high quantity. Therefore, engine exhaust gases are the most attractive heat source among the different waste heat sources due to their high exergetic content [9,17]. EGR can also be used to recover waste heat as the level of useful exergy in the EGR is higher than that of the after-turbine exhaust, and due to the higher temperature range [18]. Teng et al. [19] showed that EGR and exhaust gases together are the most interesting recoverable heat sources due to their high exergy value. Their results showed that 20% of the wasted heat can be recovered, resulting in an 18% ORC efficiency. However, Espinosa et al. [20] mentioned that using both exhaust gas and EGR adds complexity and increases cost. Boretti [21] used both the exhaust gas and the engine coolant as the heat source in his study. He stated that the fuel conversion efficiency increased by a maximum of 6.4% when using exhaust gas only and by a maximum value of 2.8% when utilizing only engine coolant. When both heat sources were combined, the efficiency increased up to 8.2%.

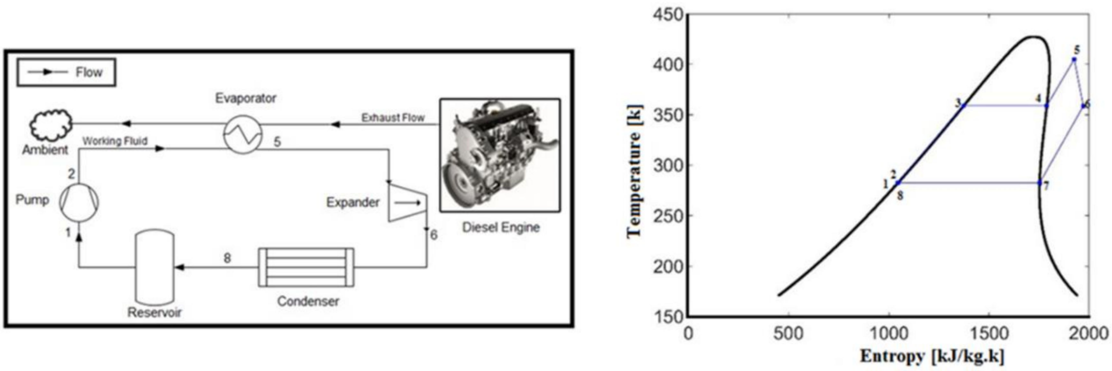
The effect of implementing an ORC WHR system in the powertrain of a diesel or gasoline power assisted vehicle in terms of ORC efficiency, engine efficiency, and powertrain performance, has been presented by many studies in the past. On a theoretical base, the ORC efficiency of a heavy-duty diesel engine and a light duty gasoline engine can reach up to 20% and 14%, respectively, by using ORC as a bottoming cycle [22]. Dolz et al. [15] proved that ORC efficiency could reach up to 6% when using a 12 L two-stage turbocharged heavy duty diesel engine. The experimental results of Cipollone et al. [23] showed that the efficiency of the regenerative ORC system ranged between 3.8–4.8% when using the exhaust gas of an IVECO F1C diesel engine (Industrial Vehicle Corporation, Turin, Italy). In another experimental study, Yang et al. [24] utilized a six-cylinder, four-stroke diesel engine where the ORC efficiency reached a maximum value of 9.9%. In a theoretical study, Katsanos et al. [25] showed that cycle efficiency could be much higher when using an exhaust gas and EGR system, and the maximum ORC efficiency achieved was around 26%. On the other hand, the experimental results of Wang et al. [26] showed a maximum Rankine efficiency of 7% using both the exhaust gas and coolant of a gasoline engine. ORC has also been investigated in gasoline applications. Galindo et al. [27] utilized the exhaust of a 2 L turbocharged gasoline engine to present a 6% ORC efficiency. Even though the exhaust quality of Diesel engines is lower than that of gasoline engines [28], the ORC shows a better performance with diesel engines than gasoline engines due to the higher mass flow rate values of higher displacement heavy duty diesel engines.

Sprouse and Depcik [8] made a comprehensive review of ORC applications in ICEs, but they concentrated on the history rather than the criteria of expander selection, cost, or the operating characteristics. Panpan et al. [29] focused on the scroll expanders only. Bao and Zhao [30] reviewed a selection of organic fluids and expansion machines in different ORC applications; however, they mainly focused on working fluids. They also discussed turbo-expanders in general, without going deeply into the different types (i.e., axial and radial turbines). Although the above literature agrees on the great importance of the expander type on the cycle performance, none of them explored the criteria of selecting the optimum expander at the early stage of a project. The target audience of this work are the engineers and scientists related to the development of renewable and clean energy technologies, specifically in the field of the integration of renewable energy with conventional processes (ORC implement to ICEs). The expansion machines are one of the prime components of these thermal power systems as they are applied for energy conversion in automotive ICEs. The novelty of the paper is three-fold in that it covers a comprehensive and comparative reporting of the most recent levels of performance achieved by applicable ORC expanders, it provides a comparative basis for cost comparisons, and it can be used as a selection tool for ORC and expander engineers and scientists. In more detail, the paper presents a performance summary of different prototypal and commercial expander types in ICEs. Also, a general comparison between the different expander technologies in terms of pros and cons is presented. In addition, the operating characteristics of each expander type are explored for different applications. More importantly, this study presents a guide based on specific speed $N_s$ for a wide range of applications. This guide can be utilized at the early stage of a project to assist in the selection of the optimum expansion machine for the corresponding application. Last but not least, this study compares the cost of the different types of expanders which can be combined with the specific speed and operating characteristics for further assisting in the expander selection.

### *1.1. Organic Rankine Cycle (ORC) Technology*

Recently, technologies to recover wasted heat in internal combustion engines have been studied intensively. According to the open literature [31–34], there are several common technologies in WHR; namely, the organic Rankine Cycle (ORC), Thermo-electric generation (TEG) [35], and turbo-compounding (TC) [36]. However, the investigation of TEG and TC is beyond the scope of this study.

The Organic Rankine Cycle (ORC) system is regarded as the most potential candidate in generating electricity from low to medium heat sources because of its simplicity and the availability of its components [37]. It is also mostly implemented in practice [38]. The Organic Rankine Cycle (ORC) is a Clausius–Rankine Cycle, but with an organic fluid instead of steam. In the ORC (Figure 1), a fluid, which is in a liquid state, is pressurized isentropically in the pump (processes 1–2). Then, the fluid is heated and vaporized in the boiler (evaporator), which causes the fluid to change its state from liquid to vapour (processes 2–5). Next, the fluid is expanded in the expander that extracts energy from the superheated working fluid to produce power (processes 5–6). Finally, it is condensed in the condenser that changes the fluid state from vapour to liquid again (processes 6–8) and is pumped again to restart the cycle. Specifically, for the waste heat to power an application on a small scale, the ORC system compared to the steam Rankine Cycle is more effective in heat to power conversion if the turbine, pump, and generator are on the same shaft and the pump is of a turbo type [8].




(**a**) (**b**)

**Figure 1.** (**a**) Schematic representation of the ORC-ICE; (**b)** Temperature-Entropy (T-s) diagram of cycle.

*1.2. Configurations of Rankine Cycles*

There are two main configurations of traditional Rankine Cycles, namely; reheat and regenerative, presented in Figure 2. In the reheat configuration, the working fluid is expanded in a high-pressure turbine (processes 1–2) and sent back to the evaporator, where it is reheated at constant pressure to the inlet temperature of the high-pressure turbine. Then, the steam is sent to a low-pressure turbine and expands to the condenser pressure (processes 2–3). The utilization of the reheat Rankine Cycle increases the average temperature at which heat is transferred to the working fluid, and hence improves the cycle efficiency by 4–5% [39]. This configuration also increases the quality at the expander exit [40], which in turn reduces the moisture content that damages turbine blades.

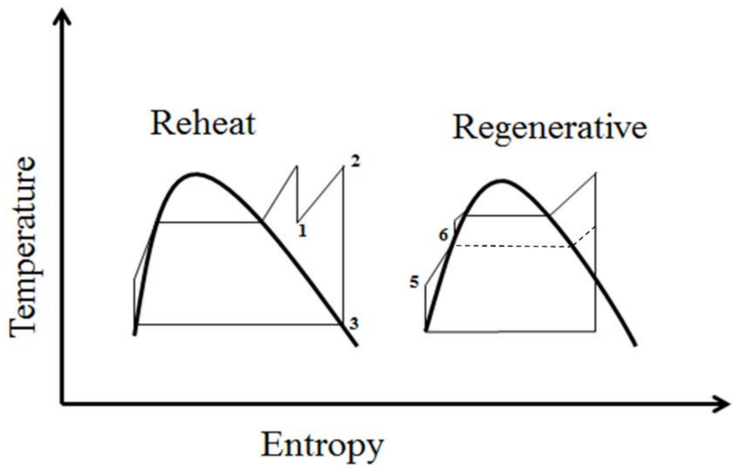



**Figure 2.** Representation of the (**left**) reheat cycle and (**right**) regenerative cycle.

The regenerative Rankine Cycle is utilized when the working fluid temperature after expansion is still relatively high. A portion of the high temperature working fluid preheats the liquid at the condenser exit before it enters the evaporator (processes 5–6). Using the regenerative Rankine cycle, the thermal efficiency will increase as a result of the elevated average temperature [40].

*1.3. ORC in Internal Combustion Engines*

Utilization of ORC systems in mobile applications is not a new idea. A first concept on a train had already been commercialized in the 1920s, taking advantage of the price difference between diesel and coal [41]. Unfortunately, this system quickly became uncompetitive because that difference was not profitable anymore [42]. Interest in waste heat recovery can be traced back to the 1970's energy crisis when the price of oil rose significantly due to the energy crisis of 1973 [43]. However, several systems were developed, mostly for trucks or marine applications, and then this interest disappeared until the 2000s, when automotive manufacturers started being interested in that technology again [42]. In 1976, Patel and Doyle [44] built a prototype of an ORC system that was used as a bottoming cycle in a Mack 676 diesel engine. The authors stated that at the peak power condition, 36 additional horsepower was produced, resulting in a gain of 13% in power without additional fuel. However, only a few commercial ORC power plants have been realized since the 1980s [45]. Research studies about ORC technology in internal combustion engines have increased since the last decade. In 2007, Endo et al. [46] installed a Rankine Cycle in a 2.0 L honda stream spark ignition engine (R20A) using both the exhaust gas and engine coolant as the heat sources. The results of the vehicle testing showed that, at constant speed (100 km/h), the thermal efficiency increased from 28.9% to 32.7% (13.2%). Also, a project by the DOE Super Truck Program [47] demonstrated better than 50% brake thermal efficiency. Quoilin [48] conducted an experimental test of a small scale ORC system using hot air at a temperature ranging from 150 to 200 °C as the heat source. His results showed a maximum cycle efficiency of 7.4%. Ringler et al. [49] analyzed the potential of the Rankine Cycle as an additional power generation process that used the waste heat of a car engine. Their experimental test showed that waste heat recovery could produce an additional power output of about 10% at typical highway cruising speeds. Hence, based on the above selective references, ORCs are very promising technologies in waste heat recovery systems. Moreover, different heat sources can be utilized, namely; engine exhaust gas, exhaust gas recirculation (EGR), and engine coolant.

Nonetheless, the weight of the ORC system and the engine power are two of the most critical parameters that affect fuel consumption and harmful exhaust emissions [50]. As the vehicle weight increases, inertia and rolling resistance increase, resulting in an increased fuel consumption as the engine is required to propel the vehicle to the required speed. Moreover, frictional forces acting on the vehicle operating at constant speeds increase because of the added weight [51]. In the open literature, only few works considered ORC weight when implemented as a bottoming cycle in automobiles. In a previous study [50], the authors proved that a 10% worse power-to-weight ratio can deliver up to 42.5% additional benefits on fuel consumption and power increase of the ORC system. In their modelling study, Oh et al. [52] stated that a 10% increase in the vehicle weight could result in a 4–6% increase in fuel consumption. Imran et al. [51] presented a method to estimate the weight of different ORC components as a function of their capacities. In order to obtain the weight of the expander, Usman et al. [51] used data of scroll compressors with similar power ratings to plot the expander weight against power. The results showed that the added weight of the ORC system (140.52 kg) caused a power loss in the range of 0.82–2.28 kW. The weight of the expander was in the range of 4.2 to 12.8% of the total weight. Battista et al. [53] considered the weight of the ORC system to study its effects on the fuel consumption of a light duty diesel engine. Assuming 50 kg as extra weight added to the original vehicle weight (3350 kg), the results showed that the fuel consumption increased by 1.25% at 1000 rpm and 0.7% at 3500 rpm. Assuming a 20 kg ORC system, Horst et al. [54] stated that the fuel consumption increased by 1.5%. In addition to the weight issue, Boretti [55,56] stated that the ORC technology has several disadvantages, such as increased backpressures, more complex packaging, more complex control, troublesome transient operation, and finally the cold start issues that prevent the uptake of the technology.

### *1.4. Working Fluids in ORC*

Choice of working fluid for an ORC system is of key importance for the cycle efficiency and net work. ORC systems should only utilize working fluids with low Global Warming Potential (GWP) and Ozone Depletion Potential (ODP) [57,58]. Based on Lee et al. [59], the system efficiency of an ORC correlates with the fluid's normal boiling point, critical pressure, and molecular weight. The main difference between conventional Rankine Cycles and organic Rankine Cycles is that the ORCs use organic fluids (hydrocarbons or refrigerants) instead of water. Organic fluids exhibit unique advantages over water or steam [60] since they are better adapted than water to the lower heat source temperatures, making the ORC systems able to efficiently produce shaft-work from medium temperature heat sources up to 370 °C [61]. Steam can be applied in applications where heat source temperatures are very high without the fear of thermal decomposition due to its thermal stability [31], while organic fluids can be used in low temperature heat sources because of their lower normal boiling point, which enables them to evaporate and recover thermal energy from low grade heat sources better than steam. Statistical investigations indicate that low-grade waste heat accounts for 50% or more of the total heat generated in industry [61]. Moreover, more advantages can be exhibited by the ORC over conventional Rankine cycles such as a simple control system, and a cheap and simple turbine [62]. Another characteristic of the organic fluids is the low speed of sound. As a result, this speed is reached faster in an ORC than in a steam cycle and constitutes an important limitation as high Mach numbers are related to higher irreversibilities due to shock losses resulting in lower turbine efficiencies [62]. However, the impact is offset by the fact that ORC turbines have more favorable specific speeds due to working fluid characteristics.

The selection of the working fluid is determined by the application and the waste heat level [63]. Based on the slope of the saturation vapour line, as shown in Figure 3, working fluids can be classified into three groups, namely; wet, dry, and isentropic. The dry and isentropic fluids have enormous advantages for turbo-machinery expanders since they leave the expander as superheated vapour and eliminate the corrosion that results from liquid droplets impingement in the turbine blades during the expansion [64]. Another advantage is that there is no need for overheating the vapour before entering the expander, which means that a smaller and cheaper heat exchanger can be used [65]. Moreover, the superheated apparatus is not needed when using dry and isentropic fluids [66]. However, if the fluid is too dry, the expanded vapour will leave the turbine with substantial superheat, which is a waste and adds to the cooling load in the condenser [65,66]. The presence of a hydrogen bond in certain molecules such as water, ammonia, and ethanol may result in wet fluid conditions due to larger vaporizing enthalpy. Hence, wet fluids are regarded as inappropriate options for ORC systems [66,67]. Isentropic fluids are free from the concern of the moisture content (fluid droplets), and the only issues of concern are their cost, chemical stability, and safety [68,69]. The next paragraph is a summary of the studies related to working fluids available in the open literature.

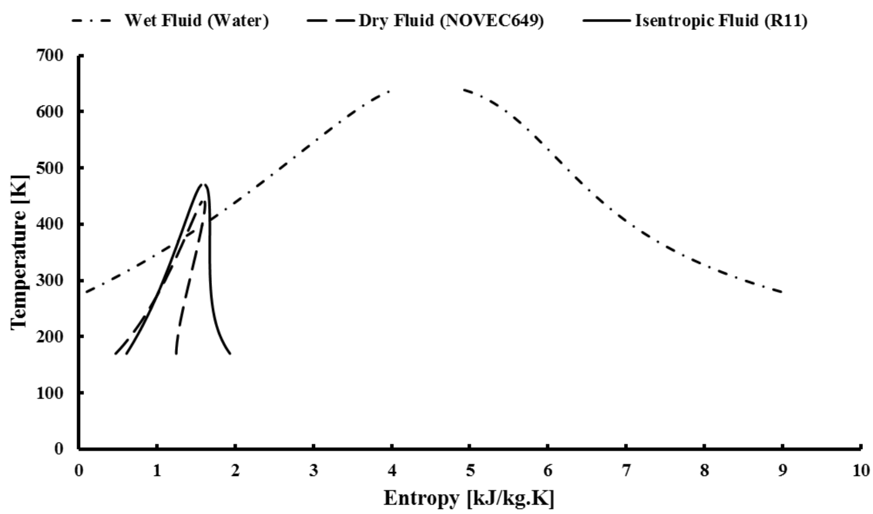



**Figure 3.** Temperature-entropy diagram for different types of working fluids.

Hung [70] analyzed and compared the efficiency and irreversibility of the ORCs using various dry fluids, namely; Benzene, Toluene, p-Xylene, R113, and R123. The result of the study showed that p-Xylene presented the highest efficiency, while the lowest efficiency was shown by Benzene. Also, R113 and R123 presented a better performance in recovering low temperature waste heat.

Saleh et al. [71] presented a study of 31 pure component working fluids for organic Rankine cycles in geothermal power plants. Among the studied fluids, n-hexane showed the highest thermal efficiency (14.14%), while R32 showed the lowest thermal efficiency (0.53%). Hung et al. [69] conducted a study to identify the suitable working fluids that may yield high system efficiencies in an organic Rankine Cycle (ORC) system. The results showed that R11 and C7H8 had the highest system efficiency, around 8.5%, while R152 had the lowest system efficiency of 6.7%. The authors stated that wet fluids with very steep saturated vapour curves in the temperature-entropy diagram had a better overall performance in energy conversion efficiencies than that of dry fluids. The authors also stated that dry fluids in general generated superheated vapour at the turbine exit, which reduced the area of network in the temperature-entropy diagram, and a generator might be needed in order to relieve the cooling load of the condenser. Wang et al. [72] constructed a MATLAB code to investigate the performance of nine working fluids. The results showed that R11, R141b, R113, and R123 had slightly higher thermal efficiencies than others. In terms of safety levels and environmental impacts, the results showed that R245fa and R245ca were the most suitable working fluids for an engine waste heat-recovery application. Shu et al. [73] investigated the influences of Alkanes as working fluids using high-temperature exhaust heat recovery of a diesel engine. Cyclohexane showed the best performance with an improvement of 10% in brake specific fuel consumption (BSFC). However, the authors stated that higher flammability and toxicity had to be taken into account when Alkane-based working fluids were used, which necessitated the utilization of good sealing and excellent ventilation. Ringler et al. [49] conducted an experimental test to investigate the results of using a high temperature source (exhaust gas) and low temperature source (engine coolant). The authors concluded that water would be a preferable working fluid for a system which used exhaust gas with a high temperature level heat source ($T > 300$ °C), whereas an alcohol (e.g., Ethanol) would be the right choice for a low temperature system. Javanshir et al. [74] studied the performance of a regenerative ORC system using 14 dry fluids. The results showed that Butane followed by iso-Butane and R113 offered the highest specific net work output. Also, working fluids with higher specific heat $(C_p)$ produce a higher specific net work output, while working fluids with a higher critical temperature produce higher thermal efficiency. Dai et al. [75] conducted an experimental test to investigate the thermal stability of hydrofluorocarbons using a fluoride ion as an indicator of the fluids decomposition. The results showed that most common hydrofluorocarbons had thermal stability temperatures that were suitable for supercritical ORCs. However, they must be used below the decomposition temperatures to ensure system safety. Similarly, Invernizzi et al. [76] investigated the thermal stability of three representative hydrocarbons used as working fluids in organic Rankine cycles (ORC): n-pentane, cyclo-pentane, and toluene, using vapour pressure as an indicator. The results showed that cyclo-pentane was very stable at 350 °C after eight hours with a decomposition rate twenty times lower than the corresponding value for n-putane. The results also confirmed that the toluene sample remained remarkably stable at higher temperatures, which nominated it as a preferable fluid for high temperature application. Saloux et al. [77] proposed a methodology for selecting the most appropriate organic fluid for WHR applications based on the primitive variables such as: the working pressures, temperatures, mass flow rate, etc. The analytical results of He et al. [78] showed that fluid R236FA had the highest thermal efficiency of 21.6%, and the fuel economy of a twelve-cylinder four stroke stationary natural gas engine could be improved by 14.7% compared to that without ORC. Recently, Seyedkavoosi et al. [79] compared three working fluids in terms of exergy efficiency using the exhaust gas of a 12 L cylinder gas-fired internal combustion engine. The investigated fluids were R123 (dry), R134a (isentropic), and water (wet). The results showed that R123 presented the best performance. The authors concluded that using dry fluids such as R123 for ICE applications could have the potential to realize maximum useful power and exergy efficiencies. Working fluid mixtures have also been proposed to increase the efficiency of the cycle by inducing a temperature glide and the main heat transfer process becomes isobaric instead of isothermal. The peculiarity allows a reduction of second law losses in both the evaporator and condenser, and the overall efficiency is increased. However, the mixtures have a lower heat transfer coefficient and require larger heat exchangers, increasing the cost and size. The control of composition and accurate refill also incur complexities, making them not very suitable for waste heat recovery application at this stage of development. [80,81].

### 1.5. Comparison between ORC System and Other Technologies (Thermo-Electric Generation and Turbo-Compounding)

The selection of appropriate WHR technology depends on the application, system size, and heat source. However, ORC systems provide an attractive combination of efficiency and affordability for engine exhaust WHR [8]. They also exhibit great flexibility, high safety, and low maintenance requirements [82]. Intensive comparison of the technologies is beyond the scope of the paper. However, a brief comparison is discussed in the next paragraphs.

Compared to turbo-compounding, ORC exhibits higher brake specific fuel consumption (BSFC) reduction potential [83]. The main disadvantage of turbo-compounding is the interaction with the engine. This interaction increases the exhaust backpressure of the engine, which affects the exhaust gas after-treatment system and the engine performance [84]. In a well-designed ORC, the net backpressure can actually be lower due to excessive cooling of the exhaust gas in the evaporator [85]. Moreover, the combination of the ORC system and the automotive engine increases the thermal efficiencies of the engine without increasing the exhaust back-pressure [86,87]. Another main drawback of turbo-compounding technology is the increase in pumping losses due the existence of the extra turbine. Mechanical turbo-compounding has traditionally only been applied to large diesel engines because of the complexity of mechanically coupling a high-speed turbine to the engine crankshaft [88]. However, the ORC system is a heat energy recovery method, which means a heavier and more expensive system, while the turbo-compounding system is a mechanical energy recovery method and hence, cheaper and lighter. Also, implementation of the ORC system requires a complex technological architecture, which makes it an unfavourable solution for a small scale mobile application [89]. Overall, the Rankine Cycle can offer substantial gains in fuel economy, potentially in the order of 20%, compared to much established turbo-compounding [33].

The other waste heat recovery technology of note in terms of the current research focus is Thermoelectric Generation (TEG). Thermoelectric Generation has three main challenges. Firstly, it has generally exhibited a substantially inferior efficiency, typically less than 4% [90]. The second challenge is the bigger size of the radiator and extended piping to the exhaust manifold [31]. Thirdly, thermoelectric generators are not mature yet and some efficient materials are yet to be manufactured [84]. However, new nano-crystalline or nano-wire thermoelectric materials are currently in the development stage to improve the conversion efficiency of thermoelectric generators [91]. When compared to ORC, TEG technology is used at high temperatures and only when small amounts of power are needed. ORC, on the other hand, is used when a low to medium temperature heat source is utilized [92]. Although TEG has a low effect on engine performance and can improve the engine power up to 17.9% [93], energy from the gases is lost and causes an increase in pumping losses, which reduces the engine performance [94]. Beside the challenges mentioned above, TEG technology is expensive and places an extra load on the cooling system due to the large temperature difference between the hot and cold surfaces [95]. It is also very time consuming since the evaluation of system stability requires prolonged aging and thermal cycling tests [96]. However, TEG technology has a light weight, while the ORC has a high volume and weight. Also, Thermoelectric technology is environmentally friendly [97], while ORC could be harmful when using toxic working fluids.

## 2. Expander Technologies for ORCs in Internal Combustion Engines (ICEs)

The selection of an appropriate expander type is very crucial since the expander is a critical component in an efficient and cost-effective ORC system. Choice of the expander type strongly depends on working conditions, type of working fluid, space and weight restrictions, and the size of the system [10]. Other important factors should be considered when selecting expanders such as a high isentropic efficiency, pressure ratio, power output, lubrication requirements, complexity, rotational speed, dynamic balance, reliability cost, working temperatures and pressures, leaking, noise, and safety [30,98]. Expanders can be classified into two groups: displacement expanders (volumetric expanders) and turbomachine expanders (velocity expanders). Most of the ORC systems have been developed with the scroll and vane type, thanks to their low cost and competitive efficiency for a micro scale power range [99]. Scroll and vane expanders have been cheaply available in the past from refrigeration or compressed air technology, where they worked in the role of compressors, while small turbines are still rather seldom (and therefore more expensive). However, these devices are not adapted for placement in small and restricted places like the interior of a conventional car [99]. On the other hand, Turbomachine expanders are preferred for a high temperature WHR application with an expected regenerated power in the order of 15 kW [100].

### 2.1. Positive Displacement Expanders (PDEs)

As the name implies, PDEs are positive displacement machines whose principal feature is to have a fixed volumetric ratio. In positive displacement expanders, fluid is forced into a closed volume, and then the fluid is pushed out with lower pressure, resulting in mechanical work due the expansion process. PDEs have several types with different working principles. These types are scroll expanders, screw expanders, reciprocating piston expanders, and rotary vane expanders, and they are schematically presented in Figure 4.

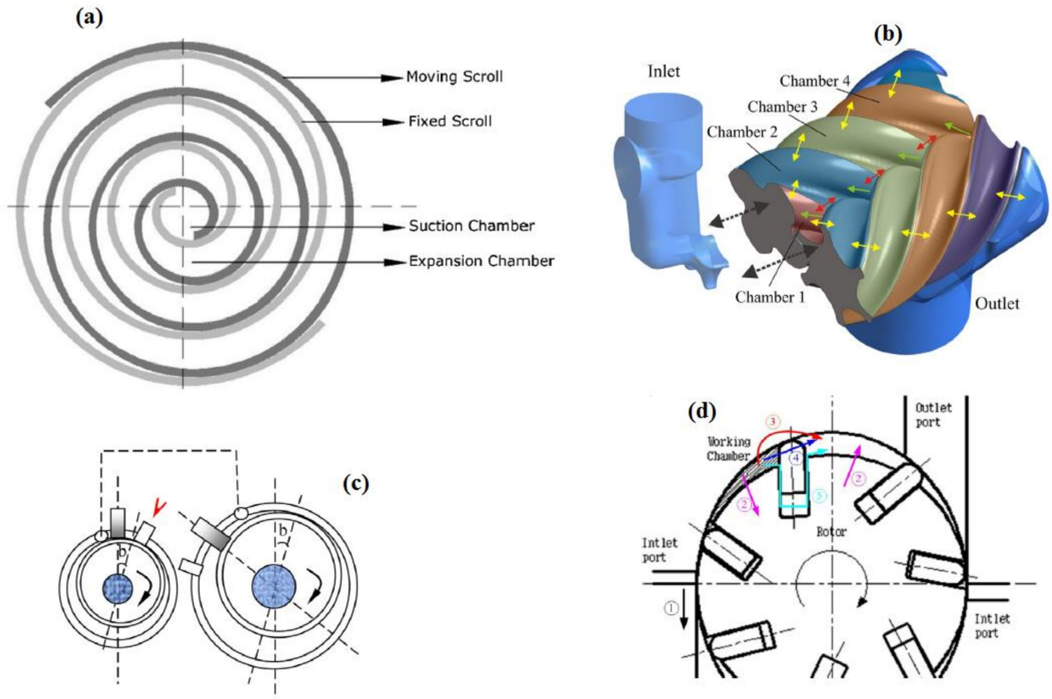



**Figure 4.** (**a**) Scroll expander [101]; (**b**) screw expander [102]; (**c**) piston expander [103]; and (**d**) rotary vane expander [104].

Compared to all PDEs, the scroll expander has the most complicated geometry [30]. It can be categorized into two main groups: compliant scroll expanders and cinematically constrained scroll expanders. In the former type, lubrication is compulsory to operate efficiently without causing wear due to the contact of the wrap sidewalls. In the latter category, lubrication is not required due to the existence of the linking mechanism between orbiting and fixed scrolls. An advantage of the cinematically constrained scroll expanders is that they do not require inlet and exhaust valves;

therefore, noise is reduced and the durability of the device is increased [62]. However, sealing is required to prevent internal leakage.

In screw expanders, lubrication is required due to the direct contact between the two lobes. Due to the high rotational speed of screw expanders, reduction gear boxes and speed control equipment might be required [30]. When a dry fluid is used as the working fluid in ORC systems, seals are required, which highly increase the cost of the machine [105]. Both scroll and screw expanders can work with wet fluids without damaging the devices since they can generally accept a large liquid mass fraction [106].

Reciprocating pistons are complex devices that require precise timing of the intake and exhaust valves [30]. They also have large friction loss, about 80% of total losses [107], due to the large interacting surfaces. However, piston expanders usually show lower internal leakage than scroll and screw expanders [106]. This also results in an easier design and construction [108]. Like some PDEs, lubrication is required in piston expanders, but involves some difficulties because the oil should be mixed with the working fluid, which reduces efficiency of the cycle [109].

Rotary vane expanders, which are often based on the Wankel concept [110,111], do not require the use of valves [112]. Compared to other expander concepts, the rotary vane expander has a simpler structure and easier manufacturing process [104]. It has low noise- vibration and high volumetric expansion ratios [113]. However, it has a poor performance at low speed due to the higher influence of internal leakage at a low speed [114], and shows implementation difficulties [115]. Also, the machine must be lubricated to minimize wear and enhance sealing [30]. Table 1 presents the summary of ORC works with various expansion machines.

**Table 1.** Summary of ORC Studies with Different Expansion Machines.

| Authors | Ref. | Expander Type | Power Range (kW) | Expander Speed (rpm) | Mass Flow Rate (kg/s) | Working Fluid |
|---|---|---|---|---|---|---|
| Han et al. | [116] | Radial | 25–250 | 7500–16,500 | 3–7.4 | R245fa |
| Zheng et al. | [117] | Radial | 640–780 | 8000 | 41.1 | R134a |
| Wong, C.S. | [118] | Radial | 16–27 | 67,660 | 1.18–1.82 | R134a |
| Nithesh et al. | [119] | Radial | 5–12 | 22,000 | 0.1–1 | R134a |
| Al Jubori et al. | [120] | Radial | 3–8 | 20,000–60,000 | 0.1–0.5 | R245fa |
| Yamamoto et al. | [121] | Radial | 0.54–0.55 | 35,000 | 0.058–0.065 | R123 |
| Russell et al. | [122] | Radial | 6.8–8 | 30,000–207,300 | 0.274 | R245fa |
| Wang et al. | [123] | Scroll | 0.58–0.64 | 2000–3700 | 0.022–0.0245 | R134a |
| Oudkerk et al. | [124] | Scroll | 2.8 | N/A | 0.139 | R245fa |
| Quoilin et al. | [37] | Scroll | 0.2–2 | 700–4000 | 0.05–0.095 | R123 |
| Ayachi et al. | [125] | Scroll | 0.2–2.5 | 3010–3090 | 0.07–0.17 | Water |
| Papes et al. | [126] | Screw | 0–8 | 400–8000 | 0.07–0.28 | R245fa |
| Zhang Y. | [127] | Screw | 10–35 | 1600–3200 | N/A | copressed air |
| Tang et al. | [128] | Screw | 200–560 | 1250–6000 | 18–66 | R123 |
| Wang et al. | [129] | Screw | 4.4–5 | 2200–3000 | 0.14 | compressed air |
| Ziviani et al. | [130] | Screw | 1–8 | 2000–3300 | 0.51 | R245fa |
| Zhang et al. | [15] | Screw | 2–11 | 1000–2600 | N/A | R123 |
| Oudkerk et al. | [131] | Piston | 0.25–2 | 1000–4000 | 0.03–0.11 | R245fa |
| Galindo et al. | [27] | Piston | 0.2–1.6 | 1000–5000 | 0.01–0.75 | Ethanol |
| Zheng et al. | [132] | Piston | 0.16–0.32 | 400–700 | N/A | R245fa |
| Lemort et al. | [133] | Piston | N/A | N/A | 0.1 | Water |
| Han et al. | [134] | Piston | N/A | 1500 | 0–1.5 | R123 |
| Vodicka et al. | [135] | Vane | 0.55–1.075 | 2550–4012 | 0.056–0.07 | hexamethyldisiloxane |
| Kolasiński | [136] | Vane | 0.1–0.4 | N/A | N/A | R123 |
| Antonelli et al. | [136] | Vane | 3.9–31 | 500–1500 | 0.012–0.032 | R245fa |
| Jia et al. | [137] | Vane | 0.8–0.955 | 800–1800 | N/A | N/A |
| Tahir et al. | [138] | Vane | 0.012–0.031 | 2600–3000 | N/A | R245fa |
| Al Jubori et al. | [120] | Axial | 2.8–7.8 | 10,000–60,000 | 0.1–0.5 | R245fa |
| Seta et al. | [139] | Axial | N/A | 2500–20,000 | 10–150 | N/A |
| Shadreck et al. | [140] | Axial | N/A | 5000–65,000 | 0.909 | R245fa |
| Lazzaretto et al. | [141] | Axial | 260–350 | 4101–14,561 | 10–100 | R245fa |

N/A: Not applicable.

#### 2.1.1. Brief History of Positive Displacement Expanders in Organic Rankine Cycle-Internal Combustion Engine

##### Scroll Expanders

In 1993, Oomori and Ogino [142] conducted an experimental test for a scroll expander with HCFC123 as the working fluid. The engine water jacket of a four-cylinder passenger car was employed as the heat source. When increasing the pressure ratio, the results showed that the energy recovery began to drop as the revolution became lower than 800 rpm due to the drop of expander efficiency. The results also showed that the expander efficiency increased up to approximately 50% at a 1000 rpm expander speed, and then began to drop due to the deterioration of sealing performance between the rotor and the housing in line with the lowered revolution speed, which resulted in leakage of the working fluid. In terms of the overall performance, the test results showed that approximately 3% of the engine output energy was recovered at the ambient temperature of 25 °C, with a maximum energy recovery of 400 W.

In 2003, Kane et al. [143] tested an ORC system to recover the thermal energy from a bio-gas diesel engine. The cycle was a bottoming cycle with R134a as the working fluid and the scroll expander was modified from standard hermetic scroll compressor units. The laboratory tests showed that the total power range was from 3 to 10 kWe, and the maximum overall superposed cycle efficiency was (14.1 ± 0.2)%. The authors stated that this performance was satisfactory due to the relatively low supply temperature (up to 165 °C) and low power range (up to 10 kWe).

In 2007, Kane et al. [144] conducted an experimental test to recover the wasted heat in a 200 kWe biogas engine coolant using a scroll expander Organic Rankine Cycle. The results showed that the ORC system could be operated even at a very low power output, 20% below its nominal design value, and a cycle net efficiency of 7% could be achieved for a low temperature application (<90 °C).

Two years later, Mathias et al. [145] presented an experimental test of two types of expanders, namely: Scroll expander and rotary vane expander. R123 was selected as the working fluid and the heat source was the exhaust gas of a stationary ICE. The scroll expander used in the test was a scroll compressor, but was modified and operated as an ORC expander in the experiment. The experimental results showed that the maximum isentropic efficiency and output power achieved by the modified expander were 83% and 2.96 kW, respectively. In addition, the reduction in diesel fuel was 4012 L, and the overall energy efficiency of the cycle was 7.7%.

In 2012, Clemente et al. [146] studied the performance of a scroll expander with R245fa and Isopentane as working fluids. The scroll efficiency and power output were higher when using R245fa as a working fluid at a 40 °C condensing temperature. Using the exhaust gas of the internal combustion engine, the authors stated that the designed cycle was able to recover 10 kW. Also, a net mechanical power of about 1 kW was delivered by the scroll expander, achieving a 10.8% thermal efficiency.

A year later, Guopeng Yu et al. [147] conducted a simulation study on an ORC system with exhaust gas and jacket water as heat sources, and with R245fa as the working fluid. The results showed that the maximum expansion power and recovery efficiency were about 14.5 kW and 9.2%, respectively. The results also showed that the maximum expander pressure ratio and isentropic efficiency were 11.7 and 60%, respectively. The authors concluded that the ORC system could recover heat in exhaust gas effectively, but behaved badly when recovering heat in jacket water.

In 2014, Kim et al. [148] studied the performance of a scroll expander in off-design conditions using a dual-loop ORC with R134a as the cycle working fluid. The scroll expander was applied in the low temperature cycle using both the engine coolant and the condensation heat from the high temperature as the heat sources. The authors claimed that at the high temperature cycle, the scroll expander was not suitable due to the uneven thermal deformation of the scrolls that was difficult to handle. The results showed that a gradual decrease in the expansion efficiency was observed during the under-expansion operation (i.e., the design pressure ratio was lower than the operating pressure ratio), whereas a rapid decrease in the expansion efficiency was apparent during the over-expansion operation (i.e., the design pressure ratio was higher than the operating pressure ratio).

Recently, Petr et al. [149] conducted a simulation study to investigate the performance of an ORC system using Ethanol as the working fluid and a scroll expander as the expansion machine. The results showed an increase in the net power output of 7% compared to a conventional controller with operation points optimized at steady-state conditions.

Screw Expanders

In 2006, Leibowitz et al. [150] studied the possibilities of recovering power from low temperature heat sources. They utilized a twin screw expander due to its availability, low cost, and ability to operate at higher rotational speeds. According to the authors, these expanders can provide adiabatic shaft efficiencies of around 70% when working at a low speed. The screw expander would then develop a gross shaft output of 24 kW. The results also showed that the total increase in the power output was 8.5%.

In 2014, Yang et al. [24] conducted an experimental study to investigate the performance of ORC and a screw expander under various diesel engine operating conditions. They used R245fa as the working fluid and exhaust gas of a six-cylinder diesel engine as the heat source. The performance of the expander was investigated at different values of expander inlet pressure and rotational speed. The results showed that the power output and the isentropic efficiency of the screw expander increased when increasing the inlet pressure and rotational speed. The expansion ratio of the expander was shown to decrease when increasing the inlet pressure and rotational speed of the expander.

Recently, Wu et al. [151] conducted an experimental test to study the influence of different engine loads and expanders torques on the performance of ORC systems. They used the exhaust gas of a diesel engine and R123 as the heat source and cycle working fluid, respectively. The results showed that the power output of the single-screw expander increased in the form of a parabola with rising expander torque. In terms of ORC efficiency, the results showed that the efficiency increased when increasing the expander torque, with a maximum value of 6.48% at around 64 Nm. For the single-screw expander, a maximum power output of 10.38 kW and shaft efficiency of 57.88% were achieved at 1538 rpm. The maximum improvement of the diesel engine with ORC over that of the diesel engine without ORC was 1.53% at 250 kW.

Piston Expanders

In 1984, Poulin et al. [152] simulated an ORC system using several adiabatic diesel configurations. The results of the study showed that the maximum recoverable power was achieved by the turbocharged non-aftercooled diesel engine with a maximum brake specific fuel consumption (BSFC) improvement of 16.2%. The authors also stated that the expander efficiency improved with the fluid temperature and decreased slightly with increasing pressure. They also stated that the expander efficiency decreased when decreasing the condensing pressure.

Four years later, Kubo [153] conducted a study to evaluate the concept of a bottoming cycle to heavy duty transport diesel engine applications using three cycles, namely: steam cycle, ORC, and Stirling cycle. In the study, most of the components were similar, except for the expansion machine. In the ORC, a turbine was used with a rated speed of approximately 20,000 rpm, while the steam system used a two-cylinder reciprocating piston expander with a rated speed of 2000 rpm. In terms of fuel consumption, the ORC showed the best improvement with a value of 13.7%, followed by the steam cycle with a value of 13.3%, while the Stirling cycle was the worst with a value of 9.1%.

In 2012, Seher et al. [154] conducted a simulation and an experimental study using a 12 L heavy duty engine with ORC as waste heat recovery technology. In the simulation study, the results showed that the effective power output of the piston machine was 12 kW. The experimental results of the prototype showed that a mechanical output power up to 14 kW was realized, with mechanical efficiency better than 85% at 1500 rpm. The authors concluded that water and ethanol are favourable when using a piston expander.

In 2013, Wenzhi et al. [115] conducted a study to investigate the combined effects of heat exchangers and a single stage piston expander on the performance of the Rankine Cycle system. The authors used the exhaust gas of a high speed turbocharged diesel engine as the heat source. They concluded that the expander efficiency increased as the intake pressure was increased until it reached 2 MPa, at which point the efficiency started to decrease. Because of the enthalpy rise, the expander efficiency, power output, and global efficiency increased as the intake temperature increased in the experimental test, and a maximum power output rise of 12% was obtained when the diesel engine operated at 80 kW/2590 rpm and the expander worked at 4 MPa of intake pressure.

The following year, Daccord et al. [155] studied the influences of utilizing axial piston expanders on the performance of an ORC used in waste heat recovery. The authors employed the exhaust gas of a 1.6 L gasoline engine as the heat source. They concluded that the piston expander with a capacity of 183 $cm^3$ and an expansion ratio of 8 was the best compromise between low and high loads. Overall, the authors stated that an axial piston expander with non-lubricated hot parts would fit perfectly with mobile application thanks to its ability to deal with droplets in transient conditions.

A recent study by Chiong et al. [156] presented a new nozzle steam piston expander to recover the exhaust energy of a Cummins 6 BT turbocharged direct injection diesel engine. The results demonstrated that the power output of the nozzle piston expander was 3.53 kW higher than the conventional one when operating at 1500 rpm. Moreover, the improvement of the brake specific fuel consumption (BSFC) was 3% higher in the nozzle piston expander.

Galindo et al. [157] conducted a simulation and an experimental test to study the performance of an ORC system when used as a waste heat recovery in internal combustion engines. A swash-plate piston expander was used as the expansion machine and Ethanol as the working fluid in the ORC system. The performance of the expander was studied at different speeds with a maximum efficiency of 58.8% and power output of 2 kW at 2500 rpm.

Rotary Vane Expanders

In 2009, Teng and Regner [18] analysed the fuel saving benefit for a class-8 truck diesel engine equipped with an ORC system using the EGR as the heat source. The authors stated that since the expansion ratio in a supercritical cycle was too large for a single-stage turbine, the turbine was replaced with a positive displacement rotary expander with an expansion ratio of 6.4 for the subcritical cycle and 33 for the supercritical one. However, considering the technological enhancements, a pressure ratio of 33 may not be considered to be large enough to be not handled by a single stage in a turbine. The results of the analysis showed that a fuel saving up to 5% could be achieved and a further improvement could be achieved if the charge air cooling was integrated into the Rankine Cycle loop.

The following year, Tahir et al. [138] conducted an experimental study to investigate the performance of ORC systems when used as a WHR system to recover wasted heat from low-temperature heat sources. The authors selected a rotary-vane-type as the expansion machine and R245fa as the working fluid. The results showed that the highest measured efficiency and maximum power of the expander were 48% and 32 W, respectively, and the measured thermal efficiency was 3.82%.

In 2011, Zhang et al. [158] conducted a steady-state experiment to investigate the performance of an ORC system using the exhaust gas of a Toyota 8A-FE gasoline engine as the heat source and R113 as the ORC working fluid. The achieved expander isentropic efficiency was 79.42%, and the practical thermal efficiency of the system was about 14.44%.

Battista et al. [53] conducted a recent study to investigate the influence of implementing an ORC system when using the exhaust gas of a turbocharged IVECO F1C diesel engine as the heat source and R236fa as the cycle working fluid. In the study, a sliding vane rotary machine was selected as the ORC expander because, according to the authors, it was noiseless, compact, flexible from a geometrical point of view (diameter/length ratio), very reliable, and would not require important maintenance actions. The results of the test showed that a gross benefit of the ORC-based unit power was at a level of 4–5%.

2.1.2. Performance of Positive Displacement Expanders

In general, the performance of volumetric expanders is expressed in terms of overall isentropic effectiveness, shown in Equation (1).

$$\varepsilon = \frac{P}{\dot{m}(h_s - h_{ex,is})} \tag{1}$$

where $P$ and $\dot{m}$ are the power and the mass flow rate displayed by the expander, respectively. $h_s$ and $h_{ex}$ are the supply and calculated isentropic exhaust enthalpies, respectively. The performance of various types of expanders is investigated based on rotational speeds and mass flow rates, as shown in Table 1, and off design performance (isentropic efficiency vs pressure ratio), as can be seen in Figures 5–8,10,11. The terms isentropic effectiveness and isentropic efficiencies have been used interchangeably when referring to expander performance in the literature.

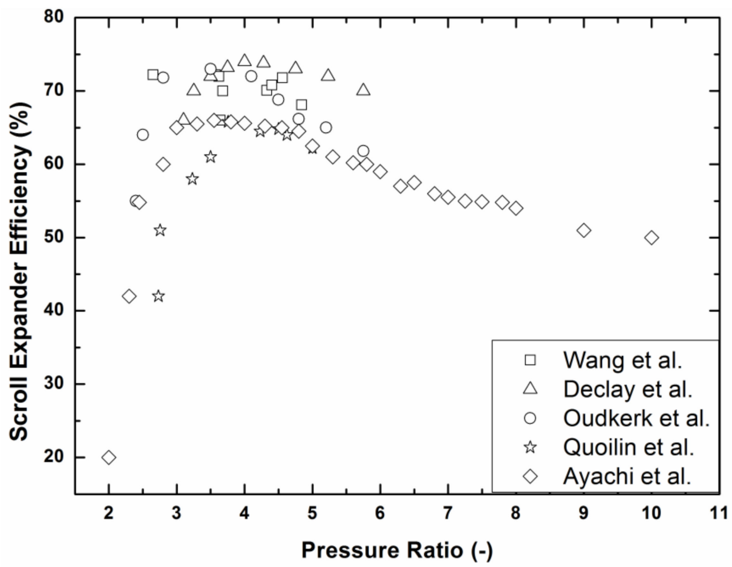



**Figure 5.** Performance of scroll expander.

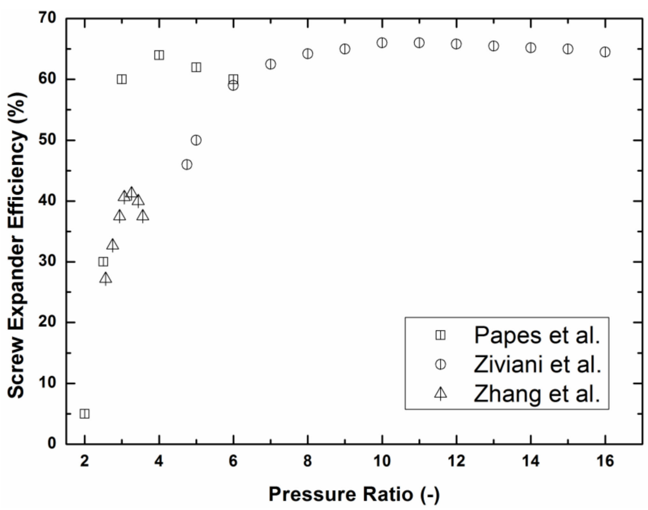



**Figure 6.** Performance of screw expander.

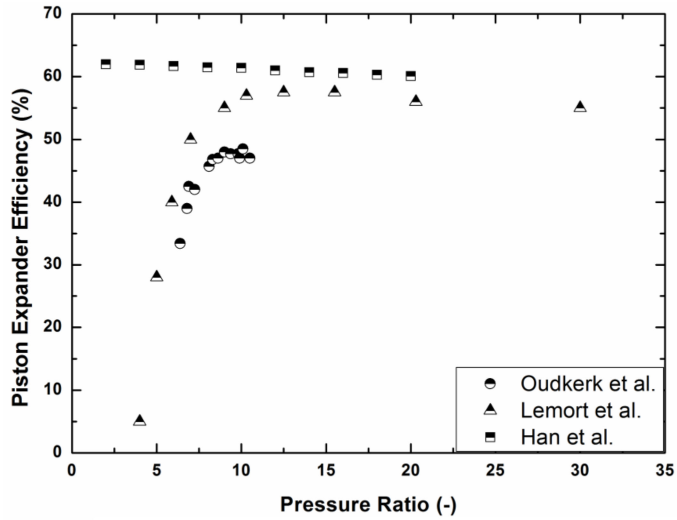



**Figure 7.** Performance of piston expander.

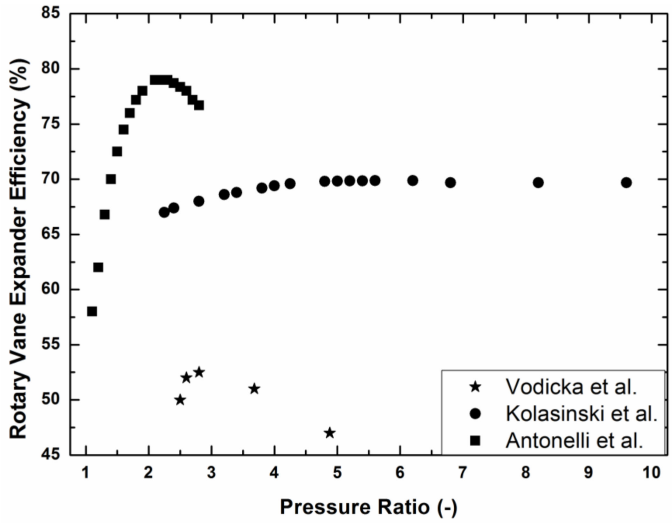



**Figure 8.** Performance of rotary vane expander.

Figure 5 presents the isentropic effectiveness of scroll expanders as a function of their pressure ratios based on the investigations achieved by [37,123–125,159]. As can be seen in Figure 5, scroll expanders can operate under high pressure ratios, with isentropic efficiencies in the range of 40–75%. The isentropic efficiency increases when increasing the pressure ratio until a certain value, and then starts decreasing. Moreover, scroll expanders are suitable choices for applications with low mass flow rates and power outputs. As can be seen in Figure 5, mass flow rates of the selected literature are in the range of 0.02–0.17 kg/s, while power outputs are in the range of 0.2–2.5 kW. The rotational speed values that result in an acceptable performance are between 700–4000 rpm.

The effectiveness of the screw expanders is investigated in Figure 6 based on the results of [126,130,160]. Efficiencies of screw expanders are relatively lower than those of scroll expanders. As can be seen in Figure 6, the efficiencies are below 65% and can be as low as 25%. On the other hand, crew expanders can handle higher mass flow rates and rotational speeds than scroll expanders, resulting in higher power outputs, as shown in Table 1. They can even produce much higher power, as shown in Tang et al. [128], due to the high mass flow rates (18–66 kg/s), which makes them preferable over the other volumetric expanders for a wide range of capacities. However, due to their high rotational speed

(400–8000 rpm) compared to the other volumetric expanders, a reduction gearbox might be required. Similar to scroll expanders, screw expanders can work under high pressure ratios (2–16).

Figure 7 presents the performance of the piston expander based on the results of [131,133,134]. Piston expanders can operate under relatively large pressure ratios (up to 30) compared to other volumetric machines with acceptable isentropic efficiencies (related to the high pressure ratios) due their large internal volume ratios. However, the power density for such machines is not very high. Among other volumetric expanders, piston expanders present lower power outputs (0.16–2 kW) with mass flow rates in the range of 0.03–1.5 kg/s. As can be seen Table 1, piston expanders operate with lower rotational speeds which make them directly attached to the generator without the need for a gearbox.

Rotary vane expanders are also capable of operating at relatively high pressure ratios up to 10, as can be seen in Figure 8, which presents the results of the studies by [112,135,136]. The reported power outputs in Table 1 are relatively low due to the lower mass flow rates (0.012–0.07). However, Antonelli et al. [112] increased the mass flow rates to high levels, which resulted in power outputs up to 31 kW. The isentropic efficiencies of such expanders are in the range of 45–80%. The lower values of efficiencies and power outputs are due to operational losses such as leakage friction loss and leakage [161]. As can be seen in Kolasinski et al. [136], presented in Figure 8, the isentropic efficiency is a flat curve over a wide range of operating conditions. Moreover, rotary vane expanders have low rotational speeds (up to 4000 rpm), as can be seen in Table 1, which enable them to be directly coupled to the generator.

Overall, rotational speeds of volumetric expanders are relatively low, which enable them to be directly coupled to the generator without the need for a reduction gearbox. Moreover, operating with two-phase fluids is a main advantage of positive displacement expanders as this increases the cycle efficiency and decreases the cost of the heat exchanger. This makes such expanders suitable choices for waste heat recovery systems integrating ORC systems with wet fluids. In terms of reported power, the screw expander is superior, which makes such expanders suitable candidates for small and medium scale applications. Scroll, piston, and vane expanders can be applied in small or micro scale applications due to the lower values of power outputs. In terms of efficiency, scroll and rotary vane expanders are the highest, while the piston presents the lowest values. However, piston expanders can operate at much higher pressure ratios than the others. In positive displacement expanders, lubrication is compulsory due to sealing purposes or the contact between their parts. Hence, an oil separator should be installed with reciprocating expanders, thus increasing the system complexity [10]. However, oil-free expanders exist, but they suffer from high leakages. In addition, reciprocating expanders, particularly the reciprocating piston expanders, are usually bulkier and heavier [162].

### 2.2. Turbo-Expanders

The operation of Turbo-expanders, or turbines, is based on a high-pressure working fluid being directed through the turbine blades, causing them to rotate as the fluid expands. As stated, the main difference between ORC and the steam cycle is the working fluid used in the cycle. In steam cycles, the enthalpy drop is much higher than that in ORCs. Thus, fewer turbine stages are required in ORCs [106]. There are two main types of turbo-expanders, namely: Axial turbine and radial turbines. There is not yet a standard approach for the selection and design of ORC turbines. Moreover, the choice of turbines depends on the application, though it is not always clear that any one type is superior [163]. Figure 9 presents three dimensional (3D) geometries of axial and radial machines.

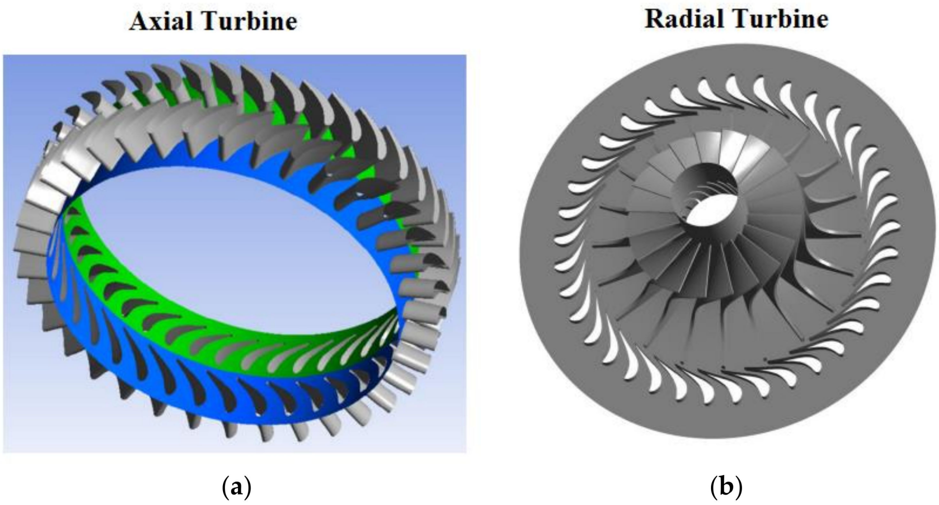



**Figure 9.** 3D geometries of (**a**) axial turbine [164], and (**b**) radial turbine [165].

The main difference between axial and radial turbines is the way the working fluid flows in relation to the shaft. In axial turbines, the flow of the working fluid is parallel to the shaft, whereas it is radial to the shaft in radial turbines. In general, axial flow turbines are used in almost all applications of gas turbine power plants, while radial turbines are used in turbochargers for commercial (diesel) engines and fire pumps [163]. Large size axial flow turbines are typically more efficient than radial flow turbines because the turning of the flow in the meridional plane is eliminated [166]. The turbines are also implemented as impulse or reaction turbines. Reaction turbines are the machines where the degree of reaction is >50% (pressure drop across turbine blades as well nozzle vanes). Inward flow radial machines are often implemented as reaction turbines and for very small scale ORC turbines often implemented as impulse turbines. Generally, axial turbines are commonly used with high flow rates and low-pressure ratios. The reason for this is that, at low mass flow, the blades of an axial turbine become very small, making it difficult to maintain small tip clearance, resulting in a significant drop in efficiency [167]. Radial turbines, on the other hand, are used with high pressure ratios and low mass flow rates, which make them practical when used in ORCs. However, there is one distinct advantage of axial turbines, which is the possibility of being air cooled when operating under high temperatures [168]. Nevertheless, cooling is not an issue in ORCs as they are always preferred when low to medium heat sources are used. In terms of the manufacturing cost, it is less expensive for radial turbines as they could be derived from standard production and are not sensitive to blade profiles [109]. Beside the above reasons, radial turbines are preferred over the axial ones because their geometries allow higher peripheral speeds than those in the axial turbines, and therefore a higher enthalpy drop per stage [106]. In addition, it was proved in a previous study [169,170] that isentropic efficiencies of radial inflow turbines can be maintained at high values, which is beneficial for ORC performance, by controlling the generator rotational speeds. Sauret and Rowlands [171] stated that radial turbines would be preferred over axial ones due to the following reasons:

Radial turbines are less sensitive to blade profile inaccuracies than axial machines, enabling high efficiencies to be maintained as size decreases.

Radial turbines are more robust under increased blade load caused by using high-density fluids at either subcritical or supercritical conditions.

Radial-inflow turbines are easier to manufacture relative to axial machines as the blades are attached to the hub. The rotor dynamic stability of the system is also improved due to a higher stiffness.

#### 2.2.1. Brief History of Turbo-Expanders in ORC-ICEs

##### Axial Turbines

Patel and Doyle [44] applied an application of using ORC in internal combustion engines in 1976. The authors used the exhaust gas of a Mack 676 diesel engine as the heat source. For the ORC system, they used a three-stage axial turbine as the expansion machine and Fluorinol-50 (mixture) as the cycle working fluid. The authors stated that the most attractive use of ORC was in large, heavy-duty diesel trucks for distance hauling, where the engine load and speed requirements were nearly constant over a large portion of operating hours and high mileages. The authors concluded that the addition of an ORC system to a long haul diesel truck could improve the fuel economy by 15% over a typical duty cycle.

Three years later, Doyle et al. [172] conducted a one-year program on a Model 676 diesel engine equipped with a bottoming ORC system using the same turbine and working fluid as in [44]. The lab results showed that an improvement of BSFC of 10–12% was achieved and this result was verified by highway tests. The authors also stated that 3120 gallons of fuel could be saved for every 100,000 miles travelled by the truck.

In 1982, Hnat et al. [173] conducted a study to recover the wasted heat in the exhaust gas of a diesel engine using ORC with toluene as the working fluid, and the Rankine Cycle with steam as the working fluid. The authors recommended that a six-stage axial steam turbine could be used for the steam Rankine Cycle and one- or two-stage turbine could be used for the ORC. In terms of cycle efficiency, the Rankine Cycle efficiencies for the toluene and steam heat recovery systems are 24% and 19%, respectively, indicating that the organic fluid will be able to convert a greater portion of the heat absorbed to mechanical or electrical power.

In the simulation study of Seher et al. [154], the results showed that at the design point, a power of approximately 10 kW could be generated at a turbine efficiency of 66%. It was also shown that at the engine full load, the turbine delivered a power up to 16 kW. When they tested the prototype, the turbine showed lower thermal power (about 9 kW maximum). According to the authors, the major reasons for that were the different gaps and different surface roughness of the prototype compared to the model assumptions.

In 2013, Kunte and Seume [170] conducted a study on utilizing wasted heat in exhaust gases of a truck-application using ORC. For the expander selection, the authors considered four turbines, namely the Pelton turbine, Heron turbine, radial inflow turbine, and axial impulse turbine, as a preselection. The authors compared the above turbines based on their performance characteristics such as efficiency, power output, and rotational speed. Based on the investigations, they selected the single-stage axial impulse turbine because of its competitive efficiency even at a high pressure ratio and the possible compact design due to a single stage design. According to the high expansion ratio, the turbine required a supersonic blade design since the high expansion ratio would lead to a big drop of enthalpy and a strong increase in the outlet velocity. At the design point, using CFD Solver Ansys CFX, the turbine provided a power of 8.39 kW, with an efficiency of 58%. The ORC efficiency and the estimated fuel consumption were 13% and 3.4%, respectively.

In 2015, Serrano et al. [174] conducted a simulation and experimental tests on recovering the wasted heat in the exhaust gas of several diesel engines of a diesel-electric train. R245fa and fluid B (was kept confidential) were selected as the working fluids. The expansion machine was an axial turbine coupled with a generator on the same axle. The simulation results showed that the expected ORC efficiency was around 6 to 7%. The electrical power generated by the turbine was expected to be 14 to 16 kW.

Recently, Cipollone et al. [175] tested an ORC-based power unit on an IVECO NEF 67 (Industrial Vehicle Corporation, Turin, Italy) turbocharged diesel engine with R245fa as the cycle working fluid and a single stage axial turbine as the expansion machine. The results showed that as the expansion ratio increased, the turbine power and isentropic efficiency increased, with maximum values of about 83% and 6.9 kW, respectively.

Radial Turbines

In 2010, Briggs et al. [176] used an ORC system equipped with a light duty diesel engine with R245fa as the working fluid and a radial inflow turbine as the expansion machine. The experimental results demonstrated a peak thermal efficiency of 42.6% for a light duty diesel engine system. The turbine-generator was designed for a peak operating speed of 80,000 rpm and the achieved power output was 4.6 kW. The authors stated that the expansion would not bring the fluid (R245fa) to a saturated vapour state, and hence two-phase flow would not be encountered in the turbine. The achieved cycle efficiency was 12.7%.

The following year, Cogswell et al. [177] designed an ORC power generation system to recover the heat from the exhaust gas of a 60 kW diesel engine. Novec649 (dodecafluoro-2-methylpentan-3-one) was selected as the working fluid since it would be non-flammable and would have near zero Global Warming Potential (GWP). The expander was a single inflow radial turbine with a 4.6 cm rotor diameter and a pressure ratio of 12. The achieved isentropic efficiency and power output were 83% and 7.8 kW, respectively. The results also showed that the ORC system generated 5.7 kW net power.

In the same year, Teng et al. [100,178] developed and tested an ORC with ethanol as the working fluid to investigate the fuel economy benefit of recovering waste heat from a 10.8-liter heavy duty truck diesel engine. The authors used a radial inflow turbine with an impeller diameter of 53 mm that would be available in a GT25 turbocharger (Garret, USA). However, the authors removed the original turbine housing and replaced it with a special design for the working fluid application where a nozzle ring with conical shaped nozzles was added. The turbine presented an isentropic efficiency of 78%. The first study [100] demonstrated that a 5% fuel saving could be achievable using the WHR Rankine cycle. However, the experimental results in the second study [178] showed that up to a 3% improvement in engine fuel consumption could be recovered using EGR and exhaust gas heat sources.

In 2013, Wolfgang Lang et al. [179] conducted a study on utilizing ORC to recover the waste heat in the exhaust gas and EGR system of a high duty diesel engine. The working fluid and the expansion machine were siloxane and a radial inflow turbine, respectively. The turbine efficiency was calculated as 78% and it would run at 25,990 rpm. The results of the ORC system showed that the turbo-generator could provide approximately 9.6 kW of additional power to the diesel engine operating in cruising conditions (150 kW power output at 1500 rpm), without significantly altering its operation.

In the same year, Takatoshi et al. [180] conducted an experimental test on the ORC that was used to recover the heat of the engine coolant with Hydro-fluoro-ether as the working fluid. The study focused on optimising each component of the cycle. In order to optimize the turbine, the nozzle angle was set as small to increase the saturation temperature in the evaporator. Also, ball bearings were applied to a shaft, and a permanent magnet type generator was adopted. The system efficiency was improved by expanding the difference of saturation temperature with improving heat exchanger performance and using a high pressure turbine. The results showed that the improvement of fuel economy reached a value of 7.5% using the developed Rankine Cycle generating system.

In 2015, Costall et al. [181] designed a radial inflow turbine to be used in an ORC system that would recover the wasted heat of an off-high way diesel engine. The authors designed three radial turbines and selected toluene as the working fluid due to its high critical temperature, which would align with the 300 °C heat source. However, the small turbine (~20 mm diameter) led to impractical blade geometry (1.6 mm blade height). The medium turbine (62.9 mm) produced 34.1 kW with a 51.5% efficiency, while the large turbine (83.0 mm) produced 45.6 kW for the maximum isentropic efficiency (56.1%).

Rudenko et al. [182] designed two nozzeless radial inflow turbines to be used as expansion machines in a dual loop ORC system using R245fa as the working fluid. The achieved total-to-static efficiency and power output of the high pressure turbine were 85.18% and 211.4 KW, while they were 91.33% and 394.6 kW for the low pressure turbine, respectively. The results also showed that up to a 19.73% power boost for the internal combustion engine could be achieved without burning additional fuel.

A recent experimental study by Guillaume et al. [183] investigated deeply the performance of a radial turbine with two working fluids (R245fa and R1233zd), and using the exhaust of a truck engine as the heat source. However, the heat wasted by the truck through the exhaust gases was simulated using an electric oil boiler coupled to the ORC loop. R245fa showed higher electrical power from the turbine-generator with a maximum value of 2.5 kW, whereas R1233zd showed a better turbine efficiency, with a maximum value of 32%.

More recently, IFPEN and ENOGIA [184] investigated the possibility of recovering the wasted heat in the coolant of a heavy duty engine. They used NOVEC649 as the working fluid, and a radial inflow turbine as the expansion machine. The maximum measured cycle efficiency was 10%, and a fuel economy improvement of about 2–3% was achieved.

The authors [185,186] proposed a design methodology for a radial inflow turbine in ORC-ICEs. The study focused on optimizing the objective function, which was the isentropic efficiency, by optimizing two variables: Exit tip radius and inlet blade angle. As the exit tip radius increases, the tip clearance loss, mainly the radial tip clearance, decreases and therefore a higher turbine performance is achieved. Moreover, the tangentil component of the velocity decreases with an increasing exit tip radius, which results in higher specific work. Also, increasing the inlet blade angle results in a higher tangential velocity, and hence higher specific work is achieved. The turbine efficiency increased 10% more compared to the conventional design.

### 2.2.2. Performance of Turbo-Expanders

Performance of turbo-expanders is expressed in terms of isentropic efficiency, as shown in Equation (2):

$$\eta_{is} = \frac{\dot{W}_{actual}}{\dot{W}_{ideal}} \tag{2}$$

where $\dot{W}$ is the rate of work done, as in positive displacement expanders, the performance of axial and radial turbines is investigated based on rotational speeds and mass flow rates, as shown in Table 1, and off-design performance, as can be seen in Figures 10 and 11.

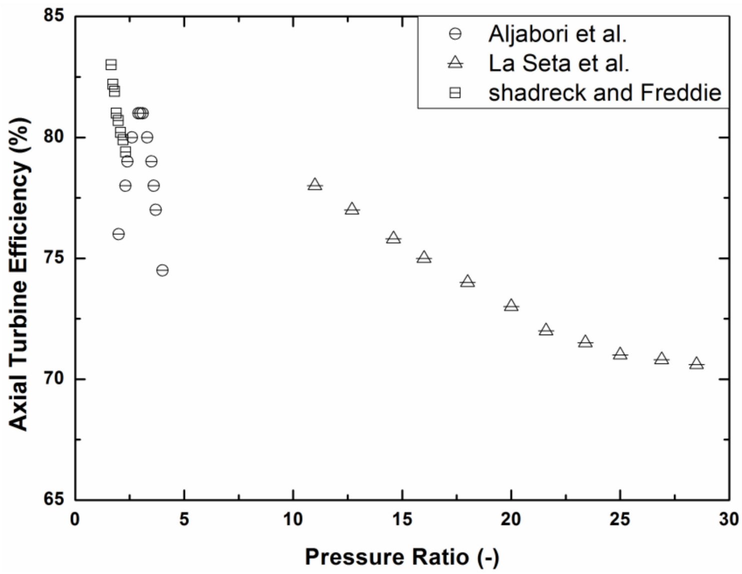



**Figure 10.** Performance of axial turbine.

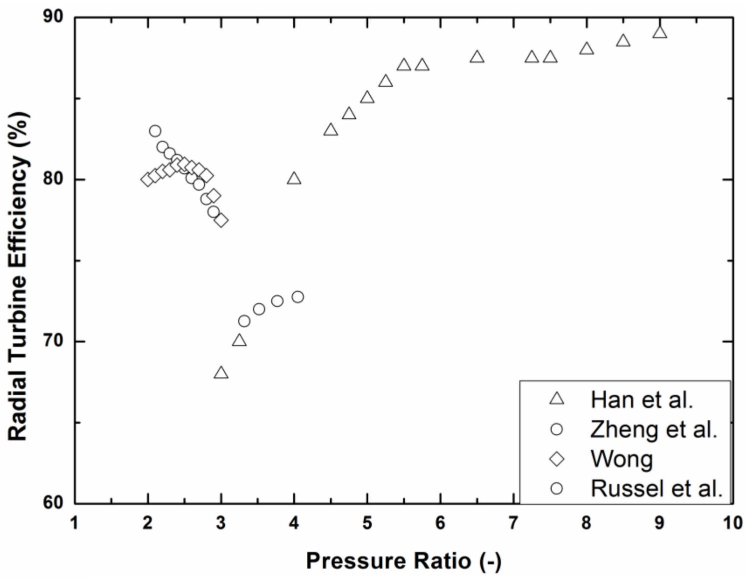



**Figure 11.** Performance of radial turbine.

In general, axial turbines are used in systems with low pressure ratios and high mass flow rates. However, exceptions to the generalization exist. As can be seen in Figure 10, which presents the results of [120,139,140], the pressure ratios are in the range of 1–5. For higher pressure ratios, a multi-stage axial turbine is required in order to achieve the required expansion with a high efficiency. However, Seta et al. [139] used a single stage axial turbine for higher pressure ratios, but the isentropic efficiency of the turbine dropped dramatically, as seen in Figure 10. Axial turbines usually present high isentropic efficiencies, as can be seen in Figure 10, which fall in the range of 75–82.5%. From Table 1, axial turbines have lower rotational speeds compared to the radial turbines, but they operate efficiently with higher mass flow rate and power conditions.

Radial inflow turbines are good candidates for systems with higher pressure ratios and low mass flow rates. As can be seen in Figure 11 that presents the results of [116–118,122], radial turbines generally show higher expansion ratios than axial turbines without getting supersonic, while their mass flow rates, presented in Table 1, are much lower. Moreover, radial turbines present high efficiency levels in off-design conditions, as can be seen in Figure 11. The isentropic efficiency can be even higher when using two stage radial turbines, as can be seen in Han et al. [116], whose isentropic efficiencies reached 89%. Moreover, the efficiency curve was almost flat at the pressure ratios 5–9. Power outputs of radial turbines are in the range of 0.54–750 kW, depending on the application, as shown in Table 1. However, radial turbines work with high rotational speeds, where a reduction gearbox might be required. Nonetheless, for micro turbines, the wheel can be directly coupled to the generator shaft.

Overall, rotational speeds of turbo-expanders are generally high, which means reduction gearboxes are required. However, micro turbines can be mounted directly to the generator shaft to eliminate the existence of gearboxes. Power outputs of turbo-expanders are higher than positive displacement expanders, which make them suitable candidates for large scale systems. One main disadvantage of turbo-expanders is that they cannot withstand two-phase fluids due to the possibility of corrosion at the turbine blades. Axial turbines present very high efficiencies for a wide range of pressure ratios. However, a multi stage configuration is required for a higher pressure ratio, which increases the weight of the turbine. On the other hand, radial inflow turbines present reasonably high efficiencies and higher power outputs. In general, the efficiency curve of the radial turbine is flatter than the axial one in the off-design performance. Compared to positive displacement expanders, the design of turbo-machines is simpler. They can also be adjusted and optimized for different operating conditions and resources, without the need of changing the overall size. Moreover, no lubrication is needed in turbo-expanders, which could spoil the working fluid, due the absence of contact seals [187].

## 3. Selection of an Optimum Expansion Machine

Expansion machines are the interface where the power production takes place. As stated, no single expansion machine is superior for all applications. However, thermodynamic operating conditions in the ORC system can be derived and expressed as dimensionless parameters that can be utilized in order to select the preferable expansion machine. In addition to the thermodynamic conditions, the operating speed of the expander has a major impact on the expander type. Moreover, size and weight should be considered in some applications such as the interior of vehicles where space is limited.

In this study, potential efficiencies and power outputs of ORC expanders based on the studied literature for different expansion machines are presented. Well-known studies in the literature such as Balje [188], Dixon and Hall [189], Badr et al. [190], and Watson and Janota [191] correlated turbine efficiency against specific speed for air turbines. Specific speed, Equation (3), is a function of operating conditions only. Therefore, it can be utilized at the early stage of the project in order to specify the optimum expander type for a certain application without considering geometrical parameters.

$$N_s = \frac{\omega\sqrt{Q}}{\Delta h_{act}^{0.75}} \tag{3}$$

Figures 12 and 13 present the results of the papers studied in Figures 5–8, Figure 10, Figure 11. Performance of expanders as a function of specific speed $N_s$ is presented, where $N_s$ is calculated by Equation (3). Since the power, speed, and flow rate in the selected studies are known, the calculation of specific speed is straightforward. Although rough, Figures 12 and 13 together can be a useful guide to the selection of the most appropriate type of expansion machine in ORC applications.

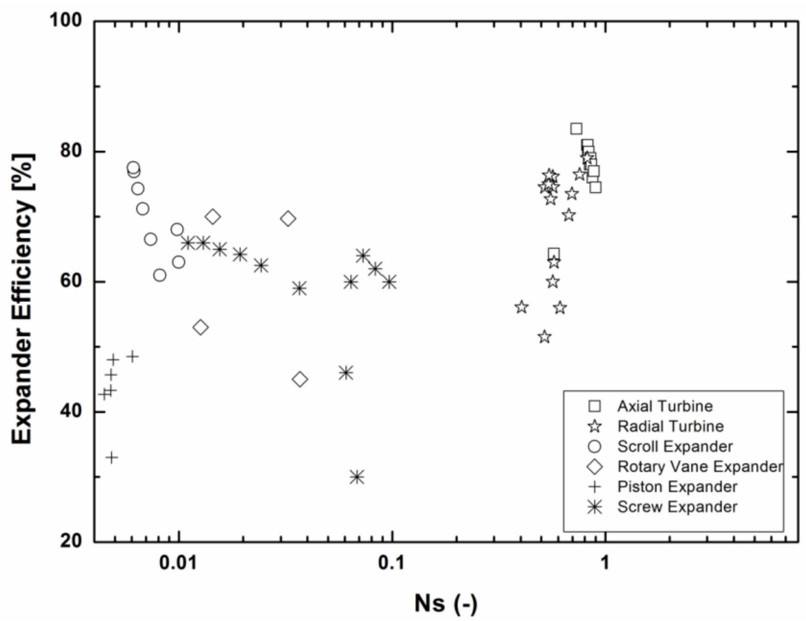



**Figure 12.** Expander efficiency vs. specific speed for different expanders.

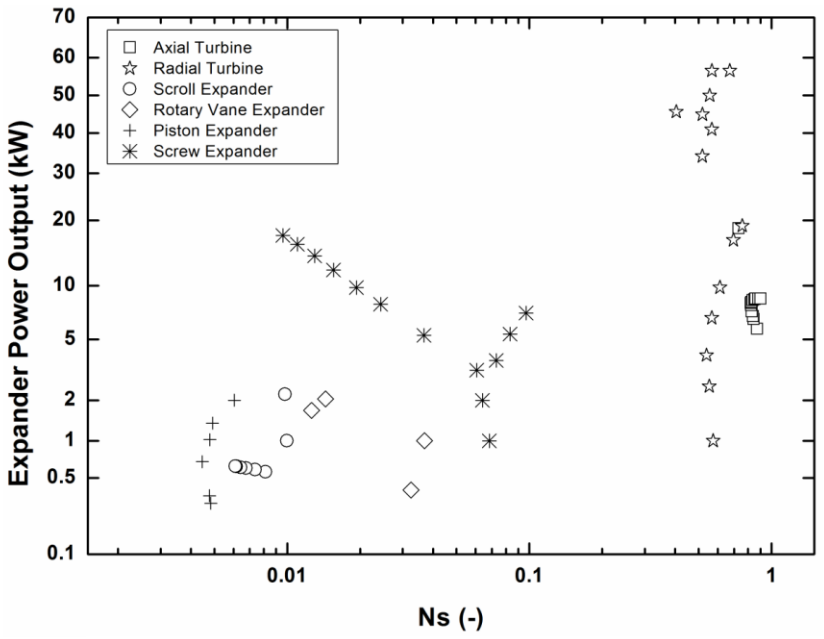



**Figure 13.** Expander power vs. specific speed for different expanders.

Generally, positive displacement expanders are preferred for low values of specific speed $(0.003 \leq N_s \leq 0.1)$. Turbomachines, on the other hand, are optimum candidates for higher values of $N_s(0.4 \leq N_s \leq 1)$. In steam Rankine cycles, the enthalpy drop is relatively higher than that in ORCs. In addition, the volume flow rate at the expander exit is lower. Therefore, the specific speed values for ORC expanders are higher than those in steam Rankine cycles based on the expression in Equation (3).

Clearly, scroll expanders present a good machine performance for all ranges of power outputs when the specific speed $N_s$ is in the range 0.006 and 0.008. Figure 13 also shows that rotary vane expanders present similar $N_s$ values for the various power outputs with optimum performance when the specific speed $N_s$ values fall in the range 0.012 and 0.037. Similarly, piston expanders are studied for applications with power outputs in the range 0.3–2 kW. The values of specific speed $N_s$ fall in the range 0.0048–0.006. Screw expanders are investigated at low to medium power applications. The power outputs fall in the range of 1–17 kW. The specific speed $N_s$ values are limited to 0.01–0.09. Overall, positive displacement expanders are preferred for low values of specific speed $N_s$. For very low values of $N_s$, scroll and piston expanders are nominated. For slightly higher values, rotary vane and screw expanders are preferable.

Turbomachines, on the other hand, present much higher values of specific speed $N_s$. Axial turbines are investigated for different power outputs that range from 5–20 kW. The specific speed $N_s$ values fall in the range of 0.7–0.9. Radial turbines are investigated for higher powers. The investigated applications produce power outputs in the range of 1–60 kW. For the whole range of power outputs, specific speed $N_s$ values fall in the range of 0.4–1.

Overall, the combination of Figures 12 and 13 can be applied as an expander's selection guide at the early stage of a project. Based on the application (required power output), the appropriate expansion machine can be nominated and its efficiency can be roughly known. Then, other constraints such as size and weight can be evaluated. For instance, positive displacement expanders are generally larger in size and heavier in weight than their turbine counterparts, which make them unfavourable, if the application is in passenger vehicles.

## 4. Cost Estimation of Expansion Machines

Expansion machines are typically the most costly component [192–196] as they need to be precision engineered [197]. It is worth mentioning that open literature lacks a direct cost comparison between the various types of expanders. Cost estimation, along with the plots in Figures 12 and 13, helps designers to decide which expander to select at the early stage of a project.

Gutiérrez-Arriaga et al. [198] presented a correlation as expressed in Equation (4) to estimate the cost of ORC expanders in terms of their power outputs. The correlation is based on an exponential scaling law and has been updated to 2014. However, using this correlation, the prices vary considerably as the power output changes which make each type of the expansion machines can have wide range of prices. Therefore, for each type, the power output is averaged and the price is estimated as depicted in Figure 14.

$$CAP_{turb} = 2237\ W_T^{0.41} \tag{4}$$

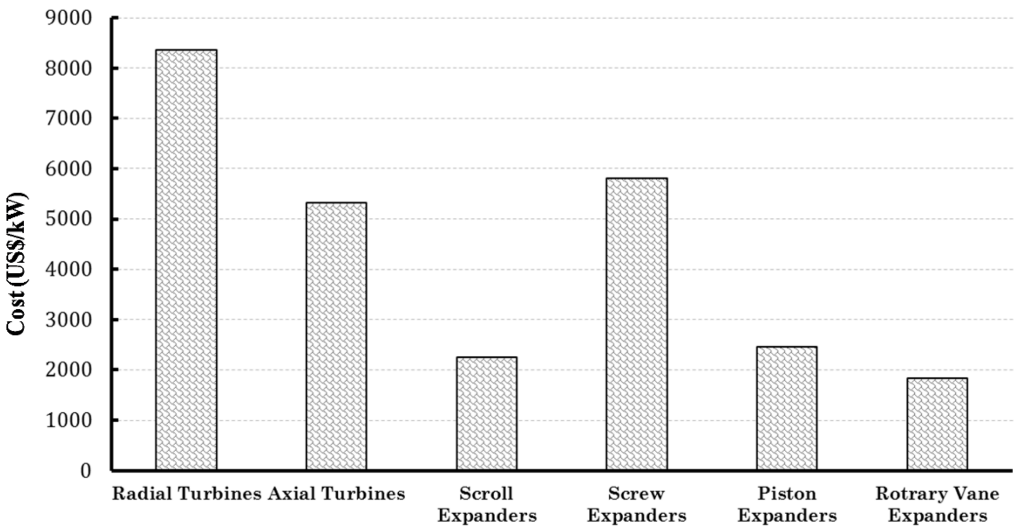



**Figure 14.** Cost estimation of expanders according to their power output only.

Clearly, radial turbines present a higher cost, which corresponds to 33% of the total cost, followed by screw expanders with 22%. Rotary vane expanders are the cheapest, with an estimated contribution of 7% compared to the total cost of the expanders. Axial turbines, piston expanders, and scroll expanders contribute to 20%, 9.4%, and 8.6%, respectively. However, the larger the system is, the more expensive the component is. The above estimations are purely based on the power outputs, regardless of the system or application.

In order to perform a more feasible comparison between the different expander technologies, it is essential to compare each type with the rest of the system's components. For each system, an expansion machine is designed in accordance with the system requirements. Therefore, the cost of each expander is compared with its own system. So, the Module Costing Technique is applied in this study. This technique is presented by Turton et al. [199], as shown in Equation (5), based on data from a survey of manufacturers during the period of May 2001 to September 2001. This correlation is widely used in the literature [200–203]. Toffolo et al. [204] utilized the aforementioned correlation to estimate the costs of a real ORC plant that was built in 2009 for the utilization of low/medium enthalpy resources in the USA. The results of their study were in a good agreement with the actual ORC equipment's cost.

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10}(X) + K_3\ [\log_{10}(X)]^2 \tag{5}$$

$C_p^0$ is the purchased equipment cost at ambient pressure and using carbon steel. $X$ is the equipment variable. For the pump and expander, it corresponds to power input and power output, respectively, in kW. For the evaporator and condenser, it corresponds to the heat exchanger area in $(\mathrm{m}^2)$. The coefficients for each component are summarized in Table 2 [199].

**Table 2.** Coefficients for Different ORC Components [199].

| Component | Purchased Cost $C_p^0$ | Coefficients | | |
|---|---|---|---|---|
| | | $K_1$ | $K_2$ | $K_3$ |
| Expander | $\log_{10}C_p^0 = K_1 + K_2\log_{10}(X) + K_3\,[\log_{10}(X)]^2$ | 2.2476 | 1.4965 | −0.1618 |
| Evaporator | | 4.6656 | −0.1557 | 0.1547 |
| Condenser | | 4.6656 | −0.1557 | 0.1547 |
| Pump | | 3.3892 | 0.0536 | 0.1538 |

The investigated works in this study are [205] (axial turbine), [206] (radial turbine), [207] (scroll expander), [160] (screw expander), [208] (piston expander), and [138] (rotary vane expander) due to the availability of the required information. Figures 15 and 16 present the distribution of the costs for each component and the contribution of each component's cost with respect to the system. Clearly, heat exchangers (evaporators and condensers) bear the brunt of the total cost, whereas pumps' cost is obviously insignificant compared to the system cost. The heat exchangers represent around $43,203 to $80,740, while pumps cost about $450 to $510. The marginal cost of evaporators and condensers is due to the different heat exchanger areas between the two components.

Last but not least, an accurate capital cost estimation should consider direct and indirect costs, such as expander's parts, installation, material, insurance, taxes, utilities, shipping, etc. Generally, Scroll expanders are popular due to their relatively low cost [98] as they can operate without inlet and exhaust valves, with a low parts count and high-volume production. They could be even cheaper if they were modified from scroll compressors. Piston expanders, on the other hand, have not become a common practice due to the need for bearings and outlet valves that increase the production cost [209]. Screw expanders are not cheap as they might require a reduction gearbox [210], and vane expanders are the cheapest as they can be designed with less weight [211], fewer mechanical parts, and without suction valves. However, lubrication requirements, technical complexities, and frictional and leakage losses are the main technical barriers in the commercialization of volumetric expanders [51]. Compared to PDEs, Turbo-expanders are in the middle regarding the cost. However, Turbo-expanders with high rotational speeds put strict requirements on the bearing, shaft seal, and strength of the rotating parts [30]. Since costs increase proportionally with the size, multi-stage turbo-expanders are even more expensive, which are not included in Figures 15 and 16.

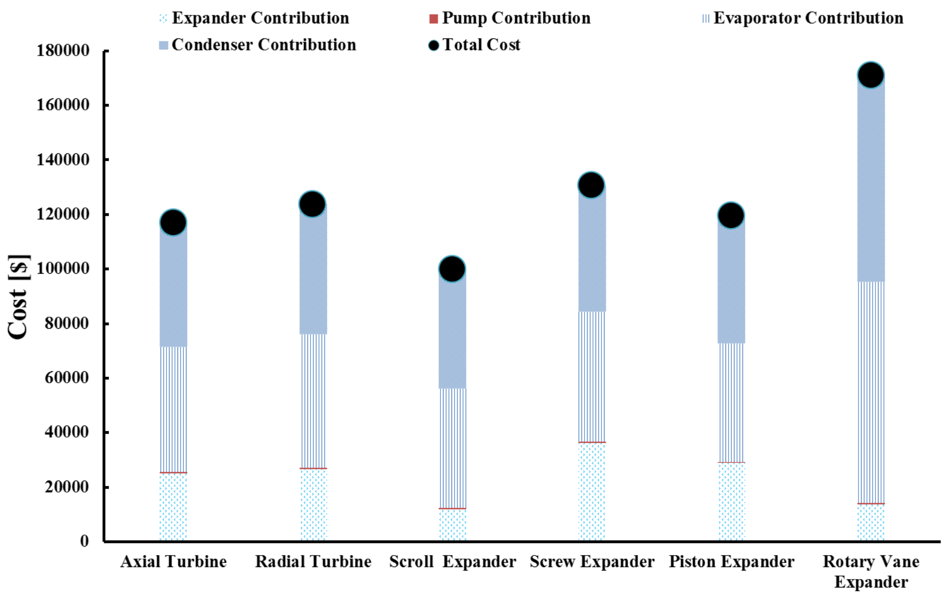



**Figure 15.** Cost estimation of different expanders based on their corresponding systems.

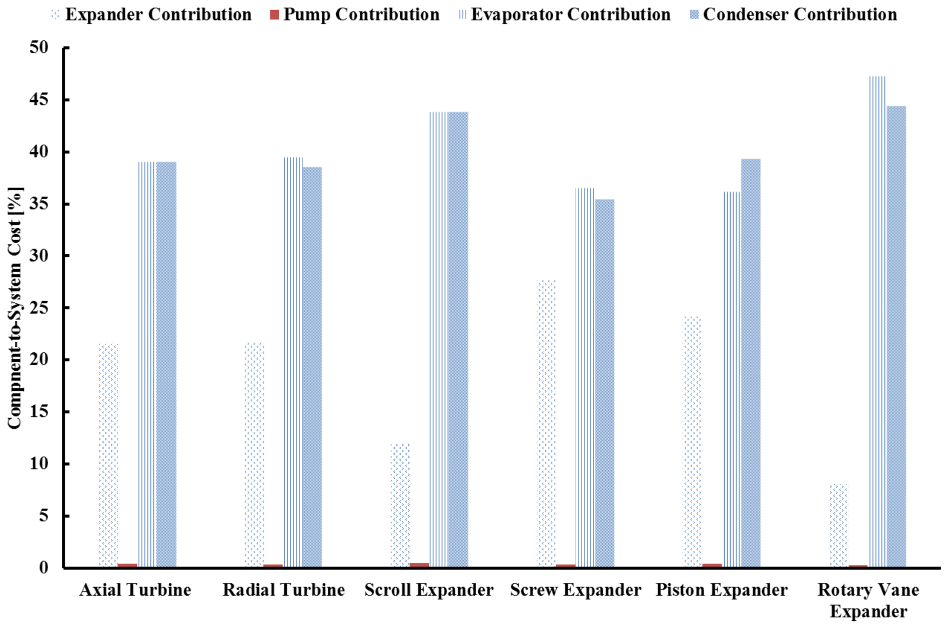



**Figure 16.** Contributions of components' costs to total system cost.

## 5. Conclusions

This paper reviewed the relevant research studies for ORCs as WHR systems in ICEs with a focus on expansion machines. Moreover, this study presented an acceptable guide that can be used in the early stage of a project in order to select the most appropriate expander type for the corresponding application. This guide consists of three plots: performance vs. specific speed, performance vs. operating characteristics, and cost estimation. The following can be concluded from this work:

Overall, the selection of the expansion machine is a trade-off between performance and cost. However, once the required power of a certain application is known, the guide can be used in order to select the optimum expander where efficiency and price can be roughly estimated.

In general, turbo-machines are more efficient when compared with positive displacement expanders, despite the size-dependent sensitivities associated with the selection of any ORC expander. Specifically, radial turbo-machines achieve the highest efficiency, as indicated by available data in the literature, but the limited availability of experimental data raises confidence concerns and requires further investigation to verify the ability of radial turbines to exceed efficiencies of 85%. Overall, the radial turbo machines were tested from the range of 1–80 kW, with efficiency values ranging from 50–85% in general.

Screw and scroll machines are well known for their ability to handling liquid phase expansion and this has been clearly confirmed in reported literature. Scroll machines performed best in the range of 1–5 kW and screw machines were a preferred option for 5–20 kW. The screw machines efficiency values were in the range of 55–70%, except for a few exceptions, while the scroll efficiency was in the range of 60–75%.

Rotary vane expanders have the smallest contribution to the overall plant cost (<10%) and screw expanders have the highest contribution to overall plant cost (>25%); even higher than that of turbo-expanders, which is around 22% of the overall plant cost.

It can be concluded that most of the selection criteria available in the literature place stress on the power capacity and specific speeds, but for ICE-ORC applications, the expanders may be challenged to withhold high temperatures and large pressure ratios. It is identified that piston expanders and screw machines are a promising solution for large expansion ratio systems. The inadequate amount of data available in the literature should influence future investigations to test expander tolerance to high temperatures and very large pressure ratios.

**Author Contributions:** A.P. and F.A. were involved in the conceived the work. F.A. compiled the work and contributed in writing, A.K.-K. contributed to form the structure, organization and presentation of the work, M.U. contributed in editing of paper in both writing and technical aspects and responded to revisions.

**Funding:** The authors would like to acknowledge the financial support of Innovate UK to this project (ref. TS/M012220/1).

**Conflicts of Interest:** The authors declare no conflict of interest.

## References

1. Intergovernmental Panel on Climate Change (IPCC). *Climate Change 2014: Mitigation of Climate Change. Contribution of Working Group III to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change*; Cambridge University Press: Cambridge, UK; New York, NY, USA, 2014.
2. International Council on Clean Transportation (ICCT). *European Vehicle Market Statistics*; ICCT: Washington, DC, USA, 2016.
3. Johnson, T.V. Review of Vehicular Emissions Trends. *SAE Int. J. Engines* **2015**, *8*, 1152–1167. [CrossRef]
4. Johnson, T.V. Diesel Emissions in Review. *SAE Int. J. Engines* **2011**, *4*, 143–157. [CrossRef]
5. Torrey, W.; Murray, D. *An Analysis of the Operational Costs of Trucking: A 2014 Update*; American Transportation Research Institute: Atlanta, GA, USA, 2014.
6. Marchionni, M.; Bianchi, G.; Karvountzis-Kontakiotis, A.; Pesiridis, A.; Tassou, S.A. Dynamic modeling and optimization of an ORC unit equipped with plate heat exchangers and turbomachines. *Energy Procedia* **2017**, *129*, 224–231. [CrossRef]
7. Arsie, I.; Cricchio, A.; Marano, V.; Pianese, C.; De Cesare, M.; Nesci, W. Modeling Analysis of Waste Heat Recovery via Thermo Electric Generators for Fuel Economy Improvement and $CO_2$ Reduction in Small Diesel Engines. *SAE Int. J. Passeng. Cars Electron. Electr. Syst.* **2014**, *7*, 246–255. [CrossRef]
8. Sprouse, C.; Depcik, C. Review of organic Rankine cycles for internal combustion engine exhaust waste heat recovery. *Appl. Therm. Eng.* **2013**, *51*, 711–722. [CrossRef]
9. Tian, H.; Shu, G.; Wei, H.; Liang, X.; Liu, L. Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE). *Energy* **2012**, *47*, 125–136. [CrossRef]
10. Wang, T.; Zhang, Y.; Peng, Z.; Shu, G. A review of researches on thermal exhaust heat recovery with Rankine cycle. *Renew. Sustain. Energy Rev.* **2011**, *15*, 2862–2871. [CrossRef]
11. Agudelo, A.F.; García-Contreras, R.; Agudelo, J.R.; Armas, O. Potential for exhaust gas energy recovery in a diesel passenger car under European driving cycle. *Appl. Energy* **2016**, *174*, 201–212. [CrossRef]
12. Kim, S.; Park, S.; Kim, S.; Rhi, S.-H. A Thermoelectric Generator Using Engine Coolant for Light-Duty Internal Combustion Engine-Powered Vehicles. *J. Electron. Mater.* **2011**, *40*, 812. [CrossRef]
13. Fu, J.; Liu, J.; Feng, R.; Yang, Y.; Wang, L.; Wang, Y. Energy and exergy analysis on gasoline engine based on mapping characteristics experiment. *Appl. Energy* **2013**, *102*, 622–630. [CrossRef]
14. U.S. Department of Energy. *Waste Heat Recovery: Technology and Opportunities in U.S. Industry*; U.S. Department of Energy Industrial Technologies Program: Washington, DC, USA, 2008.
15. Dolz, V.; Novella, R.; García, A.; Sánchez, J. HD Diesel engine equipped with a bottoming Rankine cycle as a waste heat recovery system. Part 1: Study and analysis of the waste heat energy. *Appl. Therm. Eng.* **2012**, *36*, 269–278. [CrossRef]
16. Nadaf, S.L.; Gangavati, P.B. A review on Waste Heat Recovery and Utilization from Diesel Engines. *Int. J. Adv. Eng. Technol.* **2014**, *31*, 39–45.
17. Talbi, M.; Agnew, B. Energy recovery from diesel engine exhaust gases for performance enhancement and air conditioning. *Appl. Therm. Eng.* **2002**, *22*, 693–702. [CrossRef]
18. Teng, H.; Regner, G. Improving Fuel Economy for HD Diesel Engines with WHR Rankine Cycle Driven by EGR Cooler Heat Rejection. In Proceedings of the SAE 2009 Commercial Vehicle Engineering Congress & Exhibition, Detroit, MI, USA, 20–23 April 2009; SAE International: Warrendale, PA, USA, 2009.
19. Teng, H.; Regner, G.; Cowland, C. Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part I: Hybrid Energy System of Diesel and Rankine Engines. In Proceedings of the SAE World Congress & Exhibition, Detroit, MI, USA, 16–19 April 2007; SAE International: Warrendale, PA, USA, 2007.

20. Espinosa, N.; Tilman, L.; Lemort, V.; Quoilin, S.; Lombard, B. Rankine Cycle for Waste Heat Recovery on Commercial Trucks: Approach, Constraints and Modelling. In Proceedings of the Diesel International Conference and Exhibition, Rouen, France, 26–27 May 2010.
21. Boretti, A. Recovery of exhaust and coolant heat with R245fa organic Rankine cycles in a hybrid passenger car with a naturally aspirated gasoline engine. *Appl. Therm. Eng.* **2012**, *36*, 73–77. [CrossRef]
22. Wang, T.; Zhang, Y.; Zhang, J.; Shu, G.; Peng, Z. Analysis of recoverable exhaust energy from a light-duty gasoline engine. *Appl. Therm. Eng.* **2013**, *53*, 414–419. [CrossRef]
23. Cipollone, R.; Bianchi, G.; Gualtieri, A.; Di Battista, D.; Mauriello, M.; Fatigati, F. Development of an Organic Rankine Cycle system for exhaust energy recovery in internal combustion engines. *J. Phys. Conf. Ser.* **2015**, *655*, 012015. [CrossRef]
24. Yang, K.; Zhang, H.; Song, S.; Zhang, J.; Wu, Y.; Zhang, Y.; Wang, H.; Chang, Y.; Bei, C. Performance Analysis of the Vehicle Diesel Engine-ORC Combined System Based on a Screw Expander. *Energies* **2014**, *7*, 3400–3419. [CrossRef]
25. Katsanos, C.O.; Hountalas, D.T.; Pariotis, E.G. Thermodynamic analysis of a Rankine cycle applied on a diesel truck engine using steam and organic medium. *Energy Convers. Manag.* **2012**, *60*, 68–76. [CrossRef]
26. Wang, E.H.; Zhang, H.G.; Zhao, Y.; Fan, B.Y.; Wu, Y.T.; Mu, Q.H. Performance analysis of a novel system combining a dual loop organic Rankine cycle (ORC) with a gasoline engine. *Energy* **2012**, *43*, 385–395. [CrossRef]
27. Galindo, J.; Ruiz, S.; Dolz, V.; Royo-Pascual, L.; Haller, R.; Nicolas, B.; Glavatskaya, Y. Experimental and thermodynamic analysis of a bottoming Organic Rankine Cycle (ORC) of gasoline engine using swash-plate expander. *Energy Convers. Manag.* **2015**, *103*, 519–532. [CrossRef]
28. Panesar, A.S. Waste Heat Recovery Using Fluid Bottoming Cycles for Heavy Duty Diesel Engines. Ph.D. Thesis, University of Brighton, Brighton, UK, 2015.
29. Song, P.; Wei, M.; Shi, L.; Danish, S.N.; Ma, C. A review of scroll expanders for organic Rankine cycle systems. *Appl. Therm. Eng.* **2014**, *75*, 54–64. [CrossRef]
30. Bao, J.; Zhao, L. A review of working fluid and expander selections for organic Rankine cycle. *Renew. Sustain. Energy Rev.* **2013**, *24*, 325–342. [CrossRef]
31. Saidur, R.; Rezaei, M.; Muzammil, W.K.; Hassan, M.H.; Paria, S.; Hasanuzzaman, M. Technologies to recover exhaust heat from internal combustion engines. *Renew. Sustain. Energy Rev.* **2012**, *16*, 5649–5659. [CrossRef]
32. Karvonen, M.; Kapoor, R.; Uusitalo, A.; Ojanen, V. Technology competition in the internal combustion engine waste heat recovery: A patent landscape analysis. *J. Clean. Prod.* **2016**, *112*, 3735–3743. [CrossRef]
33. Weerasinghe, W.M.S.R.; Stobart, R.K.; Hounsham, S.M. Thermal efficiency improvement in high output diesel engines a comparison of a Rankine cycle with turbo-compounding. *Appl. Therm. Eng.* **2010**, *30*, 2253–2256. [CrossRef]
34. Galindo, J.; Guardiola, C.; Dolz, V.; Kleut, P. Further analysis of a compression-expansion machine for a Brayton Waste Heat Recovery cycle on an IC engine. *Appl. Therm. Eng.* **2018**, *128*, 345–356. [CrossRef]
35. Champier, D. Thermoelectric generators: A review of applications. *Energy Convers. Manag.* **2017**, *140*, 167–181. [CrossRef]
36. Aghaali, H.; Ångström, H.-E. A review of turbocompounding as a waste heat recovery system for internal combustion engines. *Renew. Sustain. Energy Rev.* **2015**, *49*, 813–824. [CrossRef]
37. Quoilin, S.; Lemort, V.; Lebrun, J. Experimental study and modeling of an Organic Rankine Cycle using scroll expander. *Appl. Energy* **2010**, *87*, 1260–1268. [CrossRef]
38. Schuster, A.; Karellas, S.; Kakaras, E.; Spliethoff, H. Energetic and economic investigation of Organic Rankine Cycle applications. *Appl. Therm. Eng.* **2009**, *29*, 1809–1817. [CrossRef]
39. Yunus, A.C.; Boles, M.A. *Thermodynamics: An Engineering Approach, Section: 10.5*, 5th ed.; McGraw-Hill Education: New York, NY, USA, 2006.
40. Brandsar, J. Offshore Rankine Cycles. Master's Thesis, Norwegian University of Science and Technology, Trondheim, Norway, 2012.
41. Still, W.J. International Combustion Engine. England Patent 1,230,617, 19 June 1917.
42. Legros, A.; Guillaume, L.; Lemort, V.; Diny, M.; Bell, I.; Quoilin, S. Investigation on a scroll expander for waste heat recovery on internal combustion engines. In Proceedings of the International Conference on Compressors and Their Systems, London, UK, 9–10 September 2013; Volume 1, p. 10.
43. Reay, D.A. *Heat Recovery Systems: A Directory of Equipment and Techniques*; E. & F. N. Spon: London, UK, 1979.

44. Patel, P.; Doyle, E. *Compounding the Truck Diesel Engine with an Organic Rankine-Cycle System*; SAE Technical Paper 760343; SAE International: Warrendale, PA, USA, 1976.
45. Erba, M.; Biyikoglu, A. Design of Low Temperature Organic Rankine Cycle and Turbine. In Proceedings of the 4th International Conference on Power Engineering, Energy and Electrical Drives, Istanbul, Turkey, 13–17 May 2013; pp. 1065–1070. [CrossRef]
46. Endo, T.; Kawajiri, S.; Kojima, Y.; Takahashi, K.; Baba, T.; Ibaraki, S.; Takahashi, T.; Shinohara, M. *Study on Maximizing Exergy in Automotive Engines*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2007.
47. Damon, K.; Koeberlein, D. Technology and System Level Demonstration of Highly Efficient and Clean, Diesel Powered Class 8 Trucks. 2013; pp. 1–21. Available online: https://energy.gov (accessed on 18 July 2018).
48. Quoilin, S. Experimental Study and Modeling of a Low Temperature Rankine Cycle for Small Scale Cogeneration. Ph.D. Thesis, University of Liege, Liege, Belgium, 2007.
49. Ringler, J.; Seifert, M.; Guyotot, V.; Hübner, W. Rankine cycle for waste heat recovery of IC engines. *SAE Int. J. Engines* **2009**, *2*, 67–76. [CrossRef]
50. Karvountzis-Kontakiotis, A.; Pesiridis, A.; Zhao, H.; Alshammari, F.; Franchetti, B.; Pesmazoglou, I.; Tocci, L. *Effect of an ORC Waste Heat Recovery System on Diesel Engine Fuel Economy for Off-Highway Vehicles*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2017.
51. Imran, M.; Usman, M.; Park, B.-S.; Lee, D.-H. Volumetric expanders for low grade heat and waste heat recovery applications. *Renew. Sustain. Energy Rev.* **2016**, *57*, 1090–1109. [CrossRef]
52. Oh, Y.; Park, J.; Lee, J.; Do Eom, M.; Park, S. Modeling effects of vehicle specifications on fuel economy based on engine fuel consumption map and vehicle dynamics. *Transp. Res. Part D Transp. Environ.* **2014**, *32*, 287–302. [CrossRef]
53. Di Battista, D.; Mauriello, M.; Cipollone, R. Waste heat recovery of an ORC-based power unit in a turbocharged diesel engine propelling a light duty vehicle. *Appl. Energy* **2015**, *152*, 109–120. [CrossRef]
54. Horst, T.A.; Tegethoff, W.; Eilts, P.; Koehler, J. Prediction of dynamic Rankine Cycle waste heat recovery performance and fuel saving potential in passenger car applications considering interactions with vehicles' energy management. *Energy Convers. Manag.* **2014**, *78*, 438–451. [CrossRef]
55. Boretti, A. Stoichiometric H2ICE with water injection and exhaust and coolant heat recovery through organic Rankine cycles. *Int. J. Hydrogen Energy* **2011**, *36*, 12591–12600. [CrossRef]
56. Boretti, A.A. Energy Recovery in Passenger Cars. *J. Energy Resour. Technol.* **2012**, *134*, 022203. [CrossRef]
57. Franchetti, B.; Pesiridis, A.; Pesmazoglou, I.; Sciubba, E.; Tocci, L. Thermodynamic and technical criteria for the optimal selection of the working fluid in a mini-ORC. In Proceedings of the ECOS 2016—The 29th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, Postoroz, Slovenia, 19–23 June 2016.
58. Roy, J.P.; Mishra, M.K.; Misra, A. Performance analysis of an Organic Rankine Cycle with superheating under different heat source temperature conditions. *Appl. Energy* **2011**, *88*, 2995–3004. [CrossRef]
59. Lee, M.J.; Tien, D.L.; Shao, C.T. Thermophysical capability of ozone-safe working fluids for an organic rankine cycle system. *Heat Recovery Syst. CHP* **1993**, *13*, 409–418. [CrossRef]
60. Rahbar, K.; Mahmoud, S.; Al-Dadah, R.K.; Moazami, N. Modelling and optimization of organic Rankine cycle based on a small-scale radial inflow turbine. *Energy Convers. Manag.* **2015**, *91*, 186–198. [CrossRef]
61. Hung, T.C.; Shai, T.Y.; Wang, S.K. A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. *Energy* **1997**, *22*, 661–667. [CrossRef]
62. Quoilin, S. Sustainable Energy Conversion Through the Use of Organic Rankine Cycles for Waste Heat Recovery and Solar Applications. Ph.D. Thesis, University of Liege, Liege, Belgium, 2011.
63. Borsukiewiczgozdur, A.; Nowak, W. Comparative analysis of natural and synthetic refrigerants in application to low temperature Clausius–Rankine cycle. *Energy* **2007**, *32*, 344–352. [CrossRef]
64. Dong, B.; Xu, G.; Li, T.; Luo, X.; Quan, Y. Parametric analysis of organic Rankine cycle based on a radial turbine for low-grade waste heat recovery. *Appl. Therm. Eng.* **2017**, *126*, 470–479. [CrossRef]
65. Nouman, J. Comparative Studies and Analyses of Working Fluids for Organic Rankine Cycles—ORC. Master's Thesis, KTH Royal Institute of Technology, Stockholm, Sweden, 2012.
66. Liu, B.T.; Chien, K.H.; Wang, C.C. Effect of working fluids on organic Rankine cycle for waste heat recovery. *Energy* **2004**, *29*, 1207–1217. [CrossRef]

67. Zhang, L.; Zhang, C.; Liu, J.; Lin, Z.; Ye, J.; Gao, M.; Shen, Y. Nominal condensing capacity and performance evaluation of evaporative condenser. *Appl. Therm. Eng.* **2016**, *107*, 79–85. [CrossRef]
68. Chen, H.; Goswami, D.Y.; Stefanakos, E.K. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. *Renew. Sustain. Energy Rev.* **2010**, *14*, 3059–3067. [CrossRef]
69. Hung, T.C.; Wang, S.K.; Kuo, C.H.; Pei, B.S.; Tsai, K.F. A study of organic working fluids on system efficiency of an ORC using low-grade energy sources. *Energy* **2010**, *35*, 1403–1411. [CrossRef]
70. Hung, T.-C. Waste heat recovery of organic Rankine cycle using dry fluids. *Energy Convers. Manag.* **2001**, *42*, 539–553. [CrossRef]
71. Saleh, B.; Koglbauer, G.; Wendland, M.; Fischer, J. Working fluids for low-temperature organic Rankine cycles. *Energy* **2007**, *32*, 1210–1221. [CrossRef]
72. Wang, E.H.; Zhang, H.G.; Fan, B.Y.; Ouyang, M.G.; Zhao, Y.; Mu, Q.H. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. *Energy* **2011**, *36*, 3406–3418. [CrossRef]
73. Shu, G.; Li, X.; Tian, H.; Liang, X.; Wei, H.; Wang, X. Alkanes as working fluids for high-temperature exhaust heat recovery of diesel engine using organic Rankine cycle. *Appl. Energy* **2014**, *119*, 204–217. [CrossRef]
74. Javanshir, A.; Sarunac, N.; Razzaghpanah, Z. Thermodynamic analysis of a regenerative organic Rankine cycle using dry fluids. *Appl. Therm. Eng.* **2017**, *123*, 852–864. [CrossRef]
75. Dai, X.; Shi, L.; An, Q.; Qian, W. Thermal stability of some hydrofluorocarbons as supercritical ORCs working fluids. *Appl. Therm. Eng.* **2018**, *128*, 1095–1101. [CrossRef]
76. Invernizzi, C.M.; Iora, P.; Manzolini, G.; Lasala, S. Thermal stability of n-pentane, cyclo-pentane and toluene as working fluids in organic Rankine engines. *Appl. Therm. Eng.* **2017**, *121*, 172–179. [CrossRef]
77. Saloux, E.; Sorin, M.; Nesreddine, H.; Teyssedou, A. Reconstruction procedure of the thermodynamic cycle of organic Rankine cycles (ORC) and selection of the most appropriate working fluid. *Appl. Therm. Eng.* **2018**, *129*, 628–635. [CrossRef]
78. He, S.; Chang, H.; Zhang, X.; Shu, S.; Duan, C. Working fluid selection for an Organic Rankine Cycle utilizing high and low temperature energy of an LNG engine. *Appl. Therm. Eng.* **2015**, *90*, 579–589. [CrossRef]
79. Seyedkavoosi, S.; Javan, S.; Kota, K. Exergy-based optimization of an organic Rankine cycle (ORC) for waste heat recovery from an internal combustion engine (ICE). *Appl. Therm. Eng.* **2017**, *126*, 447–457. [CrossRef]
80. Astolfi, M. Technical options for organic Rankine cycle systems. In *Organic Rankine Cycle (ORC) Power Systems: Technologies and Applications*; Elsevier Science & Technology: New York, NY, USA, 2016; pp. 67–89.
81. Yang, K.; Zhang, H.; Wang, Z.; Zhang, J.; Yang, F.; Wang, E.; Yao, B. Study of zeotropic mixtures of ORC (organic Rankine cycle) under engine various operating conditions. *Energy* **2013**, *58*, 494–510. [CrossRef]
82. Wei, D.; Lu, X.; Lu, Z.; Gu, J. Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery. *Energy Convers. Manag.* **2007**, *48*, 1113–1119. [CrossRef]
83. Hountalas, D.; Mavropoulos, G. Potential for Improving HD Diesel Truck Engine Fuel Consumption Using Exhaust Heat Recovery Techniques. In *New Trends in Technologies: Devices, Computer, Communication and Industrial Systems*; InTech: Vienna, Austria, 2010; Volume 2, pp. 313–340.
84. Arnaud, L.; Ludovic, G.; Mouad, D.; Hamid, Z.; Vincent, L. Comparison and Impact of Waste Heat Recovery Technologies on Passenger Car Fuel Consumption in a Normalized Driving Cycle. *Energies* **2014**, *7*, 5273–5290. [CrossRef]
85. Hussain, Q.; Birgham, D. Organic Rankine cycle for light duty passenger vehicles. In Proceedings of the Directions in Engine-Efficiency and Emissions Research (DEER) Conference, Detroit, MI, USA, 3–6 October 2011.
86. Mahmoudzadeh Andwari, A.; Pesiridis, A.; Esfahanian, V.; Salavati-Zadeh, A.; Karvountzis-Kontakiotis, A.; Muralidharan, V. A Comparative Study of the Effect of Turbocompounding and ORC Waste Heat Recovery Systems on the Performance of a Turbocharged Heavy-Duty Diesel Engine. *Energies* **2017**, *10*, 1087. [CrossRef]
87. Bin Mamat, A.M.I.; bin Wan Mohamed, W.A.N. Thermal Analysis of Heat Recovery Unit to Recover Exhaust Energy for Using in Organic Rankine Cycle. *Appl. Mech. Mater.* **2013**, *393*, 781–786. [CrossRef]
88. Pesiridis, A. *Automotive Exhaust Emissions and Energy Recovery*; NOVA: Hauppauge, NY, USA, 2014.
89. Larjola, J. Electricity from industrial waste heat using high-speed organic Rankine cycle (ORC). *Int. J. Prod. Econ.* **1995**, *41*, 227–235. [CrossRef]
90. Crane, D.T.; Jackson, G.S. Optimization of cross flow heat exchangers for thermoelectric waste heat recovery. *Energy Convers. Manag.* **2004**, *45*, 1565–1582. [CrossRef]

91. Hsu, C.-T.; Huang, G.-Y.; Chu, H.-S.; Yu, B.; Yao, D.-J. Experiments and simulations on low-temperature waste heat harvesting system by thermoelectric power generators. *Appl. Energy* **2011**, *88*, 1291–1297. [CrossRef]
92. Hatami, M.; Ganji, D.D.; Gorji-Bandpy, M. A review of different heat exchangers designs for increasing the diesel exhaust waste heat recovery. *Renew. Sustain. Energy Rev.* **2014**, *37*, 168–181. [CrossRef]
93. Zhang, X.; Chau, K.T. An automotive thermoelectric–photovoltaic hybrid energy system using maximum power point tracking. *Energy Convers. Manag.* **2011**, *52*, 641–647. [CrossRef]
94. Yu, C.; Chau, K.T. Thermoelectric automotive waste heat energy recovery using maximum power point tracking. *Energy Convers. Manag.* **2009**, *50*, 1506–1512. [CrossRef]
95. Mamat, B.; Ihsan, A.M. Design and Development of a High Performance LPT for Electric Turbocompounding Energy Recovery Unit in a Heavily Downsized Engine. Ph.D. Thesis, Department of Mechanical Engineering, Imperial College London, London, UK, 2012.
96. Liu, W.; Jie, Q.; Kim, H.S.; Ren, Z. Current progress and future challenges in thermoelectric power generation: From materials to devices. *Acta Mater.* **2015**, *87*, 357–376. [CrossRef]
97. Liang, X.; Wang, X.; Shu, G.; Wei, H.; Tian, H.; Wang, X. A review and selection of engine waste heat recovery technologies using analytic hierarchy process and grey relational analysis. *Int. J. Energy Res.* **2015**, *39*, 453–471. [CrossRef]
98. Qiu, G.; Liu, H.; Riffat, S. Expanders for micro-CHP systems with organic Rankine cycle. *Appl. Therm. Eng.* **2011**, *31*, 3301–3307. [CrossRef]
99. Capata, R.; Hernandez, G. Proposal Design Procedure and Preliminary Simulation of Turbo Expander for Small Size (2–10 kW) Organic Rankine Cycle (ORC). In Proceedings of the ASME 2014 International Mechanical Engineering Congress and Exposition, Montreal, QC, Canada, 14–20 November 2014; Volume 6.
100. Teng, H.; Klaver, J.; Park, T.; Hunter, G.L.; van der Velde, B. *A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines—WHR System Development*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2011; Volume 1, pp. 1–11.
101. Garg, P.; Karthik, G.M.; Kumar, P. Development of a generic tool to design scroll expanders for {ORC} applications. *Appl. Therm. Eng.* **2016**, *109*, 878–888. [CrossRef]
102. Papes, I.; Degroote, J.; Vierendeels, J. Development of a thermodynamic low order model for a twin screw expander with emphasis on pulsations in the inlet pipe. *Appl. Therm. Eng.* **2016**, *103*, 909–919. [CrossRef]
103. Zhao, L.; Li, M.; Ma, Y.; Liu, Z.; Zhang, Z. Simulation analysis of a two-rolling piston expander replacing a throttling valve in a refrigeration and heat pump system. *Appl. Therm. Eng.* **2014**, *66*, 383–394. [CrossRef]
104. Yang, B.; Peng, X.; He, Z.; Guo, B.; Xing, Z. Experimental investigation on the internal working process of a CO2 rotary vane expander. *Appl. Therm. Eng.* **2009**, *29*, 2289–2296. [CrossRef]
105. Austin, D. Reid Low Temperature Power Generation Using HFE-7000 in a Rankine Cycle. Master's Thesis, San Diego State University, San Diego, CA, USA, 2012.
106. Quoilin, S.; Van Den Broek, M.; Declaye, S.; Dewallef, P.; Lemort, V. Techno-economic survey of organic rankine cycle (ORC) systems. *Renew. Sustain. Energy Rev.* **2013**, *22*, 168–186. [CrossRef]
107. Zhao, L.; Li, M.; Liu, Z. Simulation analysis of a two rolling piston expander replacing throttling valve in conventional refrigerant heat pump system. In Proceedings of the International Compressor Engineering Conference, West Lafayette, IN, USA, 16–19 July 2012. Paper 2100.
108. Badami, M.; Mura, M. Preliminary design and controlling strategies of a small-scale wood waste Rankine Cycle (RC) with a reciprocating steam engine (SE). *Energy* **2009**, *34*, 1315–1324. [CrossRef]
109. Paltrinieri, N. A Mean-Line Model to Predict the Deaign Performance of Radial Inflow Turbines in Organic Rankine Cycles. Master's Thesis, University of Padua, Padua, Italy, 2014.
110. Badr, O.; Naik, S.; O'Callaghan, P.W.; Probert, S.D. Expansion machine for a low power-output steam Rankine-cycle engine. *Appl. Energy* **1991**, *39*, 93–116. [CrossRef]
111. Badr, O.; Naik, S.; O'Callaghan, P.W.; Probert, S.D. Rotary Wankel engines as expansion devices in steam Rankine-cycle engines. *Appl. Energy* **1991**, *39*, 59–76. [CrossRef]
112. Antonelli, M.; Baccioli, A.; Francesconi, M.; Desideri, U.; Martorano, L. Operating maps of a rotary engine used as an expander for micro-generation with various working fluids. *Appl. Energy* **2014**, *113*, 742–750. [CrossRef]
113. Badr, O.; O'Callaghan, P.W.; Hussein, M.; Probert, S.D. Multi-vane expanders as prime movers for low-grade energy organic Rankine-cycle engines. *Appl. Energy* **1984**, *16*, 129–146. [CrossRef]

114. Grip, R.L. A Mechanical Model of an Axial Piston Machine. Ph.D. Thesis, KTH Industrial Engineering and Management, Stockholm, Sweden, 2009.
115. Wenzhi, G.; Junmeng, Z.; Guanghua, L.; Qiang, B.; Liming, F. Performance evaluation and experiment system for waste heat recovery of diesel engine. *Energy* **2013**, *55*, 226–235. [CrossRef]
116. Han, S.; Seo, J.; Choi, B.-S. Development of a 200 kW ORC radial turbine for waste heat recovery. *J. Mech. Sci. Technol.* **2014**, *28*, 5231–5241. [CrossRef]
117. Zheng, Y.; Hu, D.; Cao, Y.; Dai, Y. Preliminary design and off-design performance analysis of an Organic Rankine Cycle radial-inflow turbine based on mathematic method and CFD method. *Appl. Therm. Eng.* **2017**, *112*, 25–37. [CrossRef]
118. Wong, C.S. Design Process of Low Temperature Organic Rankine Cycle (LT-ORC). Ph.D. Thesis, University of Canterbury, Christchurch, New Zealand, 2015.
119. Nithesh, K.G.; Chatterjee, D. Numerical prediction of the performance of radial inflow turbine designed for ocean thermal energy conversion system. *Appl. Energy* **2016**, *167*, 1–16. [CrossRef]
120. Al Jubori, A.; Daabo, A.; Al-Dadah, R.K.; Mahmoud, S.; Ennil, A.B. Development of micro-scale axial and radial turbines for low-temperature heat source driven organic Rankine cycle. *Energy Convers. Manag.* **2016**, *130*, 141–155. [CrossRef]
121. Yamamoto, T.; Furuhata, T.; Arai, N.; Mori, K. Design and testing of the Organic Rankine Cycle. *Energy* **2001**, *26*, 239–251. [CrossRef]
122. Russel, H.; Rowlands, A.; Ventura, C.; Jahn, I. Design and Testing Process for a 7 kW Radial Inflow Refrigerant Turbine at the University of Queensland. In Proceedings of the ASME Turbo Expo 2016: Turbomachinery Technical Conference and Exposition, Seoul, Korea, 13–17 June 2016; p. V008T23A036.
123. Wang, H.; Peterson, R.B.; Herron, T. Experimental performance of a compliant scroll expander for an organic Rankine cycle. *Proc. Inst. Mech. Eng. Part A J. Power Energy* **2009**, *223*, 863–872. [CrossRef]
124. Oudkerk, J.-F.; Quoilin, S.; Declaye, S.; Guillaume, L.; Winandy, E.; Lemort, V. Evaluation of the Energy Performance of an Organic Rankine Cycle-Based Micro Combined Heat and Power System Involving a Hermetic Scroll Expander. *J. Eng. Gas Turbines Power* **2013**, *135*, 042306. [CrossRef]
125. Ayachi, F.; Ksayer, E.B.; Neveu, P.; Zoughaib, A. Experimental investigation and modeling of a hermetic scroll expander. *Appl. Energy* **2016**, *181*, 256–267. [CrossRef]
126. Papes, I.; Degroote, J.; Vierendeels, J. New insights in twin screw expander performance for small scale ORC systems from 3D CFD analysis. *Appl. Therm. Eng.* **2015**, *91*, 535–546. [CrossRef]
127. Zhang, Y. Experimental Study on the Performance of Single Screw Expander with 195 mm Diameter Screw. In Proceedings of the 2nd International Seminar on ORC Power Systems, Rotterdam, The Netherlands, 7–8 October 2013.
128. Tang, H.; Wu, H.; Wang, X.; Xing, Z. Performance study of a twin-screw expander used in a geothermal organic Rankine cycle power generator. *Energy* **2015**, *90*, 631–642. [CrossRef]
129. Wang, W.; Wu, Y.; Ma, C.; Liu, L.; Yu, J. Preliminary experimental study of single screw expander prototype. *Appl. Therm. Eng.* **2011**, *31*, 3684–3688. [CrossRef]
130. Ziviani, D.; Gusev, S.; Lecompte, S.; Groll, E.A.; Braun, J.E.; Horton, W.T.; van den Broek, M.; De Paepe, M. Characterizing the performance of a single-screw expander in a small-scale organic Rankine cycle for waste heat recovery. *Appl. Energy* **2016**, *181*, 155–170. [CrossRef]
131. Oudkerk, J.; Dickes, R.; Dumont, O.; Lemort, V. Experimental performance of a piston expander in a smallscale organic Rankine cycle. In Proceedings of the 9th International Conference on Compressors and Their Systems, London, UK, 5–9 September 2015.
132. Zheng, N.; Zhao, L.; Wang, X.D.; Tan, Y.T. Experimental verification of a rolling-piston expander that applied for low-temperature Organic Rankine Cycle. *Appl. Energy* **2013**, *112*, 1265–1274. [CrossRef]
133. Lemorta, V.; Guillaumea, L.; Legrosa, A.; Declayea, S.; Quoilina, S. A Comparison of Piston, Screw and Scroll Expanders for Small-Scale Rankine Cycle Systems. In Proceedings of the 3rd International Conference on Microgeneration and Related Technologies, Naples, Italy, 15–17 April 2013.
134. Han, Y.; Li, R.; Liu, Z.; Tian, J.; Wang, X.; Kang, J. Feasibility analysis and performance characteristics investigation of spatial recuperative expander based on organic Rankine cycle for waste heat recovery. *Energy Convers. Manag.* **2016**, *121*, 335–348. [CrossRef]

135. Vodicka, V.; Guillaume, L.; Mascuch, J.; Lemort, V. Testing and Modelling A Vne Expander Used in an ORC Working with HexamethyldisIloxane (MM). In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015.
136. Kolasiński, P. The influence of the heat source temperature on the multivane expander output power in an Organic Rankine Cycle (ORC) system. *Energies* **2015**, *8*, 3351–3369. [CrossRef]
137. Jia, X.; Zhang, B.; Pu, L.; Guo, B.; Peng, X. Improved rotary vane expander for trans-critical CO2 cycle by introducing high-pressure gas into the vane slots. *Int. J. Refrig.* **2011**, *34*, 732–741. [CrossRef]
138. Tahir, M.B.M.; Yamada, N.; Hoshino, T. Efficiency of Compact Organic Rankine Cycle System with Rotary-Vane-Type Expander for Low-Temperature Waste Heat Recovery. *Int. J. Environ. Sci. Eng.* **2010**, *2*, 11–16.
139. La Seta, A.; Meroni, A.; Andreasen, J.G.; Pierobon, L.; Persico, G.; Haglind, F. Combined Turbine and Cycle Optimization for Organic Rankine Cycle Power Systems—Part B: Application on a Case Study. *Energies* **2016**, *9*, 393. [CrossRef]
140. Shadreck, M.; Situmbeko, F.L. Inambao Small Scale Axial Turbine Preliminary Design and Modelling. *Int. J. Eng. Res. Technol.* **2016**, *5*, 683–690.
141. Lazzaretto, A.; Manente, G. A new criterion to optimize ORC design performance using efficiency correlations for axial and radial turbines. *Int. J. Thermodyn.* **2014**, *17*, 173–181. [CrossRef]
142. Oomori, H.; Ogino, S. *Waste Heat Recovery of Passenger Car Using a Combination of Rankine Bottoming Cycle and Evaporative Engine Cooling System*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 1993.
143. Kane, M. Small hybrid solar power system. *Energy* **2003**, *28*, 1427–1443. [CrossRef]
144. Kane, M.; Favrat, D.; Gay, B.; Andres, O. Scroll expander organic Rankine cycle (ORC) efficiency boost of biogas engines. In Proceedings of the ECOS 2007, Padova, Italy, 25–28 June 2007; pp. 1–8.
145. Mathias, J.A.; Johnston, J.R.; Cao, J.; Priedeman, D.K.; Christensen, R.N. Experimental Testing of Gerotor and Scroll Expanders Used in, and Energetic and Exergetic Modeling of, an Organic Rankine Cycle. *J. Energy Resour. Technol.* **2009**, *131*, 012201. [CrossRef]
146. Clemente, S.; Micheli, D.; Reini, M.; Taccani, R. Energy efficiency analysis of Organic Rankine Cycles with scroll expanders for cogenerative applications. *Appl. Energy* **2012**, *97*, 792–801. [CrossRef]
147. Yu, G.; Shu, G.; Tian, H.; Wei, H.; Liu, L. Simulation and thermodynamic analysis of a bottoming Organic Rankine Cycle (ORC) of diesel engine (DE). *Energy* **2013**, *51*, 281–290. [CrossRef]
148. Kim, Y.; Shin, D.; Kim, C. Optimization of Design Pressure Ratio of Positive Displacement Expander for Vehicle Engine Waste Heat Recovery. *Energies* **2014**, *7*, 6105–6117. [CrossRef]
149. Petr, P.; Schröder, C.; Kohler, J.; Graber, M. Optimal Control of Waste Heat Recovery Systems Applying Nonlinear Model Predictive Control (NMPC). In Proceedings of the 3rd Seminar on ORC Systems, Brussels, Belgium, 12–14 October 2015.
150. Leibowitz, H.; Smith, I.K.; Stosic, N. Cost Effective Small Scale ORC Systems for Power Recovery from Low Grade Heat Sources. In Proceedings of the 2006 ASME International Mechanical Engineering Congress and Exposition, Chicago, IL, USA, 5–10 November 2006; Volume 2006, pp. 521–527.
151. Wu, Y.; Lei, B.; Wang, W.; Zhang, Y. Experimental Study on Organic Rankine Cycle System with Single-Screw Expander for Waste Heat Recovery from Diesel. In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015; pp. 1–10.
152. Poulin, E.; Demler, R.; Krepchin, I.; Walker, D. *Steam Bottoming Cycle for an Adiabatic Diesel Engine*; DOE/NASA-0030-1; NASA-CR-168255 ON: DE85001318; Foster-Miller Inc.: Waltham, MA, USA, 1984.
153. Kubo, I. *Technical and Economic Study of Stirling and Rankine Cycle Bottoming Systems for Heavy Truck Diesel Engines*; DOE/NASA/0361-1; NASA-CR-180833; CTR-0723-87001; Cummins Engine Co., Inc.: Washington, DC, USA, 1987.
154. Seher, D.; Lengenfelder, T.; Gerhardt, J.; Eisenmenger, N.; Hackner, M.; Krinn, I. Waste Heat Recovery for Commercial Vehicles with a Rankine Process. In Proceedings of the 21st Aachen Colloquium Automobile and Engine Technology, Aachen, Germany, 8–10 October 2012.
155. Daccord, R.; Darmedru, A.; Melis, J. Oil-Free Axial Piston Expander for Waste Heat Recovery. In Proceedings of the SAE 2014 World Congress & Exhibition, Detroit, MI, USA, 8–10 April 2014; SAE International: Warrendale, PA, USA, 2014.
156. Chiong, M.C.; Rajoo, S.; Romagnoli, A. *Nozzle Steam Piston Expander for Engine Exhaust Energy Recovery*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2015; pp. 2–6.

157. Galindo, J.; Dolz, V.; Royo, L.; Haller, R.; Melis, J. Study of a Volumetric Expander Suitable for Waste Heat Recovery from an Automotive IC Engine Using n ORC with Ethanol. In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015; pp. 1–17.
158. Xinxin, Z.; Ke, Z.; Sujuan, B.A.I.; Ying, Z.; He, M. Exhaust Recovery of Vehicle Gasoline Engine Based on Organic Rankine Cycle. In Proceedings of the SAE 2011 World Congress & Exhibition, Detroit, MI, USA, 12–14 April 2011; SAE International: Warrendale, PA, USA, 2011.
159. Declaye, S.; Quoilin, S.; Guillaume, L.; Lemort, V. Experimental study on an open-drive scroll expander integrated into an ORC (Organic Rankine Cycle) system with R245fa as working fluid. *Energy* **2013**, *55*, 173–183. [CrossRef]
160. Zhang, Y.-Q.; Wu, Y.-T.; Xia, G.-D.; Ma, C.-F.; Ji, W.-N.; Liu, S.-W.; Yang, K.; Yang, F.-B. Development and experimental study on organic Rankine cycle system with single-screw expander for waste heat recovery from exhaust of diesel engine. *Energy* **2014**, *77*, 499–508. [CrossRef]
161. Mahmoud, A.M.; Sherif, S.A.; Lear, W.E. Frictional and Internal Leakage Losses in Rotary-Vane Two-Phase Refrigerating Expanders. *J. Energy Resour. Technol.* **2010**, *132*, 21007–21010. [CrossRef]
162. Latz, G.; Andersson, S.; Munch, K. Selecting an Expansion Machine for Vehicle Waste-Heat Recovery Systems Based on the Rankine Cycle. *SAE Int.* **2013**, *1*, 1–15. [CrossRef]
163. Gorla, R.S.; Khan, A.A. *Turbomachinery: Design and Theory*; Marcel Dekker, Inc.: New York, NY, USA, 2005; ISBN 0-8247-0980-2.
164. Khalil, K.M.; Mahmoud, S.; Al-Dadah, R.K.; Ennil, A.B. Investigate a hybrid open-Rankine cycle small-scale axial nitrogen expander by a camber line control point parameterization optimization technique. *Appl. Therm. Eng.* **2017**, *127*, 823–836. [CrossRef]
165. Dong, B.; Xu, G.; Luo, X.; Zhuang, L.; Quan, Y. Analysis of the supercritical organic Rankine cycle and the radial turbine design for high temperature applications. *Appl. Therm. Eng.* **2017**, *123*, 1523–1530. [CrossRef]
166. Patterson, D.J.; Kruiswyk, R.W. An Engine System Approach to Exhaust Waste Heat Recovery. In Proceedings of the Diesel Engine-Efficiency and Emissions Research (DEER), Detroit, MI, USA, 13–16 August 2007.
167. Saravanamuttoo, H.; Rogers, G.; Cohen, H.; Straznicky, P. *Gas Turbine Theory*, 6th ed.; Pearson Education Limited: London, UK, 2009.
168. *TMI Staff & Contributers*; Turbomachinery International: Norwalk, CT, USA, 2012.
169. Alshammari, F.; Karvountzis-Kontakiotis, A.; Pesiridis, A. Radial turbine expander design for organic rankine cycle, waste heat recovery in high efficiency, off-highway vehicles. In Proceedings of the 3rd Biennial International Conference on Powertrain Modelling and Control (PMC 2016), Loughborough, UK, 7–9 September 2016.
170. Kunte, H.; Seume, J. *Partial Admission Impulse Turbine for Automotive ORC Application*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2013; Volume 6.
171. Sauret, E.; Rowlands, A.S. Candidate radial-inflow turbines and high-density working fluids for geothermal power systems. *Energy* **2011**, *36*, 4460–4467. [CrossRef]
172. Doyle, E.; DiNanno, L.; Kramer, S. *Installation of a Diesel-Organic Rankine Compound Engine in a Class 8 Truck for a Single-Vehicle Test*; SAE Technical Paper 790646; SAE International: Warrendale, PA, USA, 1979.
173. Hnat, G.; Patten, J.S.; Bartone, L.M.; Cutting, J.C. Industrial Heat Recovery with Organic Rankine Cycles. In Proceedings of the 4th Industrial Energy Technology Conference, Houston, TX, USA, 4–7 April 1982; pp. 524–532.
174. Serrano, D.; Smague, P.; Tona, P.; Leduc, P.; Mintsa, A.C.; Leroux, A.; Chevalier, A.P. Improving Train Energy Efficiency by Organic Rankine Cycle (ORC) for Recovering Waste Heat from Exhaust Gas. In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015.
175. Cipollone, R.; Di Battista, D.; Bettoja, F. Performances of an ORC power unit for Waste Heat Recovery on Heavy Duty Engine. *Energy Procedia* **2017**, *129*, 770–777. [CrossRef]
176. Briggs, T.E.; Wagner, R.; Edwards, D.; Curran, S.; Nafziger, E. *A Waste Heat Recovery System for Light Duty Diesel Engines*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2010.
177. Cogswell, F.J.; Gerlach, D.W.; Wagner, T.C.; Mulugeta, J. Design of an Organic Rankine Cycle for Waste Heat Recovery from a Portable Diesel Generator. In Proceedings of the ASME 2011 International Mechanical Engineering Congress and Exposition, Denver, CO, USA, 11–17 November 2011; Volume 4, pp. 707–716.

178. Park, T.; Teng, H.; Hunter, G.L.; van der Velde, B.; Klaver, J. *A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines—Experimental Results*; SAE Technical Paper; SAE International: Warrendale, PA, USA, 2011; p. 1.
179. Lang, W.; Colonna, P.; Almbauer, R. Assessment of Waste Heat Recovery from a Heavy-Duty Truck Engine by Means of an ORC Turbogenerator. *J. Eng. Gas Turbines Power* **2013**, *135*, 042313. [CrossRef]
180. Furukawa, T.; Nakamura, M.; Machida, K.; Shimokawa, K. A Study of the Rankine Cycle Generating System for Heavy Duty HV Trucks. In Proceedings of the SAE 2014 World Congress & Exhibition, Detroit, MI, USA, 8–10 April 2014; SAE International: Warrendale, PA, USA, 2014.
181. Costall, A.W.; Gonzalez Hernandez, A.; Newton, P.J.; Martinez-Botas, R.F. Design methodology for radial turbo expanders in mobile organic Rankine cycle applications. *Appl. Energy* **2015**, *157*, 729–743. [CrossRef]
182. Rudenko, O.; Moroz, L.; Burlaka, M.; Joly, C. Design of Waste Heat Recovery Systems Based on Supercritical ORC for Powerful Gas and Diesel Engines. In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015; pp. 1–10.
183. Guillaume, L.; Legros, A.; Desideri, A.; Lemort, V. Performance of a radial-inflow turbine integrated in an ORC system and designed for a WHR on truck application: An experimental comparison between R245fa and R1233zd. *Appl. Energy* **2017**, *186*, 408–422. [CrossRef]
184. Leduc, P.; Smague, P.; Leroux, A.; Henry, G. Low temperature heat recovery in engine coolant for stationary and road transport applications. *Energy Procedia* **2017**, *129*, 834–842. [CrossRef]
185. Alshammari, F.; Karvountzis-Kontakiotis, A.; Pesiridis, A.; Minton, T. Radial Expander Design for an Engine Organic Rankine Cycle Waste Heat Recovery System. *Energy Procedia* **2017**, *129*, 285–292. [CrossRef]
186. Karvountzis-Kontakiotis, A.; Alshammari, F.; Pesiridis, A.; Franchetti, B.; Pesmazoglou, I.; Tocci, L. Variable Geometry Turbine Design for Off-Highway Vehicle Organic Rankine Cycle Waste Heat Recovery. In Proceedings of the THIESEL 2016 Conference on Thermo-and Fluid Dynamic Processes in Direct Injection Engines, Valencia, Spain, 13–16 September 2016.
187. Andreas, P. Weib Volumetric Expander versus Turbine—Which Is the Better Choice for Small ORC Plants. In Proceedings of the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, 12–14 October 2015; pp. 1–10.
188. Balje, O.E. *Turbomachines: A Guide to Design, Selection, and Theory*; Wiley: New York, NY, USA, 1981; ISBN 0471060364.
189. Dixon, S.L.; Hall, C. *Fluid Mechanics and Thermodynamics of Turbomachinery*; Butterworth-Heinemann: Oxford, UK, 2010.
190. Badr, O.; Probert, S.D.; O'Callaghan, P.W. Selecting a working fluid for a Rankine-cycle engine. *Appl. Energy* **1985**, *21*, 1–42. [CrossRef]
191. Watson, N.; Janota, M.S. *Turbocharging the Internal Combustion Engine*, 1st ed.; MacMilan: New York, NY, USA, 1982.
192. Pierobon, L.; Nguyen, T.-V.; Larsen, U.; Haglind, F.; Elmegaard, B. Multi-objective optimization of organic Rankine cycles for waste heat recovery: Application in an offshore platform. *Energy* **2013**, *58*, 538–549. [CrossRef]
193. Lecompte, S.; Huisseune, H.; van den Broek, M.; De Schampheleire, S.; De Paepe, M. Part load based thermo-economic optimization of the Organic Rankine Cycle (ORC) applied to a combined heat and power (CHP) system. *Appl. Energy* **2013**, *111*, 871–881. [CrossRef]
194. Quoilin, S.; Declaye, S.; Tchanche, B.F.; Lemort, V. Thermo-economic optimization of waste heat recovery Organic Rankine Cycles. *Appl. Therm. Eng.* **2011**, *31*, 2885–2893. [CrossRef]
195. Yu, G.; Shu, G.; Tian, H.; Wei, H.; Liang, X. Multi-approach evaluations of a cascade-Organic Rankine Cycle (C-ORC) system driven by diesel engine waste heat: Part B-techno-economic evaluations. *Energy Convers. Manag.* **2016**, *108*, 596–608. [CrossRef]
196. Garg, P.; Orosz, M.S.; Kumar, P. Thermo-economic evaluation of {ORCs} for various working fluids. *Appl. Therm. Eng.* **2016**, *109*, 841–853. [CrossRef]
197. Meyer, D.; Wong, C.; Engel, E.; Krumdieck, S. Design and Build of a1 Kilowatt Organic Rankine Cycle Power Generatio. In Proceedings of the 35th New Zealand Geothermal Workshop, Rotorua, New Zealand, 18–21 November 2013.

198. Gutiérrez-Arriaga, C.G.; Abdelhady, F.; Bamufleh, H.S.; Serna-González, M.; El-Halwagi, M.M.; Ponce-Ortega, J.M. Industrial waste heat recovery and cogeneration involving organic Rankine cycles. *Clean Technol. Environ. Policy* **2015**, *17*, 767–779. [CrossRef]
199. Turton, R.; Bailie, R.; Whiting, W.; Shaeiwitz, J. *Synthesis and Design of Chemical Processes*, 3rd ed.; Pearson Education, Inc.: London, UK, 2009.
200. Lemmens, S. Cost Engineering Techniques and Their Applicability for Cost Estimation of Organic Rankine Cycle Systems. *Energies* **2016**, *9*, 485. [CrossRef]
201. Walraven, D.; Laenen, B.; D'haeseleer, W. Economic system optimization of air-cooled organic Rankine cycles powered by low-temperature geothermal heat sources. *Energy* **2015**, *80*, 104–113. [CrossRef]
202. Liu, X.; Wei, M.; Yang, L.; Wang, X. Thermo-economic analysis and optimization selection of ORC system configurations for low temperature binary-cycle geothermal plant. *Appl. Therm. Eng.* **2017**, *125*, 153–164. [CrossRef]
203. Li, S.; Dai, Y. Thermo-economic comparison of Kalina and $CO_2$ transcritical power cycle for low temperature geothermal sources in China. *Appl. Therm. Eng.* **2014**, *70*, 139–152. [CrossRef]
204. Toffolo, A.; Lazzaretto, A.; Manente, G.; Paci, M. A multi-criteria approach for the optimal selection of working fluid and design parameters in Organic Rankine Cycle systems. *Appl. Energy* **2014**, *121*, 219–232. [CrossRef]
205. Shengjun, Z.; Huaixin, W.; Tao, G. Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation. *Appl. Energy* **2011**, *88*, 2740–2754. [CrossRef]
206. Pei, G.; Li, J.; Li, Y.; Wang, D.; Ji, J. Construction and dynamic test of a small-scale organic rankine cycle. *Energy* **2011**, *36*, 3215–3223. [CrossRef]
207. Jung, H.-C.; Taylor, L.; Krumdieck, S. An experimental and modelling study of a 1 kW organic Rankine cycle unit with mixture working fluid. *Energy* **2015**, *81*, 601–614. [CrossRef]
208. Li, M.; Wang, J.; He, W.; Gao, L.; Wang, B.; Ma, S.; Dai, Y. Construction and preliminary test of a low-temperature regenerative Organic Rankine Cycle (ORC) using R123. *Renew. Energy* **2013**, *57*, 216–222. [CrossRef]
209. Zywica, G.; Kaczmarczyk, T.; Ihnatowicz, E. Expanders for dispersed power generation: Maintenance and diagnostics problems. *Trans. Inst. Fluid-Flow Mach.* **2016**, *131*, 173–180.
210. Zywica, G.; Kaczmarczyk, T.Z.; Ihnatowicz, E. A review of expanders for power generation in small-scale organic Rankine cycle systems: Performance and operational aspects. *Proc. Inst. Mech. Eng. Part A J. Power Energy* **2016**, *230*, 669–684. [CrossRef]
211. Kolasinski, P.; Błasiak, P.; Rak, J. Experimental and Numerical Analyses on the Rotary Vane Expander Operating Conditions in a Micro Organic Rankine Cycle System. *Energies* **2016**, *9*, 606. [CrossRef]