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Review

Review of Organic Rankine Cycles for Internal Combustion Engine Waste Heat Recovery: Latest Decade in Review

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Abstract: The last decade (2013–2023) was the most prolific period of organic Rankine cycle (ORC) research in history in terms of both publications and citations. This article provides a detailed review of the broad and voluminous collection of recent internal combustion engine (ICE) waste heat recovery (WHR) studies, serving as a necessary follow-on to the author's 2013 review. Research efforts have targeted diverse applications (e.g., vehicular, stationary, and building-based), and it spans the full gamut of engine sizes and fuels. Furthermore, cycle configurations extend far beyond basic ORC and regenerative ORC, particularly with supercritical, trilateral, and multi-loop ORCs. Significant attention has been garnered by fourth-generation refrigerants like HFOs (hydrofluoroolefins), HFEs (hydrofluoroethers), natural refrigerants, and zeotropic mixtures, as research has migrated away from the popular HFC-245fa (hydrofluorocarbon). Performance-wise, the period was marked by a growing recognition of the diminished performance of physical systems under dynamic source conditions, especially compared to steady-state simulations. Through advancements in system control, especially using improved model predictive controllers, dynamics-based losses have been significantly reduced. Regarding practically minded investigations, research efforts have ameliorated working fluid flammability risks, limited thermal degradation, and pursued cost savings. State-of-the-art system designs and operational targets have emerged through increasingly sophisticated optimization efforts, with some studies leveraging "big data" and artificial intelligence. Major programs like SuperTruck II have further established the ongoing challenges of simultaneously meeting cost, size, and performance goals; however, off-the-shelf organic Rankine cycle systems are available today for engine waste heat recovery, signaling initial market penetration. Continuing forward, next-generation engines can be designed specifically as topping cycles for an organic Rankine (bottoming) cycle, with both power sources integrated into advanced hybrid drivetrains.

Keywords: organic Rankine cycle; engine waste heat recovery; advanced powertrains; fuel efficiency; engine waste heat; engine bottoming cycles; gaseous emissions prevention

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1. Introduction

Ten years ago, in 2013, the forerunning article to this review was published, describing the historical development and current state of research on recovering waste heat from engines using organic Rankine cycles (ORCs) [1]. Formally beginning in 1976 with Patel and Doyle describing the first internal combustion engine waste heat recovery (ICE–WHR) system utilizing an ORC [2], the article elucidated the growth of the field from its nascency to being regarded as the most promising approach for ICE–WHR over a nearly 40-year period. The unfamiliar reader is referred to that article for a chronological account of the developments and innovations that provided a foundation and sparked the subsequent explosion of research efforts in the ORC for ICE–WHR field.

The present article is classified as an update-style review article, serving as a continuation of the previous review article and aiming to provide a seamless, comprehensive history. This format reflects the maturity of the research field, and by intensively focusing on a 10-year period, it allows greater depth than ordinary reviews.

Structurally, the main body of this review proceeds chronologically, honoring each research effort by highlighting the most novel aspects and recognizing true innovations never previously described. This approach naturally facilitates the use of temporal context to identify the impacts of external influences (e.g., costs and legislation) on research trends. By contrast, vision-style articles skewing towards a narrative form have a distinct yet still important role, as most scholarly publications acknowledge; in fact, several vision articles are cited in this article.

The remainder of this introduction is arranged as follows: Section 1.1 outlines the essential ORC concepts necessary to establish key nomenclature and physical phenomena, Section 1.2 examines ICE waste heat sources to characterize the available waste streams, Section 1.3 presents recent contextual information that has influenced studies of the past decade, and Section 1.4 offers a brief synopsis of pre-2013 research to ground the reader in the foundations of recent efforts.

As a brief note before proceeding, it is observed that for any readers interested in the research and environmental/societal conditions contributing to the seminal publication of Patel and Doyle [2], Appendix A offers an early (pre-ICE–WHR) ORC history, and Appendix B describes the historical context of the transformative 1940s–1970s decades. Let us paraphrase Cicero: to be ignorant of (prior research) is to remain always a child.

1.1. Organic Rankine Cycle Concepts

What constitutes an organic Rankine cycle? As in modern organic chemistry, organic working fluids have at least one carbon atom on their (molecular) interior and may feature a wide range of other constituents [3], often produced through chemical synthesis, rather than being sourced from natural origins [4]. The Rankine cycle has been widely defined as a power cycle completed most basically through four idealized processes (isentropic compression, isobaric heat addition, isentropic expansion, and isobaric heat rejection), with numerous modifications (e.g., superheat, reheat, and regeneration) available for improving efficiency and overcoming practical challenges [1]. Thus, the Rankine cycle can be conceptualized as a power generation framework within which innovations occur, rather than any singular collection of thermodynamic states or processes.

These concepts are explored in far greater depth in Section 2 with the works of Peris et al. [5], Lecompte et al. [6], Apostol et al. [7], and others.

1.2. Waste Heat Recovery Source Characteristics

Primary waste heat streams from modern engines are engine exhaust, engine coolant, EGR (exhaust gas recirculation) coolers, and CACs (charge air coolers). Rather than the traditional "one-third rule of engines," today's best engines more closely follow a new rule of thumb that might be called the "half-quarter-quarter rule of engines," with roughly half of fuel energy becoming useful work (primarily as motive and secondarily for accessories), roughly a quarter being rejected at medium temperatures (primarily via exhaust and secondarily via EGR heat rejection), and roughly a quarter being rejected at low temperatures (primarily via coolant and secondarily via CACs). (Combustion inefficiencies, engine radiative heat losses, etc. are minor by comparison).

Each waste heat stream has a unique profile comprised of the quality of heat (indicated by temperature), amount of heat, steadiness/fluctuation, warm-up time, and source fluid characteristics. Aside from having a similar amount of heat, engine exhaust and coolant have nearly opposite profiles. Exhaust is hotter, gaseous, and more dynamic, and it has a shorter warm-up period.

Take a PSA (Peugeot/Citroën) DV6 TED4 1.4 L turbodiesel engine for example. Exhaust temperatures (after the turbocharger) typically fall between 250–400 °C [8]. By contrast, engine coolant temperatures remain low to limit in-cylinder temperatures and prevent the thermal degradation of lubricating oil. Common coolant temperature levels are 65–85 °C, corresponding to oil temperatures of 90–100 °C [9].

For the purposes of thermal bottoming cycles, the Carnot limit (η_{max}) on efficiency is much greater with hotter sources (higher T_H), giving engine exhaust more potential from a thermodynamic perspective than engine coolant. Likewise, since EGR is significantly hotter than charge air, EGR is the most attractive of the smaller heat streams (despite having a lower mass flow rate).

$$\eta_{max} = \frac{T_H - T_C}{T_H} = 1 - \frac{T_C}{T_H} \tag{1}$$

Studies have shown relatively similar waste heat conditions with hydrogen engines [10] with around 3% higher thermal efficiencies, due in part to higher rates of heat release at combustion [11]. Lean mixtures, low combustion temperatures, and high expansion ratios, such as those present in homogeneous-charge compression ignition (HCCI) engines, result in lower exhaust temperatures; however, these engines may still be good candidates for ORC-based WHR due to the low-temperature performance of ORCs and since the nascent uses of HCCI engines will occur in applications for which high efficiency is paramount [12]. Heat source considerations are discussed throughout the review of Section 2, and research gaps are identified for ORC–WHR from alternative engines/fuels in Section 4.

1.3. Recent and Future (Projected) External Influences

On a simple level, four main external conditions need to be present for the increased adoption of ORCs for ICE–WHR. Here, "external" is not directly related to ORC research and development, which is considered "internal" and is the subject of the main body of this article (Section 2). While the external influences are intertwined, a brief introduction can occur independently. Section 1.3.1 describes the demand for ICEs, setting the potential market size for ICE–WHR. Section 1.3.2 details fuel costs since one of the main motivations for WHR is fuel savings, even though other motivations may be wholly sufficient (e.g., legal, environmental). Section 1.3.3 covers regulations and policies, recognizing that fuel economy standards historically accelerate the adoption of efficiency-based technologies, and policies determine the availability of research funding. Section 1.3.4 outlines the alternatives to ORCs for ICE–WHR since competing technologies are also advancing alongside ORCs.

1.3.1. Demand for Internal Combustion Engines

According to Grand View Research, "the global internal combustion engine market is expected to reach 370,693.0 thousand (i.e., 370.6930 million) units by 2030 registering a CAGR (compound annual growth rate) of 9.3% during the forecast period. Rising disposable income in various nations and increased automobile use throughout the world are a major factor driving the market" [13]. This projection includes forecasts on regulations (including ICE bans [14]) and notes a branching of the technological development among OEMs (original equipment manufacturers) towards hydrogen, natural gas, and electric (or partially electric) vehicles. Similarly focused studies (with different date ranges) from other firms project moderately slower growth (e.g., Acumen projects a CAGR of 4.67% from 2018–2026 [15], while Technavio projects a CAGR of 8.32% from 2022–2027 [16]).

Taking only a couple of key metrics as examples on a multifaceted topic, the US EIA (Energy Information Administration) projects international growth of conventional (gasoline, diesel, etc.) light-duty vehicles (LDVs) through 2038 with a gradual growth in electric vehicle market share up to 31% at the end of the projection in 2050 [17]. (The projections are for OECD (Organization for Economic Co-operation and Development) countries. Separate reports are available for other major countries, such as China [18] and India [19]).

Recent industry efforts illustrate that new hydrogen and natural gas engines are being researched, developed, and deployed (including for use in hybrid drivetrains) [20]. Toyota has announced "that it is developing a hydrogen-fueled combustion engine" for "sports vehicles," and Cummins is introducing "a new natural gas heavy-duty powertrain" [13]. These projects illustrate continued interest in burning alternative fuels in ICEs, expanding and improving upon the many natural gas and bi-fuel engines already in service.

1.3.2. Combustion Fuel Costs

Accompanying the rise in sales of electric vehicles has been an increase in fuel prices, driven in part by global events like the Russia–Ukraine and Israel–Hamas wars. As shown in Figure 1, US combustion fuel prices have grown at a significant rate over the past two decades, enhancing the financial incentives for fuel savings.

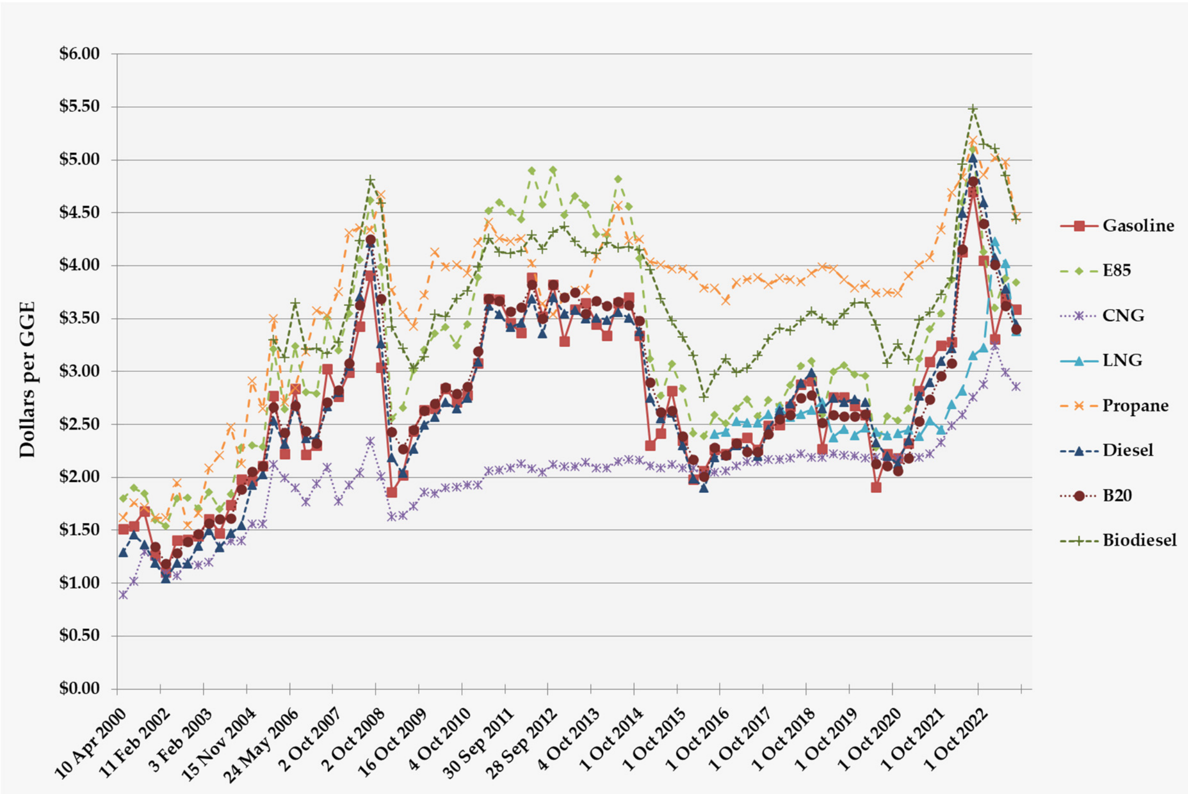

Figure 1. Historical US fuel prices on a gallons-of-gasoline-equivalent (GGE) basis [21].

These trends have global significance due to the globalized oil market, but they are not globally representative, as certain oil-producing countries have persistently low fuel costs of <1 USD/gallon (Venezuela, Libya, and Iran), whereas areas with significant gas taxes have costs consistently >10 USD/gallon (e.g., Nordic countries—Iceland and Norway). This variability is shown in Figure 2.

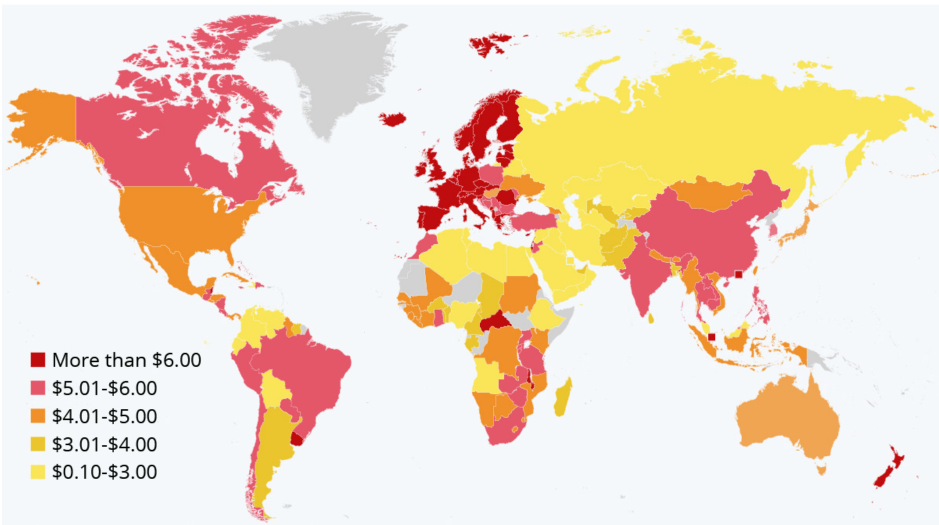

Figure 2. Global gasoline prices per gallon in May 2022 [22].

US residential electricity prices rose at a slightly slower pace than gasoline prices, nearly doubling from 0.0815 USD/kWh (April 2000) to 0.1591 USD/kWh (July 2023) [23]. As with gasoline, the US falls near the global median in electricity prices, with certain Middle Eastern and African countries offering cheaper electricity, and most European countries having expensive electricity. In recent years, fueling a gasoline vehicle has been roughly 2–2.5 times more expensive than charging an electric vehicle in the US (on a USD/mi basis).

1.3.3. Regulations and Policy

Regulations for internal combustion engines vary widely not only by region [24] but also across individual provinces [25] or states [26]. Furthermore, engine emissions are often regulated independently, based on their application. For example, US engines are subjected to different regulations for stationary power engine-generators, long-haul truck engines, heavy-duty equipment engines, lawn mower engines, vehicle engines, motorcycle engines, marine engines, aviation engines, and so on. Consequently, the present study encourages researchers to consult large-scale studies to understand the regulatory environment now and in the future [24,27,28].

Being the most prevalent and regulated engines, vehicle engines are subject to progressively tighter emissions limits and fuel economy standards. In the US, the most stringent regulations are the Advanced Clean Cars II standards of California, while China recently implemented National VI B standards [29], and Europe is finalizing the Euro 7 regulations [30]. Incrementally tighter regulations drive innovation, such as incorporating ORC–WHR technologies to limit CO_2 emissions. By contrast, certain regions are planning explicit engine bans decades in the future; however, that approach tends to be less popular among citizens, so certain governments may opt instead to “effectively ban” engines through unrealistic emissions and/or fuel economy thresholds. For the majority of countries, engines will continue being prevalent, and local environments will benefit from engines incorporating more advanced combustion and gas flow techniques [31], as well as more efficient aftertreatment systems (including reduced cold start emissions) [28], burning alternative fuels (natural gas, hydrogen, methanol, biodiesel, etc.) [32–35], and incorporating WHR [36].

1.3.4. Alternative ICE–WHR Methods

While the majority of ICE–WHR efforts continue to be based on the Rankine cycle [37], various alternative approaches consistently garner attention from researchers. Stirling cycles, for example, continue to be developed for ICE–WHR and industrial WHR [38,39], and similarly, thermoelectric generators (TEGs) are considered for ICE–WHR [25,40] and other WHR [41,42]. TEGs are designed with a diverse group of materials [43,44], ultimately aiming to assuage long-standing concerns over cost and efficiency. Turbocompounding is a simpler, yet less effective, option [45]. Six-stroke engines [46], Kalina cycles [47], and Brayton cycles [48] are also discussed to a lesser extent.

Advancements in WHR technology for non-ICE applications also contribute to ICE–WHR development. For example, the expansion of ORCs in industry [49] and the energy sector [49,50] gives ORCs additional advantages in other sectors, including ORCs for ICE–WHR [50]. More specifically, working fluid and ORC system component economies of scale improve with additional applications, and improved component designs and control strategies are developed, advancing the efficiency and affordability of ORCs. Continued research on Rankine cycles (across applications) also generates promising concepts like trilateral cycles, which are considered a modified Rankine cycle [51].

1.4. Pre-2013 Historical Synopsis of ORCs for ICE–WHR

Technological developments from the mid-1970s to the early 2010s are described in [1], which is considered background reading for the present article. Pioneering efforts on ORC–WHR from engines occurred in 1976 with Patel and Doyle of Thermo Electron Corporation [2] as a redirection of efforts oriented towards replacing ICEs with ORCs [52], as described in Appendix A. The first extended testing of such a system was reported in 1979 as a collaboration between Thermo Electron Corporation and Mack Trucks [53].

Through the ensuing decades, dozens of studies were carried out on ORC performance with different working fluids (e.g., Marciniak [54]), methodologies for selecting working fluids (e.g., Badr et al. [55,56]), the thermal stability of working fluids (e.g., Angelino and Invernizzi [57]), alternative small-scale expanders (e.g., Leibowitz [58]), different degrees of superheat (e.g., El Chammas and Clodic [59]), achieving transient control (e.g., Endo [60]), and system optimizations (e.g., the thermo-economic optimization of Quoilin et al. [61]). Studies of the period cover light-duty vehicles (e.g., Hussain and Brigham [62]), commercial long-haul trucks (e.g., Espinosa [63]), ships (e.g., Schmid [64]), hybrids (e.g., Arias et al. [65]), and stationary ICEs (e.g., Vaja and Gambarotta [66]).

The period leading up to the previous review was especially fruitful with works like the BMW turbosteamer described by Freymann et al. [67], featuring two WHR loops, a high-temperature steam Rankine, and a low-temperature ORC using ethanol. Although the system was complex and costly, the authors saw the potential for improving the four-cylinder spark-ignition engine's efficiency by 15% [68]. A pared-down version of the system was suitable for packaging in a BMW 5 sedan, for which an innovative impulse turbine could improve fuel economy "by up to 10% on long-distance journeys" [68].

The main body of this review article describes another chapter, the most recent chapter, in the history of ICE–WHR via ORC. In other words, this article intentionally builds on the previous article to comprise a seamless literature review corpus, offering researchers a complete chronological history, which is the format most capable of showing historical trends, contextualizing innovations, considering external influences, and projecting future directions.

2. Latest Decade in Review

This main section presents the available ORC for ICE–WHR literature published in roughly the past decade. A brief clarification on the timeline: the first articles cited are from 2012 because the original review article was finalized, accepted, and posted online in 2012 (despite a journal issue date of 2013).

The compilation of articles occurred over the entire decade through monitoring relevant thermal/energy publications (e.g., SAE International and Applied Energy), employing date-restricted Google Scholar and database (e.g., WorldCat and FirstSearch) keyword searches, surveying research community/forum (e.g., Academia and ResearchGate) recommendations, and tracking governmental and industry project reports/updates. Studies including complete ORC systems receive detailed treatment, whereas narrower studies on specific aspects of the topic (e.g., expanders and working fluids) are covered more briefly (or cited in the summary of Section 3). Also, relevant review papers and vision articles are described briefly and tabulated in Appendix C. Significant effort was made to capture and highlight the important works of all researchers.

2.1. 2012 (Partial)

Qiu and Hayden [69] studied a micro-CHP (combined heat and power) system with a natural gas burner, for which power generation occurs via a thermoelectric generator (TEG) "topping cycle" and an ORC "bottoming cycle," forming a dual-cycle micro-CHP system (sized for a residential house). Although ICEs are prominently mentioned as a possible micro-CHP system component, the system under study has no ICE. Pertinently, the investigation evaluates the potential of integrating thermoelectrics with ORCs for enhanced system performance. The authors use experimental results from two TEG modules, each containing 325 PbSnTe (lead–tin–telluride) thermoelectric couples in combination with a computer model of an ORC with R245fa working fluid and a scroll expander to find a combined electrical efficiency of up to 17%. Optimal performance occurs for burner operating temperatures ~970 °C and wall temperatures ~600 °C, at which thermoelectric power generation peaks at 22% of the total power output, suggesting that thermoelectrics can significantly boost ORC performance.

Shu et al. [70] also feature a combination of a thermoelectric generator and an ORC, yet their system focuses on WHR from a WD10D235 engine. The overall TEG-ORC system for ICE–WHR is shown in Figure 3. Engine exhaust encounters the TEG before entering the ORC evaporator, where additional exhaust WHR occurs, and the cycle features two main stages of preheating, the first with engine coolant and the second via internal heat exchange with the working fluid exiting the turbine expander.

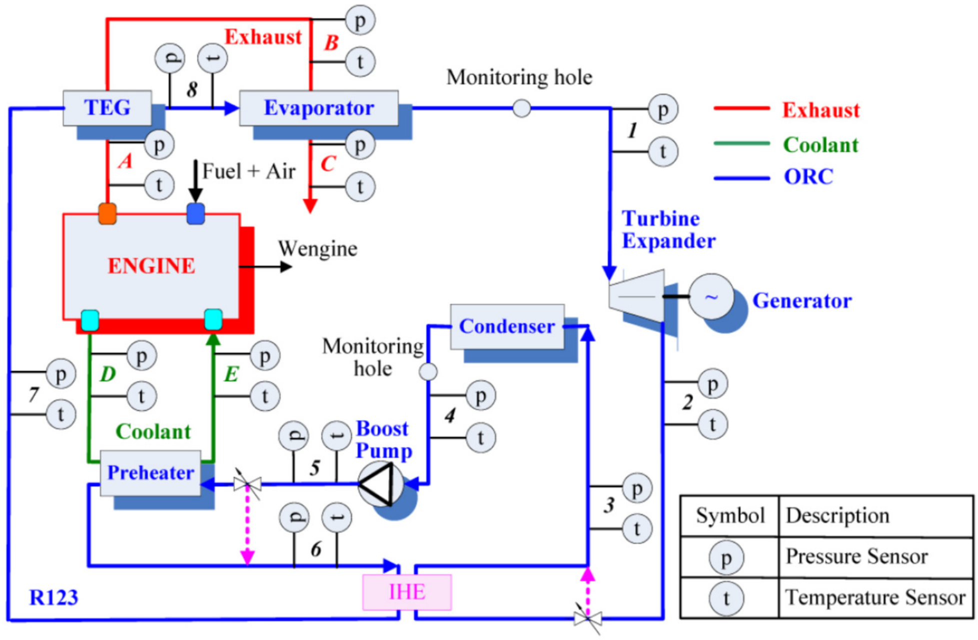

Figure 3. Schematic diagram of combined TEG-ORC waste heat recovery system from an engine, showing the flows of exhaust (red), coolant (green), and the ORC (blue) [70]. Reprinted with permission from Shu et al., SAE Technical Papers; published by SAE International 2012. The authors intentionally place the TEG immediately after the engine, generating TEG power while reducing exhaust temperatures, citing the susceptibility of organic fluids to damage with exposure to high-temperature exhaust. TEG-ORC system evaluation occurs through building a simulation model in MATLAB/Simulink, using a TEG model validated using the results of Hussain et al. [71] and generating performance maps across condensing and evaporating pressures. With R123 as the working fluid, the maximum TEG-ORC system performance boosts engine efficiency from 40.28% to 45.00%. For the thermoelectric generator composed of "p-type tellurium, antimony, germanium and silver (TAGS)/n-type PbTe," the authors lament the high cost and note that "only a small portion of the power output" is generated via the TEG. (TEG production varies between ~1.25–1.70 kW, with the overall system producing ~22.0–27.7 kW). Consequently, while cooling the exhaust prior to the ORC's heat recovery heat exchanger is considered an important feature, and the ability to power accessories with thermoelectrics is valuable, the authors emphasize the need for reduced TEG costs.

Latz et al. [72] also offer a theoretical study, intensively examining the relative attributes of different working fluids for basic subcritical and supercritical Rankine cycles. Different inorganic fluids (water and ammonia), organic fluids (ethanol, methanol, HFO-1234yf, HCFC-123, and HFC-152a), and mixtures (R430A, R431A, water/ammonia, water/ethanol, and water/methanol) are selected for the study despite certain known operational challenges (e.g., the frost point of water). Using Engineering Equation Solver (EES) and REFPROP with reasonable cycle parameters (e.g., $p_{max} = 60$ bar, $p_{min} = 1$ bar, $T_{cond} = 323$ K, and $\eta_{exp} = 0.9$), the authors perform first-law (energy-based) and second-law (exergy-based) analysis. This parametric analysis generates a multitude of results across three different maximum cycle temperatures (420 K, 520 K, and 670 K), which represent different heavy-duty diesel engine (HDDE) operating conditions. Among the selected working fluids, the supercritical Rankine cycle did not to offer a performance increase sizable enough to justify the significantly higher operating pressures, and a zeotropic mixture of 80% water/20% methanol (by mass) is the most favored subcritical Rankine working fluid. While this mixture is stable to 620 K, the authors note the possibility of gaining another 30 K of stability (to 650 K) by switching to water/ethanol with only a modest performance penalty (mostly at mid-range temperatures).

Heberle et al. [73] perform an early exergoeconomic optimization of a geothermal ORC. Despite the non-ICE heat source, the effort is included here to show the application of the exergoeconomic method of Bejan et al. [74] to ORCs, which parallels the thermo-economic optimization of Quoilin et al. [61], and to draw attention to the important related work of the authors over the decade (including [75–77]). For this study, the geothermal water temperature is 120 °C, slightly exceeding typical engine coolant temperatures. Exergy costing is performed in the mold of Tsatsaronis and Winhold [78], where exergy flow rates ($\dot{E}$) are multiplied by specific costs (c) to obtain cost stream rates ($\dot{C}$), as follows:

$$\dot{C}_i = c_i \dot{E}_i \tag{2}$$

Cost stream rates are summed in combination with other costs (e.g., operation and maintenance). Optimization can be defined as the minimum total cost rate of the product (electricity) ($\dot{C}_{P,total}$) or the minimum specific cost rate of the product ($c_{P,total}$), and the authors choose the latter based on the system's varying power output.

Additional exergoeconomic metrics, such as relative cost difference (r_K) and exergoeconomic factor (f_K), highlight opportunities for performance improvement or cost savings at the component level. With the preheater as an example, both metrics remained high across the simulated heat exchanger pinch point temperature differences and working fluids (isobutane and isopentane), suggesting that "it should be attempted to reduce the capital investment costs of the component at the expense of efficiency". Regarding the entire system, the authors conclude that "under exergoeconomic criteria isobutane with a minimum temperature difference of 3 K at evaporation and 7 K at condensation is to favour".

Chys et al. [79] examine the potential of zeotropic mixtures as ORC working fluids, considering two generic heat sources with temperatures of 150 °C and 250 °C and mass flow rates of 15 kg/s and 25 kg/s, respectively. As mentioned by Latz et al. [72], this same year, zeotropic mixtures were commonly explored as an avenue to reduce the irreversibilities associated with isothermal evaporation/condensation, an effect that is discussed in considerable depth. For instance, the use of fluids with "large differences in evaporation temperatures" allows for steeper temperature glide during a phase change, yet the "fractionating" (one fluid predominantly vapor and the other predominantly liquid) can cause significant composition shifts around leakages. The authors simulate the performance of 10 pure fluids (plus the azeotropic mixture Solkatherm), 8 two-component mixtures, and 3 three-component mixtures for Rankine cycles with and without a recuperator. (These fluids are numbered 1–21). Using a FluidProp (with REFPROP) model capable of identifying the optimal thermodynamic states for the cycles, the authors analyze cycle performance across mixture compositions, including simultaneously varying the concentration of two components of each three-component mixture. The use of binary mixtures is shown to increase power production by 12.3% for the 150 °C source and 5.5% for the 250 °C source, with trinary mixtures and the inclusion of a recuperator offering little additional power. The authors note as a caution that certain influences of mixture composition on the performance of individual components (pump, turbine, and heat exchangers) may be oversimplified or unaccounted for, and certain mixtures are patented.

Saidur et al. [80] offer a broad survey-type overview of the "latest developments and technologies on waste heat recovery" from ICE exhaust. Four different WHR technologies receive a section: thermoelectrics, six-stroke ICEs, Rankine bottoming cycles, and turbochargers. Each section covers basic concepts and cites recent findings, making an effort to outline the technological challenges and recent advances for each WHR method, offering a helpful introduction for researchers who are new to the ICE–WHR landscape.

Boretti [81] published a study on using an ORC with R245fa working fluid to recover engine exhaust and/or coolant waste heat from a naturally aspirated 1.8 L spark-ignition engine for use in charging the battery pack of a hybrid passenger car. The simulation-based study uses a WAVE engine model and GT–COOL ORC model (with off-the-shelf components) to generate full BMEP performance maps for each WHR system architecture. The results indicate that the ORC–WHR options offer the following increases in fuel efficiency: 3.4–6.4% for exhaust only, 1.7–2.8% for coolant only, and 5.1–8.2% for combined (exhaust + coolant). The author notes that, despite these promising results, certain energy conversions and additional engine backpressure are not included, and "major downfalls" of the technology include the cost, weight, control, and packaging of the ORC system.

An additional study by Boretti [82] involves a novel Rankine cycle "conceived evolving the BMW Turbo steamer concept," operating from a turbocharged 1.6 L direct-injection engine burning ethanol. (Baseline data, such as transmission and final drive ratios, are derived from a 4 L naturally aspirated gasoline engine). In this research and another related work [83], the author describes that, rather than a dual-cycle ORC using R245fa, there could be a single-loop Rankine cycle using water, with the engine's cylinder head serving as the cycle's boiler/pre-heater. Also, instead of using WHR power to charge batteries, the cycle's expander is integrated with the transmission through an electric clutch and motor/generator. To perform the analysis, the studies use Lotus Vehicle Simulation (LVS) software, simulating over the New European Driving Cycle (NEDC). (The NEDC begins at "cold start conditions," consisting of four successive Economic Commission for Europe (ECE) cycles, followed by a single Extra Urban Driving Cycle (EUDC)). In addition to being more realistic than steady-state simulations, the NEDC allows the author to examine the potential for exhaust heat to improve the single-loop cycle's cold start performance.

Without WHR, the engine coolant warms in 600 s, and the oil stabilizes by 1000 s, so applying the modeling assumption that the Rankine cycle will not produce power during the first 600 s of the NEDC is considered conservative. (Exhaust WHR should reduce warm-up times). The integrated single-loop Rankine cycle improved fuel economy by 4.2% over the full NEDC, including 6.4% during 120 km/h cruising. Going further, a mild-hybrid architecture with regenerative braking and thermal engine start/stop achieves a 10.2% fuel economy improvement.

Arunachalam et al. [84] explore all the waste heat sources of heavy-duty diesel truck engines and the potential for recovering heat from multiple streams in a single-loop Rankine cycle. The engine exhaust, charge air cooler, and EGR cooler are identified as viable heat sources, whereas the engine coolant is ruled out because of its low temperature (limiting recovery efficiency) and the large heat exchangers required. The specifics of the study include the use of IPSEpro software to model the Rankine cycle, operating from a Volvo D-13 engine, simulated at 12 points of the European Stationary Cycle (ESC 13). For the cycle's working fluid, the authors initially favor a mixture of 80% water/20% methanol (by mass) before ultimately using pure water due to the difficulties in calculating transport properties of the mixture using REFPROP (v9.0).

After simulating WHR from different combinations of the three viable heat sources previously identified, with single- and dual-loop Rankine configurations, the authors conclude that the most attractive system performs only EGR recovery with a single-loop Rankine. For engine operation hovering between a 25–50% load, the results show virtually no benefit in additionally recovering heat from the exhaust or charge air cooler. According to the authors, "It is evident that even with more quantity of heat energy, it is the quality of the heat that determines the useful power output". Regarding the dual-loop ORC, an increase in turbine power from the single-loop EGR-only output of 11.7 kW to the dual-loop tri-source (EGR + exhaust + CAC) output of 16.3 kW is not sufficient to justify the additional weight, complexity, and cost.

De Ojeda and Rajkumar [85] of Navistar detail the historical strategies used to meet emissions regulations (multiple injections, multiple turbochargers, EGR cooler, and aftertreatment devices), new technologies under study (an injection system matched to a combustion chamber and specially designed electrical turbocompounding), and future technologies to be evaluated (variable valve timing and organic Rankine cycle) as part of the SuperTruck project funded by the US Department of Energy. Navistar's MAXXFORCE13 diesel engine is shown to reach 46.5% efficiency with the newly evaluated technologies, up 3.5% from base, yet an exploration ORCs as part of a "total vehicle perspective" is still required to reach the SuperTruck program's 50% target (evaluated at 65 mph, under level conditions, with a 65,000 lb. vehicle weight). High and low NO_X combustion modes are considered within the context of approaching fuel economy "from a total vehicle perspective" since engine and emissions efficiency goals must be achieved together in a "cost effective manner".

Skarke et al. [86] explore the application of an R245fa ORC to the Ohio State University EcoCAR, a student PHEV (plug-in hybrid electric vehicle). The vehicle powertrain consists of a 1.8 L E85 (an ethanol blend containing up to 85% ethanol) engine coupled with an 82-kW motor/generator at the front of the vehicle with a 103 kW motor at the rear. Simulations over the NEDC (including urban and highway conditions) are accomplished using a validated energy-based vehicle simulator with a "quasi-static model of an Organic Rankine Cycle (ORC)". Under the relatively low engine speed and torque conditions present during the NEDC, only around 1.7% of the fuel energy is recovered, whereas 2.4% of fuel energy is recovered during 65 mph highway driving. Since 34.05% of fuel energy becomes engine power at highway conditions, the additional 2.4% of fuel energy resulting in ORC power constitutes a 7% decrease in fuel consumption. The next stage of the authors' exploration of ORC is planned to include modeling engine warm-up and "the thermal dynamics of the aftertreatment system".

2.2. 2013

Quoilin et al. [87] published an impactful broad survey on ORCs, describing the opportunities available for ORCs in different applications (biomass, geothermal, solar thermal, and waste heat recovery) and opining with insights on working fluids and cycle components. The authors split WHR into two classifications, industrial and ICE sources. Being limited by the wide scope of the survey on ICE–WHR, the authors merely review a handful of the most well-known studies, predominantly from large OEMs. Regardless of limitations, the article is highly effective in tabulating theoretical and practical ORC knowledge. A brief sampling is included in the following paragraph, emphasizing far-reaching insights and ICE–WHR specifics.

The survey includes profiles of ORC manufacturers (applications, sizes, heat source temperatures, working fluids, and expanders), mostly geolocated in Europe and the US, making predominantly larger-scale systems. Connections between the research and industrial communities are increasingly vital as the ever-growing body of ORC knowledge gains application. Smaller WHR ORCs are shown to suffer from high specific costs (€6000+/kW at 10–20 kW vs. €900–2100/kW at >100 kW), which is fleshed out as the article proceeds into discussing working fluids and system components. Among dozens of treatments in the literature, perhaps the most succinct outline of working fluid selection criteria is found in the article, due in part to identifying real-world tradeoffs. For instance, high critical temperature fluids tend to be capable of higher thermal efficiencies (with lower back work ratios) and have higher thermal stability; however, the same fluids tend to have lower vapor densities, requiring higher volumetric flow rates and consequently larger components (e.g., heat exchangers). The article also features tabulated results of narrower studies focused on working fluid selection and expander performance. Unfortunately, working fluid studies range significantly in source temperature and size, making those findings less applicable to ICE–WHR than the positive displacement expander performance results, which are skewed towards smaller systems. For the surveyed research, scroll expanders provide isentropic efficiencies exceeding 70%, whereas screw expanders tend to perform below 60%.

Auld et al. [88] published a similarly structured study at the same time. Their coverage includes three "case studies": (1) ICEs, (2) hot brines, and (3) industrial plants. Subsequently, ORC modeling is treated with frequent reference to a geothermal study by Aneke et al. [89], in which the authors create a model in IPSEpro with fluid property support from REFPROP.

Tian et al. [90] introduce a "combined power and cooling cycle driven by exhaust" WHR from a diesel engine generator. The "basic combined" organic Rankine cycle–compression refrigeration cycle (ORC–CRC) consists of a four-component ORC powered via exhaust heat, for which the expander power mechanically drives the compressor of the four-component (vapor) compression refrigeration cycle. Two improvements to the "basic combined" system are also simulated, namely adding a heat accumulator to the ORC and replacing the refrigeration cycle's throttling valve with an expander (sending additional power to the compressor). For the working fluid, carbon dioxide is chosen for both cycles, which operate transcritically (within and above the vapor dome) due to the low critical temperature of carbon dioxide (30.98 °C). Four main advantages of carbon dioxide are given: its low cost, environmental friendliness, safety, and good thermodynamic properties (especially its refrigeration capacity). The authors construct a simulation model in EES, using known exhaust conditions and an evaporation temperature of 0 °C, with assumed isentropic efficiencies and pinch point temperature differences. Appropriately, the authors measure system performance with a uniquely defined coefficient of performance (COP) equal to the CRC evaporator heat input divided by the ORC heat input. Among an array of results across maximum ORC and CRC pressures, an especially noteworthy coefficient of performance for the basic combined ORC–CRC is around 0.19, occurring at a maximum ORC pressure of 10 MPa and an optimal CRC maximum pressure of 8.8 MPa. The ORC heat accumulator provides significant improvements to the COP when simulated with large temperature drops of 100–300 °C.

Garg et al. [91] study the potential of mixtures of isopentane and R-245fa for ORC–WHR from various heat sources with temperatures ranging from 380–425 K. The preferred working fluid formulation is a 70/30 mixture (molar basis) of isopentane to R-245fa, which is simultaneously less flammable than isopentane and has a lower GWP than R245fa. And while the study's heat source temperatures lie below exhaust temperatures, the relatively high critical temperature (444.6 K) of the mixture indicates potential for application to higher temperature heat sources. Using MATLAB, with REFPROP for thermodynamic properties, the authors build a regenerative ORC computer model to evaluate the performance of the mixture. Key parameters include an 85% pump efficiency, a conservative turbine efficiency (65%) representative of small capacity systems, pinch point temperature differences $\geq$ 5 K, and a leakage loss functionally dependent on the pressure difference. Calculated efficiencies for the regenerative zeotropic mixture ORC are ~10.7–13.2% across the heat source temperature range. Through irreversibility analysis to find the main reasons for the ORC achieving roughly half the Carnot maximum, the authors identify the turbine as the main source of irreversibility and highlight a method used in solar ORC design for determining the required heat source temperature based on the desired system output power and turbine/expander inlet conditions.

Bari and Hossain [92] detail the creation of an experimental WHR apparatus placed within the exhaust stream of a 13B Toyota diesel engine. Two commercially available shell-and-tube heat exchangers are installed, one to produce saturated vapor (saturated steam) and the second to produce superheated vapor (steam). Unfortunately, the off-the-shelf heat exchangers exhibit underwhelming heat recovery performance, with typical effectiveness values of ~0.45–0.55. To achieve greater WHR, the authors design a 2-m-long baffled shell-and-tube heat exchanger in SolidWorks and optimize its performance with ANSYS (for a 2200 RPM engine speed), again with one heat exchanger functioning as a vapor generator and one as a superheater. While validating the model against experiments, the authors recognize that ANSYS overestimates the heat transfer performance by around 10%, perhaps due to (unaccounted-for) tube fouling. Final simulations of heat recovery, coupled with chosen isentropic efficiencies for a pump and expander, form the basis for the authors to estimate the WHR performance of a steam Rankine cycle. Under ideal circumstances (operating at around 30 bar), the system increases engine power by 23.7% and improves BSFC (brake-specific fuel consumption) by 20%.

Latz et al. [93], the authors of the previously discussed article on working fluid selection [72], published a study describing expander selection for both a heavy-duty diesel engine (12.8 L) and a light-duty gasoline engine (2.0 L). Importantly, the work is closely tied to the "similarity concept" of Nichols [94], which is centered on the use of two primary and two secondary parameters for selecting an expander. The primary parameters are specific speed (N_s) and specific diameter (D_s), which are a function of expander rotational speed (N) and expander diameter (D), respectively, while both are dependent on the volumetric flow rate ($\dot{V}$) out of the expander and the specific enthalpy drop (Δh) across the expander.

$$N_s = N \frac{\dot{V}_{exp,out}^{1/2}}{\Delta h_{exp}^{3/4}} \tag{3}$$

$$D_s = D \frac{\Delta h_{exp}^{1/4}}{\dot{V}_{exp,out}^{1/2}} \tag{4}$$

(Note that Nichols uses adiabatic head (H_{ad}) instead of a specific enthalpy drop). Secondary parameters are the Mach number and Reynolds number of the flow entering the expander, which have a smaller impact on expander operation.

When the large diesel is first investigated, an 80/20 (by mass) water/methanol mixture is the favored working fluid based on earlier work, and consideration is given to heat recovery from the exhaust only, EGR only, exhaust + EGR (parallel), and exhaust + EGR (serial). Using a Rankine cycle model and engine data, the flow rates and enthalpy drops needed for the specific speed calculations are identified. Furthermore, by assuming a speed range for displacement expanders (<4000 rpm) and small turbines (50,000–120,000 rpm), the resulting specific speeds can be calculated. As shown in Figure 4, a displacement expander would need to operate between 0.01 and 0.1, whereas a turbine expander would operate between 0.5 and 3. Consequently, since turbine expanders do not operate well below 6, a reciprocating piston (displacement) expander is appropriate for the heavy-duty diesel WHR system. (For additional clarity, consult Figure A1 of [93]).

As an academic/illustrative exercise, the recovery of exhaust heat from the gasoline engine is performed using R123, providing lower specific enthalpy drops and higher specific speeds, enabling the selection of a turbine expander. Thus, rather than pursuing maximum performance, since low pressure ratios cause low turbine efficiencies, the study demonstrates the use of thermodynamic parameters to ensure system compatibility with a certain type of expander, which comports with the primary aim of the article.

Auld et al. [88] published a comparative study of WHR using ORCs at power levels from 10 kW to 10 MW, considering the dry and isentropic fluids recommended by Tchanche et al. [95]. Three WHR cases are examined: ICEs, hot brines, and industrial plants. Employing a distinctive tactic, the MATLAB and FluidProp model specifies two pinch point temperature differences (evaporator and condenser) while leaving the condenser pressure to be calculated iteratively. This approach is distinguishable from that of Aneke et al. [89], in which the turbine outlet pressure (into condenser) is specified. Regarding the comparative nature of the study, emphasis is placed on the importance of designing to maximize power output, rather than cycle efficiency, for which the turbine inlet pressure (TIP) is a balance between heat input (inversely related to TIP) and cycle efficiency (proportionally related to TIP). (A practical upper limit of TIP as 81% of the critical pressure is also noted). The optimal cycle (the subcritical, superheated, and trilateral flash cycle) is shown to depend on the T-H profile of the source fluid, particularly when the "gradient of the source T-H profile is shallower than that of the ORC working fluid" [88].

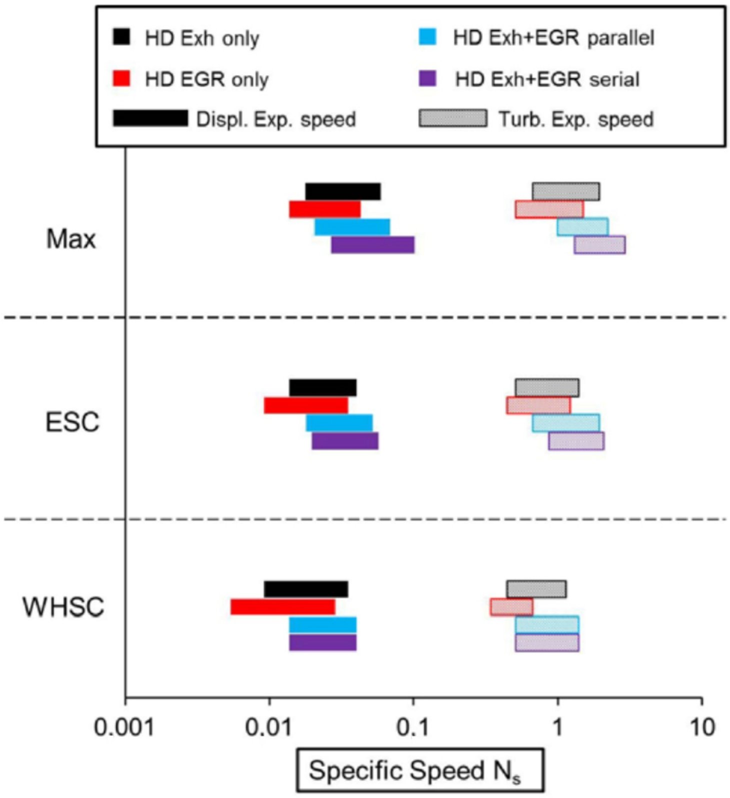

Figure 4. Expander-specific speeds for displacement and turbine expanders under different engine operating conditions and WHR system configurations [93]. Reprinted with permission from Latz et al., SAE Technical Papers; published by SAE International 2013.

Koeberlein [96] gives an update to Cummins's SuperTruck Program, based on a 2009 Peterbilt 386 Tractor and Cummins 15L ISX Engine. (SuperTruck program presentations are given on a regular basis at conferences [97,98] and to the government [96,99–101], so practical limitations preclude the reporting of every project update). Spanning from April 2010 to April 2014, the program starts at a baseline engine brake thermal efficiency (BTE) of 42%, and 51.1% is demonstrated at the time of the project update. This BTE milestone satisfies program Objective 1, which is 50% BTE. Moreover, 2.8% of the gains are attributed to WHR, which is later detailed as "Cummins organic Rankine cycle" WHR. In accordance with a previous presentation [99], the ORC is said to perform the recovery of EGR and exhaust heat. As Cummins pursues the 55% BTE of Objective 3, "WHR integration" is stated as one of the mechanisms for fuel economy improvement with the WHR system being assigned a total fuel economy improvement goal of 6%. (Objective 2 is associated with freight efficiency, which is weakly connected to the ORC system, mainly in the form of additional vehicle weight).

Sprouse III and Depcik [102] explore the use of ORCs with dry fluids to perform exhaust WHR on much smaller diesel engines. In fact, the waste heat source is a single-cylinder Yanmar diesel engine generator running at five separate operating points, each with known exhaust conditions (e.g., temperature, pressure, and A/F). A MATLAB and REFPROP model is designed based on an important observation from the literature that dry fluids scarcely benefit from superheat [59]. In essence, this means that saturated vapor (quality = 1) can be pulled from the top of the evaporator and fed to the expander, where the dry fluid's positive saturated vapor curve slope guarantees vapor conditions at the exit. This leads to the use of two-zone heat exchangers, where separate heat transfer coefficients are used to account for the enhanced heat transfer associated with two-phase liquid–vapor mixtures. Surface areas are allowed to vary in each zone of the heat exchangers (with the total surface area fixed), according to the area required to reach the vapor dome (heating in the evaporator and cooling in the condenser), and basic fan laws account for the use of forced convection of air across the condenser.

Longer-chained hydrocarbon fluids are shown to operate efficiently at higher pressures and expansion ratios, being more suited to screw expanders, whereas the other fluids (e.g., R245fa) fit more closely within the operating range of scroll expanders. Practical considerations lead to the exclusion of certain fluids, such as the low condensing pressure of toluene, benzene, and hexane and the high evaporating pressures of butane. The remaining fluids are ordered based on net power production and overall system efficiency (not merely ORC thermal efficiency), and pentane is found to provide around a 10% increase in engine generator efficiency, with thermal efficiencies approaching 15% for mid–high engine load operating points. The authors are among the first to posit the connection between designing the dry fluid ORC without superheat and using two-zone (moving-boundary-type) heat exchanger models, as well as illustrating that increasingly efficient heat transfer in the two-phase liquid–vapor portion of the evaporator (compared to the liquid portion) can drive the optimal flow configuration towards a parallel flow despite a counter-flow being preferable under most circumstances.

Jadhao and Thombare [48] published a review of all ICE–WHR structured on a conceptual basis to explain the basic function of each WHR technology and the optimal circumstances for employing each technology in a style necessitating relatively few literature citations for each WHR approach. The article covers basic information about thermoelectric generators and Rankine cycles (e.g., steam and ORC), as well as a variety of other WHR options (e.g., Stirling cycle, refrigeration, and turbocompounding). Perhaps one of the most important aspects of the article is the emphasis on using WHR to reduce thermal pollution, an underappreciated environmental benefit on top of reducing fuel consumption. Like the article of Saidur et al. [80], the article works well as an introductory text.

El-Emam and Dincer [103] build on the exergoeconomic work of Heberle et al. [73] and others, publishing an article focused on geothermal ORCs. The article has much to offer ICE–WHR researchers, with an early, thorough presentation of exergy, exergoeconomic, and sustainability-focused analysis. Specifically, the authors use exergetic analysis to analyze the "exergy rates" ($\dot{E}_X$) (and exergy destruction rates, $\dot{E}_{Xd}$) occurring throughout the cycle, leading to the determination of the cycle's exergetic efficiency (ψ), and ultimately the sustainability index (SI).

$$\psi = 1 - \frac{\sum \dot{E}_{Xd}}{\dot{E}_{X,\ heat\ input}} \tag{5}$$

$$SI = \frac{1}{1-\psi} \tag{6}$$

The exergoeconomic analysis uses a format similar to Equation (2), where cost factors like the purchase equipment cost (PEC) are determined using separate correlations for expanders, heat exchangers, and pumps as functions of design parameters (e.g., work output and surface area). Purchase equipment costs are translated to total capital investment costs, as well as operations and maintenance costs, using literature-based scaling factors. Using Engineering Equation Solver (EES), the authors simulate a regenerative ORC using isobutane across dead state temperatures, expander inlet temperatures, and evaporator pressures, leading to plots of energy and exergy efficiencies, as well as total costs. At the base geothermal temperature of 165 °C, the system has an energy efficiency of 16.37% and an exergy efficiency of 48.8%, and around 75% of the exergy destruction is shown to occur in the two large heat exchangers (the evaporator and the condenser).

Lecompte et al. [104] elucidate a four-step methodology for designing an ORC for a combined heat and power system used in building applications. In the system, ICE waste heat is utilized for building heating and ORC power generation. The process includes (1) sizing the system for fixed conditions, (2) modeling the part-load regime, (3) simulating quasi-static part-load operation over a year, and (4) optimally sizing the system. Unique to the study are the full-year simulation and the minimization of the pervasive use of specific investment cost (SIC) as "the objective criterion". At the end of the study, R152a is the preferred working fluid, with a system input heat of 2050 kW and an optimal SIC of 2210 €/kWe.

$$SIC = \frac{Cost_{Components} + Cost_{Labor}}{\dot{W}_{net}} \quad (7)$$

Peris et al. [5] investigate the use of six different ORC configurations with ten (non-flammable) working fluids for WHR from ICE coolant. In this way, their article provides ORC researchers with a comparative evaluation across typical ORC configurations, while performing WHR on the significant quantity of energy is often discarded based on its low exergetic quality. As shown in Figure 5, the configurations are as follows:

(a) Basic ORC (BORC)—the simplest configuration with the familiar four components.
(b) Regenerative ORC (RORC)—the basic configuration with a regenerator added, using the heat remaining after expansion to preheat the fluid before the evaporator. (A regenerator could also be called a recuperator).
(c) Double regenerative ORC (DRORC)—a complex configuration with heat entering one RORC at a higher temperature and then the remaining heat entering another RORC at a lower temperature with a common condenser.
(d) Reheat regenerative ORC (RRORC)—a regenerative configuration with a second heating stage and second expansion stage.
(e) Ejector ORC (EORC)—a unique configuration with a portion of the condensed fluid being pressurized by a separate pump and heated in a second evaporator before passing into an ejector, which is a mixing chamber where the fluid stream just mentioned pulls (or sucks) the fluid exiting the expander, reducing expander backpressure.
(f) Transcritical regenerative ORC (TRORC)—a regenerative configuration in which a pump pressurizes the working fluid above critical pressure, often called "supercritical".

Each configuration is considered in combination with each working fluid (R134a, R245fa, R227ea, RC318, R236fa, SES36, R125, R218, Novec649, and Novec7000). An evaluation of thermodynamic cycle performance occurs using an energy and isentropic efficiency-based model. Also, the "turbine size parameter" of Invernizzi et al. [105], which is essentially the turbine diameter metric of Latz et al. [93] without the specific diameter (see Equation (3)), provides insights into feasibility. In this way, the DRORC using SES36 offers a thermal efficiency of 7.15%; however, the additional components and expander size of the DRORC lead the authors to prefer the RORC with R236fa or RRORC with R134a, both achieving 6.55% thermal efficiency.

Pan and Wang [106] offer an avenue to improve the thermodynamic modeling of ORCs beyond using constant isentropic expander efficiencies, namely by using a calculated internal efficiency for a radial flow turbine. The internal efficiency ($\eta_{internal}$) incorporates the influences of nozzle loss (ζ_c), rotor blade loss (ζ_B), exit velocity loss (ζ_b), friction loss (ζ_f), and leakage loss (ζ_y).

$$\eta_{internal} = \left(1 - \zeta_c - \zeta_b - \zeta_B - \zeta_f\right)\zeta_y \quad (8)$$

The most important physical phenomenon this scheme accounts for is the increased friction losses incurred in proportion to increases in the expansion ratio. As such, larger expansion ratios are correlated with lower internal efficiencies and, consequently, lower cycle efficiencies. Since expander performance is a major driver of cycle performance, this expander analysis is an important aspect of determining the optimal cycle conditions under different heat source conditions. In addition, cycle performance results clearly illustrate that the fluids with the lowest thermal efficiencies (e.g., HFC227ea) have the highest power production, whereas the fluids with the highest efficiencies (e.g., HCFC141b) have the lowest power production.

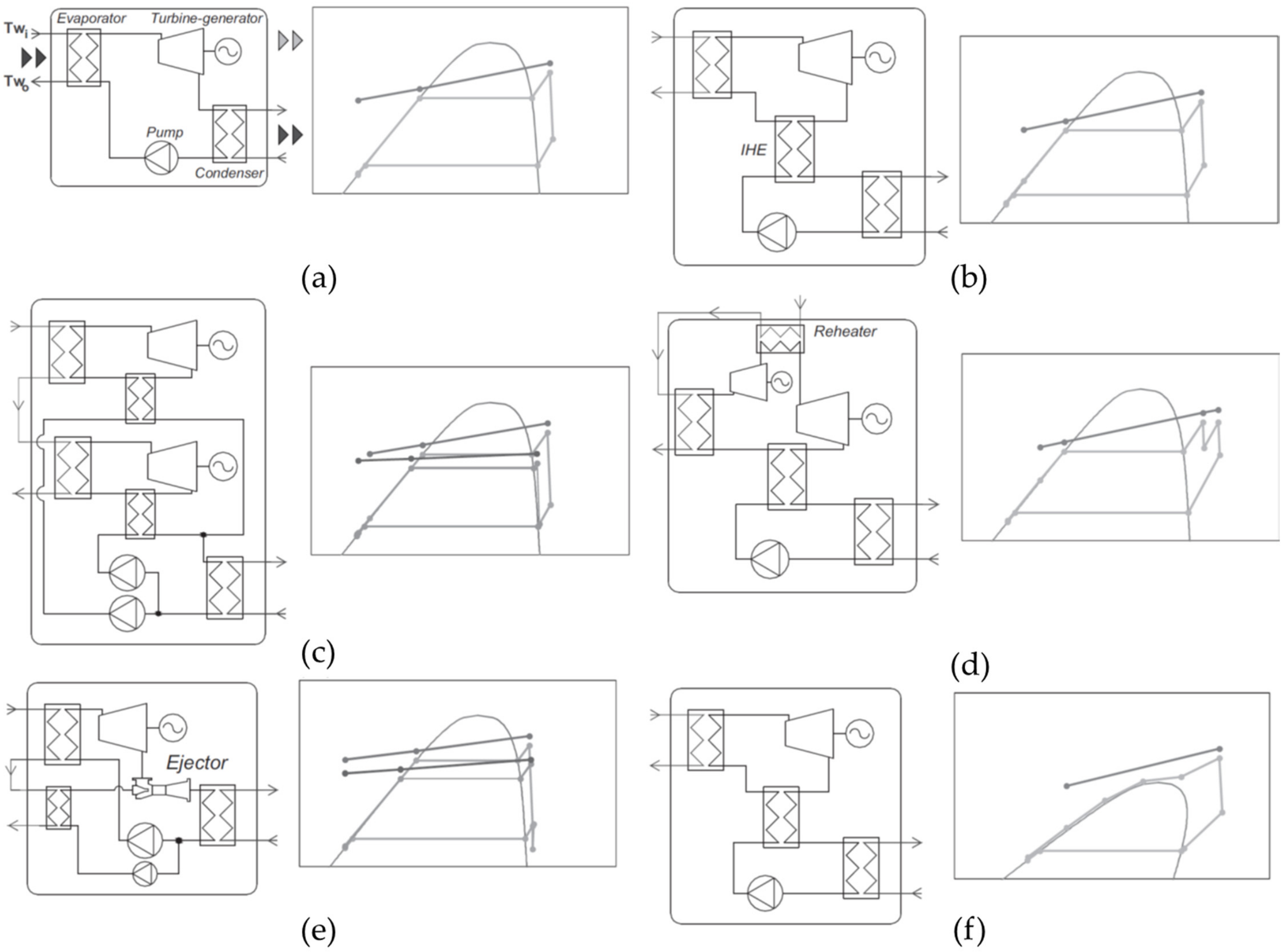

Figure 5. ORC configurations and associated temperature-specific entropy (T-s) diagrams: (**a**) BORC, (**b**) RORC, (**c**) DRORC, (**d**) RRORC, (**e**) EORC, and (**f**) TRORC [5]. Reprinted with permission from Peris et al.; Applied Thermal Engineering, published by Elsevier 2013.

2.3. 2014

Shu et al. [107] offer an expansion of the ORC configurations presented by Peris et al. [5], introducing a new dual-loop ORC (DORC) to recover waste heat from the exhaust and engine coolant of a six-cylinder turbodiesel. (Generally speaking, the use of multiple heat sources expands the locus of possible ORC configurations, and a similar configuration is studied by Bae et al. [108]). As shown in Figure 6, the system consists of two separate cycles: one high temperature (HT) and one low temperature (LT). The HT cycle (the central blue loop) is a steam Rankine, recovering heat from the engine exhaust and rejecting heat to the LT cycle. The LT cycle (the outer pink loop) undergoes three successive heating processes, receiving heat from the engine coolant, HT cycle, and leftover exhaust heat.

Investigation of the overall dual-loop ORC occurs using an Equation Evaluation Solver (EES) model and an energy- and exergy-based analysis with a structured logic-based solution algorithm designed to fix thermodynamic states based on certain chosen parameters (e.g., pinch point temperature differences, isentropic efficiencies, and HT condenser temperature). The authors focus on five engine operating points and evaluate six candidate working fluids for the LT cycle (R124, R134a, R245fa, R600, R600a, and R1234yf). A key result is the preference of R1234yf for the LT cycle working fluid due to having the highest net output power (36.77 kW) and exergy efficiency (55.05%) even though R600 and R245fa have the highest thermal efficiencies (nearly identical at just over 20%).

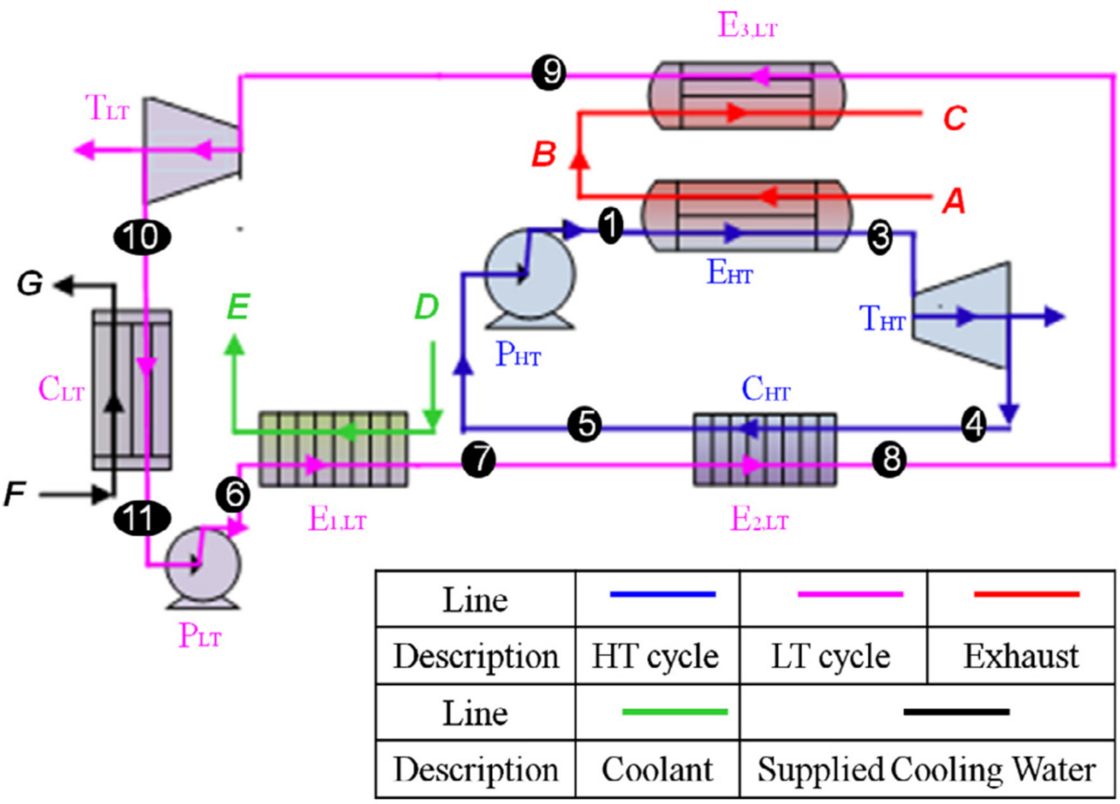

Figure 6. Schematic of a dual-loop organic Rankine cycle using coolant and exhaust heat [107]. Reprinted with permission from Shu et al., Applied Energy; published by Elsevier 2014.

Meinel et al. [109] describe a "two-stage" (or "saturator") ORC concept for WHR from a biogas-fueled ICE. Rather than using the 490 °C exhaust to heat the working fluid directly, the system includes an intermediate thermal oil loop at 240 °C (and 20 bar). Regarding the ORC concept, the two-stage ORC uses partially expanded fluid to "saturate" the partially pressurized working fluid (see Figure 7) in essentially the same style as a regenerative vapor power cycle with an open feedwater heater.

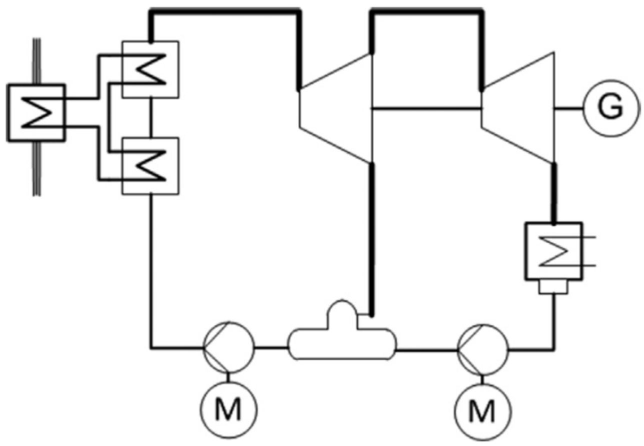

Figure 7. Dual-stage organic Rankine cycle with an intermediate thermal oil loop. [109]. Reprinted with permission from Meinel et al., Applied Thermal Engineering; published by Elsevier 2014.

The two-stage cycle improves the performance of wet fluids (e.g., ethanol) and isentropic fluids (e.g., R245fa and R236ea) by around 2.25% over a standard cycle when using a minimum engine exhaust temperature of 130 °C (to prevent acid condensation) and a minimum pinch point temperature difference (exhaust gas to thermal oil) of 10 °C. (This exceeds the improvement seen with a recuperative ORC (RORC), which is approximately 1.20%). Meanwhile, dry fluids (e.g., n-pentane) benefit more from a recuperator than from using the two-stage configuration.

Shu et al. [110] examine the potential of alkanes as working fluids for engine exhaust waste heat recovery, recognizing that engine exhaust temperatures exceed the temperatures of typical ORC heat sources (and the thermal stability of typical ORC fluids). Linear alkanes (pentane, hexane, heptane, octane, nonane, and decane), branched alkanes (isopentane and isohexane), and cyclic alkanes (cyclopentane and cyclohexane) are included in the study, based on having relatively high critical temperatures, moderate (near-ambient) condensing pressures ("for some of them"), a low environmental impact (zero ODP (ozone depletion potential), and very low GWP (global warming potential). The engine is a supercharged six-cylinder diesel generator, and the WHR system is a basic ORC (BORC). Computer modeling occurs with selected pinch point temperature differences and isentropic efficiencies, with thermodynamic property calculations performed via REFPROP. Among the key results are system irreversibility ($\dot{I}_{ORC}$), the exergy destruction factor (EDF), the volumetric flow ratio (VFR), and the turbine size parameter (SP).

$$\dot{I}_{ORC} = \sum \dot{I}_i \tag{9}$$

$$EDF = \frac{\dot{I}_{ORC}}{\dot{W}_{net}} \tag{10}$$

$$VFR = \frac{\dot{V}_{t,\ out}}{\dot{V}_{t,in}} \tag{11}$$

$$SP = \frac{\left(\dot{V}_{t,out}\right)^{1/2}}{\left(\Delta H_t|_{\Delta s=0}\right)^{1/4}} \tag{12}$$

Tabulated results (see Table 1) show several long-chain alkanes requiring high VFR (above the preferred maximum of 50 [105]) to operate at high maximum pressure, whereas cyclic alkanes (cyclohexane and cyclopentane) are the preferred fluids when accounting for performance and practical indicators. The authors further compare water to cyclohexane, highlighting the efficiency benefits of water and the practical limitations that may cause cyclohexane to be preferable for diesel exhaust WHR.

Table 1. Organic Rankine cycle performance and operational parameters for maximum evaporating pressure across different working fluids [110]. Reprinted with permission from Shu et al., Applied Energy; published by Elsevier 2014.

Working Fluids	$\eta(\%)$	P_{net} (kJ/kg)	I_{ORC}(kW)	$EDF(-)$	$VFR(-)$	SP(m)	P_{evap} (kPa)	T_{evap} (°C)	P_{cond} (kPa)
Decane	18.13	86.17	28.24	1.180	2590.73	1.168	1898.3	337.1	1.52
Nonane	17.95	85.34	29.01	1.224	1012.62	0.761	2041.6	313.4	3.98
Octane	17.67	83.97	30.01	1.287	404.61	0.497	2200.3	287.0	10.45
Heptane	17.10	81.22	31.40	1.392	157.44	0.325	2379.5	257.1	28.04
Cyclohexane	19.32	91.68	29.46	1.157	122.46	0.250	3529.8	269.0	51.90
Hexane	16.09	76.38	33.38	1.574	54.18	0.213	2499.0	221.3	76.42
Isohexane	15.31	72.68	34.33	1.701	42.95	0.189	2526.7	212.0	100.64
Cyclopentane	17.78	84.34	32.30	1.379	32.76	0.165	3313.6	215.3	142.44
Pentane	14.08	66.82	36.64	1.974	17.98	0.142	2581.4	179.3	214.54
Isopentane	13.32	63.22	37.60	2.142	14.55	0.128	2632.8	171.1	273.13

Toffolo et al. [111] opine on the selection of a working fluid and cycle configuration for an ORC operating from common geothermal source temperatures (130–180 °C). While not focused on ICEs, the authors' Heatsep method [112–114] is not limited to any particular type of heat source, so it is briefly mentioned here. The procedure essentially starts with a minimalistic power cycle system and identifies possible heat transfer networks as an objective, rather than being pre-determined, which is a unique aspect of the work.

Lecompte et al. [115] examine a range of zeotropic fluids from the available literature for use in low-temperature ORCs (~150 °C), emphasizing the use of second-law (exergy-based) analysis in evaluation, rather than (or in addition to) the standard first-law analysis. Let us treat this low-temperature study with similar brevity. Using zeotropic mixtures instead of pure fluids increases the recuperative ORC performance by 7.1–14.2%.

Furukawa et al. [116] of Hino Motors detail a hydrofluroether (HFE) ORC for ICE–WHR from a 9 L diesel engine with a two-stage turbocharger. Rather than a completely new design, the ORC system is actually "Generation #2" of an earlier system, allowing a number of upgrades to be enumerated: (1) expanding the temperature difference between evaporation and condensation by replacing offset heat exchanger fins with louvered fins, (2) using a small nozzle angle within the turbine to increase the inlet pressure, (3) adding a recuperator to enhance the cycle efficiency, and (4) adopting a gear pump to replace a leaky, inefficient trochoid pump. A novel aspect of the design is the heat source; the engine coolant leaving the engine undergoes an additional heating process using engine exhaust and EGR heat before entering the evaporator (see Figure 8). Under vehicle conditions described as "cruising on expressway," the engine coolant was 105 °C, and the regenerative ORC improves vehicle fuel economy by 7.5% (versus 3.8% from Generation #1). Combining this result with other efforts by Hino Motors, including downsizing from a 13 L engine, heavy-duty HV truck fuel consumption is reduced by 27%.

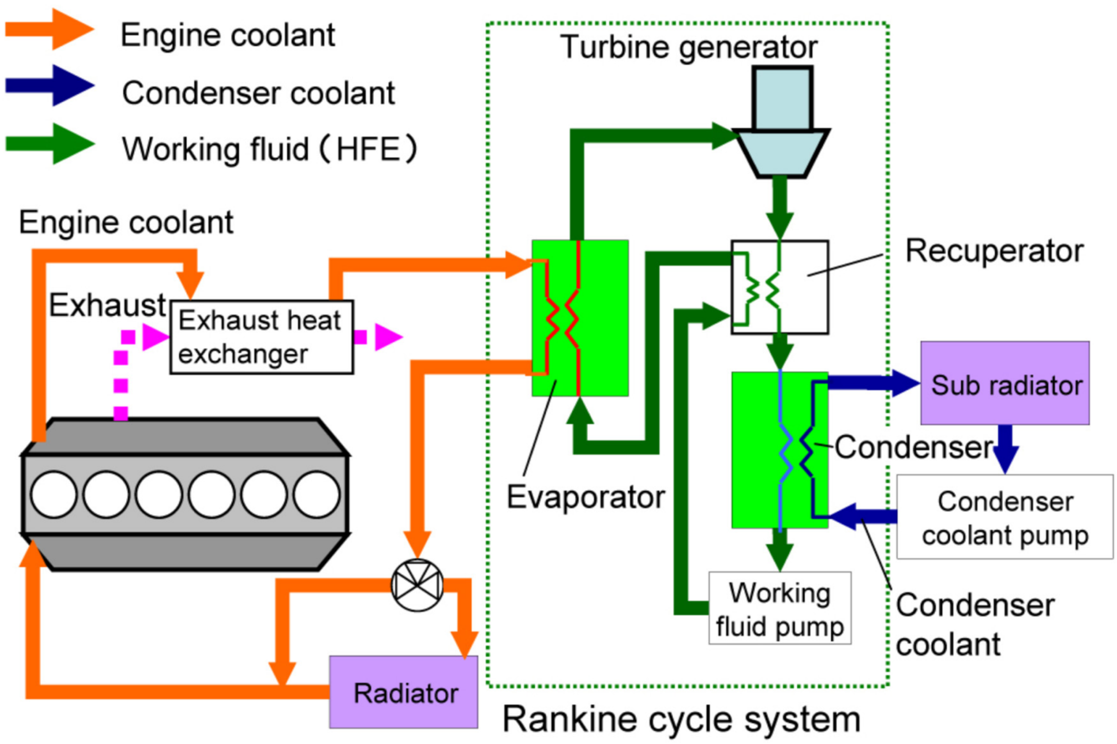

Figure 8. Schematic of engine exhaust (and EGR not pictured) heating engine coolant before passing through a Rankine cycle evaporator [116]. Reprinted with permission from Furukawa et al., SAE Technical Papers; published by SAE International 2014.

Larsen et al. [117] develop regression models to predict the maximum thermal efficiency obtainable using an ORC, predominantly for industrial waste heat in the temperature range of 80–360 °C. Although the work cites ICE-based efforts of Teng et al. [118] and Wang et al. [119], which feature correlations based on working fluid properties, the authors build on correlation efforts based solely on heat source and process parameters [120]. Four correlations are generated using multiple regression analyses with hundreds of optimized study results. Regenerative ORC and basic ORC correlations are developed for two heat source temperature ranges: lower temperature (80–180 °C) and medium temperature (180–360 °C). All four correlations include the source temperature in ($T_{hs,i}$) and out ($T_{hs,o}$) and polytropic expander efficiency ($\eta_{p,e}$), whereas only the medium temperature correlations depend on the condenser temperature (T_c) and pinch point temperature difference (ΔT_{pp}). Perhaps the correlation most applicable to large ICE–WHR is the correlation for regenerative ORCs with heat sources between 180 and 360 °C, which is calculated using Equation (13). Regarding the statistical evaluation, all adjusted R^2 values exceed 0.96, and nearly all *p*-values fall within the range of 1×10^{-30} to 1×10^{-100}.

$$\eta_{th,max} = -12.76 + 0.06428T_{hs,i} + 0.05897T_{hs,o} + 0.2576\eta_{p,e} - 0.1727T_c - 0.1556\Delta T_{pp} \quad (13)$$

Koeberlein [100] provides another Cummins SuperTruck Program update in mid-2014, addressing the overall vehicle-wide project, with modest coverage of the WHR efforts. Two key WHR points are made: that the ORC turbine expander optimization effort is complete, adding an extra 1.8 HP (~13.2 HP to ~15 HP output) in the 34,000–40,000 rpm range, and the heat exchanger architecture is rearranged to preheat the low-pressure loop. These successes provide another 0.7% increase in BTE.

Delgado and Lutsey's [121] white paper remarks on the importance and successes of the public–private partnership represented by the US SuperTruck Program, including all four industry teams (Cummins, Daimler, Navistar, and Volvo). The paper highlights the different strategies of each team in achieving brake thermal efficiency and freight efficiency targets. Pertinently, Cummins and Daimler are pursuing ORCs, whereas Navistar and Volvo are using turbocompounding. Cummins's ORC uses a recuperator and has a "parallel loop" receiving heat from engine coolant and lubricant [122]. Consistent with the Cummins updates, the system is stated to account for a 3.6% improvement in BTE, with a system weight of around 300 lb. The packaging of the system is represented in Figure 9.

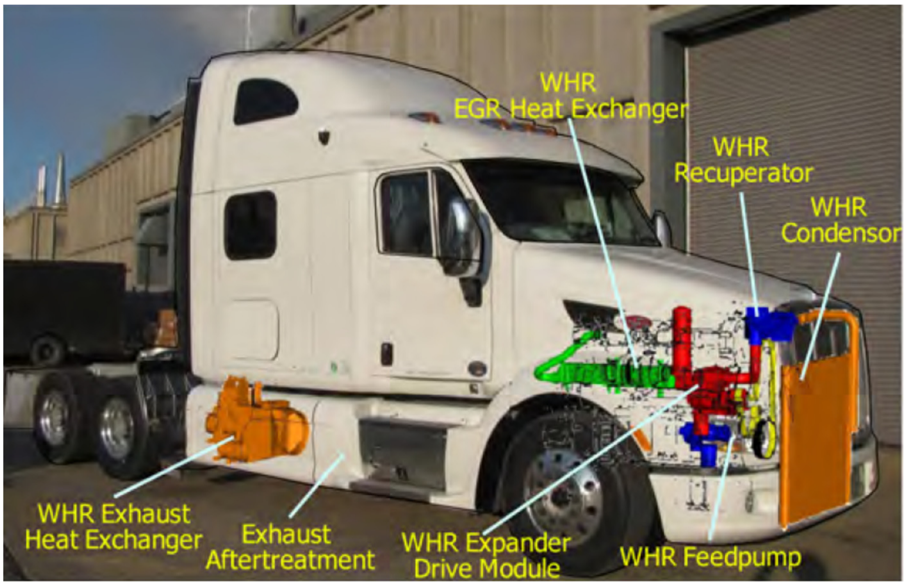

Figure 9. Cummins SuperTruck ORC system configuration in a class 8 truck [122]. Reprinted with permission from Stanton, SAE Technical Papers; published by SAE International 2013.

Daimler's electric-output ORC, by contrast, has a target of 2.5% improvement in BTE. Exhaust WHR with ethanol provides 1.3%, with an additional 0.7% planned by adding radiator and charge air cooler WHR and a final 0.5% from WHR system refinement. The other two industry partners are shown to be considering ORCs as work progresses towards the steeper 55% BTE target (beyond the 50% target at the time of the paper). Navistar, despite not initially planning an ORC, is stated to have an "additional efficiency pathway to achieving 55% BTE. . .” including “. . . possibly an ORC system”. Similarly, “Volvo is planning to implement both turbocompounding and WHR technologies”. Generally, Cummins considers WHR systems “fairly common” for stationary diesels and aims to innovate the systems towards economic viability [123].

Shu et al. [124] describe a “multi-approach evaluation system” (MA-ES), which evaluates competing ORCs in three “ways,” using energetic, exergetic, and economic metrics. While the economic analysis again relies on the “chemical engineering plant cost index” and related parameters, researchers can use any costing approach and employ this method of tabulating the first-law efficiency, second-law efficiency, and cost for competing ORC designs. Moreover, the application of the MA-ES method is detailed for two case studies of diesel engine ORC–WHR, Case A for exhaust WHR (including working fluid selection) and Case B for exhaust + coolant WHR (including ORC configuration selection). Case B, for example, compares the single-loop (subcritical and transcritical) and dual-loop ORC systems, with the dual-loop system excelling in performance and moderately beating the single-loop systems on economic metrics (electricity production costs (EPCs) of 0.67–0.73 USD /kWh and a depreciated payback period (DPP) of 2.8–4.5 years).

Jradi and Riffat [125] provide an example of ORC research expanding on tri-generation. The tri-generation system of the study is biomass-fueled; an ORC provides the heat output and power, and the cooling loop is combined with a desiccant-based dehumidification loop. While the system could operate with an ICE or ICE–ORC, the authors’ system uses the ORC as the prime mover.

Sauret and Gu [126] advance the capabilities of using numerical simulations for ORC turbines, for which high-density fluids pose a modeling challenge. Interested researchers should consult the background work of Pasquale et al. [127] and the historical synopsis of modeling efforts, which show the value of multi-objective optimizations, including the entire ORC, and featuring a detailed turbine model. The effort employs Axcent to develop a full, 3D, solid geometric model of a radial-inflow turbine with subsequent use of the CFD modeling of ANSYS-CFX to examine nominal and off-design operation. The specific study involves the fluid R143a under geothermal conditions within a 400-kW system. Ultimately, the simulation model is capable of matching known performance with over 90% fidelity, and replacing the Peng–Robinson equation of state with REFPROP is identified as an avenue for obtaining additional accuracy.

Hsu et al. [128] also published an expander-focused ORC study, but rather than a theoretical study on radial-inflow turbines, the effort uses experimental means to investigate a hermetic oil-injected twin-screw expander. The expander has a built-in volume ratio of 4.8 and an output capacity of 50 kW. Operation with a volume ratio exceeding the built-in volume ratio is “under-expansion,” whereas a volume ratio below the built-in volume ratio is “over-expansion”. Moreover, since the volume ratio (or expansion ratio) is roughly linear in relation to the pressure ratio [102], increasing the pressure ratio to improve cycle efficiency causes deviations from the ideal (built-in) volume ratio (under-expansion conditions). To an extent, the experimental results support using a slight under-expansion since overexpansion causes sharp declines in expander efficiency. Staying near the built-in volume ratio, with a pressure ratio just below 5, the screw expander is able to achieve 72.4% isentropic efficiency. The authors recommend considering expander efficiency as important while also recognizing that avenues to increase working fluid mass flow rate and cycle efficiency are often more fruitful.

Pasetti et al. [129] examine the thermal stability of three fluids (cyclopentane, isopentane, and n-butane) for operation in an ORC at temperatures between 220–350 °C. A brief clarification: while not all fluid-centered studies are able to be highlighted in this review article, the work of Pasetti et al. [129] is especially relevant due to being ORC-focused, presenting an overview of the history of assessing thermal stability (or thermal decomposition) of working fluids and including an experimental study using a specific survey method consisting of vapor pressure measurements through incrementally increasing fluid exposure temperatures. The experiments involve placing an instrumented test circuit into a furnace for 80 h and subsequently measuring the mass (to detect leakage) and vapor pressure (to detect decomposition). Fluids are exposed to increasing furnace temperatures "until the thermal decomposition of the fluid is reached" [129]. Table 2 shows the primary results, which generally agree with the available literature on these individual fluids.

Table 2. Thermal decomposition results indicating the onset of decay and the associated decomposition rate [129].

Fluid	Maximum Thermally Stable Temperature	Estimated Decomposition Rate
Cyclopentane	275 °C	1.8 %/year
Isopentane	290 °C	1.2 %/year
n-Butane	310 °C	5 %/year

Shu et al. [130] also published an ORC fluid-focused study to conclude this productive year, choosing to address the flammability of hydrocarbons (cyclopentane, cyclohexane, and benzene) in combination with two retardants (R11, R123). These large hydrocarbon molecules are known to offer exceptional performance for engine exhaust WHR, yet at high temperatures, they pose fire risks (in addition to the thermal stability concerns just discussed). Here, the authors aim to ameliorate flammability risks while obtaining the cycle performance advantages of zeotropic mixtures, all while starting at a high baseline performance level. Despite the retardants offering poor ORC performance as pure fluids, cycle performance increases with increasing retardant content up to a certain composition, after which performance decreases, suggesting the presence of an optimal concentration (if flammability has been sufficiently reduced). For example, a 0.7/0.3 mixture of benzene/R11 offers an efficiency 7.12% above pure benzene, with greatly diminished flammability.

2.4. 2015

Grljušić et al. [131] describe the use of an ORC within a CHP plant of a ship for which the main powerplant is a 6S70MC-C7-T1 engine. While Rankine cycles have been used in ships for some time [64,132], this ORC aims to replace a diesel generator for powering ship accessories with a portion of the heat rejected from the ORC being used to provide heating to areas of the ship. The adoption of an ORC, rather than a steam Rankine, is partially based on MAN recommendations for smaller marine engines (15,000–25,000 kW). The authors use a supercritical ORC since the heat source exceeds 240 °C [120] with very similar configurations for R245fa and R123. For the R245fa configuration, fluid heating occurs with the "jacket water cooler," "scavenge air cooler," and engine exhaust in the "boiler," and the 108.02 °C fluid exiting the expander is utilized to satisfy the heating needs of the ship (except heating the heavy fuel oil (HFO)). Interestingly, with the engine operating at a 70% engine load, due to the inclusion of the ORC, the CHP plant electrical efficiency improves by 4.75–5.5%, and the heat + electrical efficiency improves by 9.9–10.5%. While these gains are on par with those of previous studies, the authors specifically note that, at high engine loads, the electrical needs of the ship are not sufficient to necessitate the recovery of a high percentage of exhaust heat, limiting the effectiveness of the WHR system. Another concept involves excess exhaust heat being "converted to electrical energy" to "gain propulsion through the propeller shaft generator" [131].

Kulkarni and Sood [133] design a dual-loop ORC for heavy-duty diesel engine WHR, with the low-temperature loop (using R236fa) recovering heat from the engine coolant and high-temperature cycle condenser, whereas the high-temperature loop (using R245fa) captures engine exhaust heat (see Figure 10). In developing a MATLAB + REFPROP computer model, packaging considerations cause the authors to use subcritical pressures. Analytically, heat exchanger calculations occur using the number of transfer units (NTU) method with specified heat transfer coefficients. Component isentropic efficiencies are 60% for pumps and 85% for turbines. Steady-state analysis assuming a constant pressure operation at five different ESC points suggests a thermal efficiency of 10.3%, with power output varying between ~8.2 and 14.2 kW. This performance is found to be comparable to the literature's results, and the stated next step is "transient analysis".

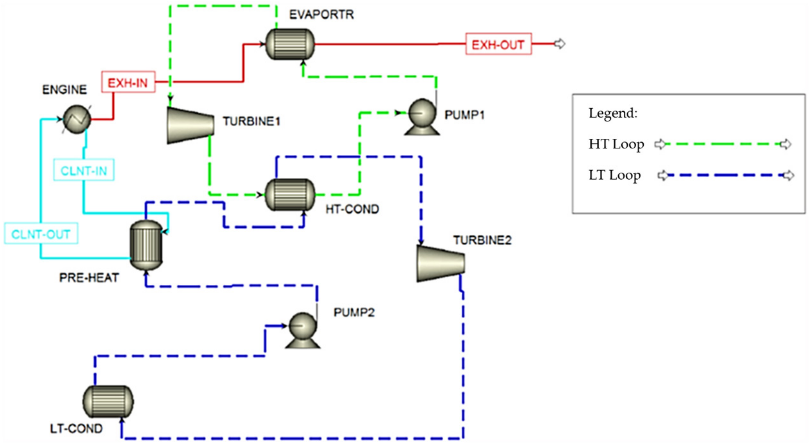

Figure 10. Schematic of dual-loop ORC with an HT (high-temperature) loop for exhaust WHR and an LT (low-temperature) loop for coolant WHR [133]. Reprinted with permission from Kulkarni and Sood, SAE Technical Papers; published by SAE International 2015.

Shen et al. [134] also use a detailed evaporator model in exploring an HDDE–ORC, specifically recovering heat from the exhaust of a 9.7 L diesel engine (280 kW rated power). Central to the evaporator modeling is a multi-zone scheme, using log mean temperature difference (LMTD) equations and heat transfer coefficients obtained through published correlations. With R245fa as the working fluid, a simple ORC without superheat (a two-zone evaporator model) and with superheat (a three-zone evaporator model) are simulated to examine not only the cycle performance but also specifically the physical state of the organic fluid through the evaporator. Regarding performance, the authors observe the ORC with the three-zone evaporator (with superheat) outperforming the two-zone evaporator at a high engine load/speed; however, the two-zone evaporator is preferred under normal operating conditions. Differences in the evaporator are shown to be largely responsible for this, as the mass flow rate of the working fluid is hindered due to the superheating process, especially at low–mid-engine operating points. This occurs despite the authors also noting that most of the heat absorption occurs in the "pre-heating" zone, with the second-most in the "two-phase" zone and the least in the "overheated" zone.

Delgado and Lutsey [36] published another white paper, this time not specifically focused on the SuperTruck program but instead a voluminous forecast of advanced tractor–trailer technologies for the 2020–2030 decade. At the time, US long-haul truck regulations were being slightly tightened for 2017, and regulations for 2020 and beyond were under development, motivating the authors to suggest regulatory structures that accommodate diverse advanced technologies, promoting all economically viable approaches. A West Virginia University collaboration [135] provides engine mapping and a "2020+ with WHR" engine featuring an ORC. Based on a stakeholder workshop and published literature, the authors say that an ORC "can reduce fuel consumption by 2% to 8% in line-haul applications". The authors view ORC–WHR as attractive if heat rejection, packaging, and safety issues are addressed, especially for high mileage and heavy load applications.

Allouache et al. [136] describe a recuperated ORC (using R245fa) with superheat for generating electrical power from the exhaust of a 15 L turbodiesel, a significant scaling up from a previous effort targeting a 1.9 L turbodiesel. Certain performance parameters were carried over from ORNL (Oak Ridge National Laboratory) work on the smaller engine, such as the turbine/generator efficiency of 65%, recuperator efficiency of 13%, and pump efficiency of 39%. While those parameters carried over, the heat exchanger recovering heat from the engine exhaust went from shell-and-tube to finned for compactness. An experimental heat exchanger with a middle section for exhaust flow and top/bottom sections for a working fluid flow is installed on a 6.7 L Cummins turbodiesel at SwRI (Southwest Research Institute). Experimental measurements include the exhaust pressure drop for Reyolds numbers between 800 and 2200, including EGR and non-EGR operation. Optimization efforts on the cold side of the evaporator demonstrate benefits with increased fin density to a substantial extent. In fact, while the experimental heat exchanger features a 92% open flow area, the ORC net power shows a maximum at only around 5% open flow area, below which the cycle suffers from excessive pumping power consumption. Through the optimization efforts, the refrigerant-side heat transfer coefficient appears to increase from ~0.15–0.2 kW/(m^{2}K) to ~9–17 kW/(m^{2}K). An increase in engine power of around 5% is calculated across 12 engine speed/load combinations, showing a similar overall thermal efficiency improvement of 5%. These figures agree with the 4.5% thermal efficiency obtained via ONRL.

Hu et al. [137] investigate the use of variable inlet guide vanes and variable evaporator pressure to find the most effective ORC control strategy. Although the study is focused on geothermal oil production sources (“oil field-associated water” at 90 °C and 10 kg/s), control strategies apply across heat sources, and the study employs the ultra-common fluid R245fa. The variable inlet vane method is termed a “constant pressure operation,” whereas the variable evaporator pressure method is termed a “sliding pressure operation”. The investigation suggests that low heat source flow rates favor constant pressure operation. Optimal ORC performance under fixed geothermal temperatures, across geothermal mass flow rates, is 4.7% for constant pressure and 11.0% for sliding pressure, showing a benefit to varying evaporator pressure under specific source conditions.

Sokolsky et al. [138] published a vision-style article evaluating SuperTruck technologies for “tactical wheeled vehicles”. Despite the JP8 single-fuel policy for military vehicles, certain US-based tactical trucks may burn diesel fuel, making them candidates for the diesel-specific technologies under development by the SuperTruck teams. At the time of the paper, Navistar had foregone its full hybrid design and not built a prototype; meanwhile, Daimler’s prototype was displayed in March 2015 with an advanced aerodynamic design and 12.2 mpg (see Figure 11). On WHR, the authors state that all teams with prototypes include ORCs, with “significant efforts” directed towards that means of improving BTE. In the view of the authors, all SuperTruck ORCs “are appropriate for tactical wheeled vehicles and broad-scale military vehicle deployment. . .” [138].

Figure 11. Freightliner semi on display following SuperTruck program efforts [138]. Reprinted with permission from Sokolsky et al., NDIA Ground Vehicle Systems Engineering and Technology Symposium; Published by NDIA 2015.

Lemmens [139] provides one of the most significant ORC costing articles to date, recapping previous costing efforts and methods and leading to a case study comparing estimated vs. actual costs. Increased numbers of literature studies include economic information, allowing literature data to be charted comparing specific investment costs (SICs) against system power. Waste heat recovery systems are shown to fall principally between geothermal and solar systems, with geothermal systems being the cheapest and solar being the most expensive. Geothermal systems, however, tend to benefit from being predominantly larger-scale. Waste heat recovery systems under 6000 kW appear to have estimated SICs consistently between 2000 and 4000 €/kW, agreeing with Quoilin et al.'s [61] small-scale (< 5 kW) WHR values between 2136 and 4260 €/kW. Different accuracy levels of ORC plant cost estimates are described, for which, in the absence of vendor pricing, researchers may use the classic power-sizing model and subsequently adjust for inflation using cost indices (within approximately 4–5 years [140]). The case study example is a WHR system, although the temperature range is 150–250 °C, and the output power is 375 kW, making the system moderately lower-temperature and larger than typical ICE ORCs. Interestingly, the actual purchased equipment cost (PEC) of 3280 €/kW significantly exceeds the estimated PEC of 1843 €/kW, and the real costs for the "ORC module" (expander, generator, and pump) comprise a much larger (69%) proportion of the equipment costs than do estimated costs (52%). As a final caution, the authors emphasize that the results of the case study are not generalizable and that the current estimating techniques are rough approximations and not substitutes for vendor quotes.

Erlandsson et al. [141] (of TitanX Engine Cooling Holding AB) acknowledge the growth in ICE–WHR interest and thoroughly discuss the parameters and options for heat exchange and waste heat disposal. The study surrounds several components, including a low-temperature radiator (WHR condenser), a vehicle A/C system (with an A/C condenser), a charge air cooler (CAC), a high-temperature radiator, and a fan. Unique to the study is the use of KULI ("a 0D hydraulic simulation tool") to simulate the "underhood cooling airflow" processes occurring on the environment-side of WHR systems. Among the authors insights are recommendations to include sufficient WHR bypassing as an element of maintaining ordinary component function (e.g., sufficient engine cooling) and limit fan work (up to 40% of Rankine power) through the efficient ambient air exposure of heat exchangers (with a minimal added air drag and internal pressure drop).

Apostol et al. [7] offers a broad comparison of ORC configurations for HDDE–WHR, covering six configurations, paralleling Peris et al. [5]. After an initial screening, the authors consider eight working fluids (R245fa, SES36, R1336mzz, MM, ethanol, toluene, n-pentane, and n-dodecane). Using an EES model, with exhaust and coolant heat recovery, noteworthy performance is exhibited by the relatively new fluid R1336mzz. Specifically, the fluid performs well in several configurations, including preheater ORC, regenerative ORC, and 2ORC (as the low-temperature cycle fluid). These performance characteristics are additional benefits to "its positive safety and environmental characteristics".

2.5. 2016

Grelet et al. [142] published a very thorough simulation-focused study of ORC–WHR from heavy-duty diesels, wherein a simplified 0D model is expanded to a dynamic-time-based 1D model using finite volume modeling. In addition to transient conditions, the authors also expound upon the additional constraints present for vehicle-based ORCs in comparison with stationary ORCs. These real-world factors cause Horst et al. [143] to state, in reference to a passenger car steam Rankine cycle system operating dynamically, "while the WHR system could improve fuel economy by 3.4%, restrictions in power output due to the architecture of the on-board electric system, package considerations, increased weight, cooling demand and exhaust gas backpressure lead to a reduction of fuel saving potential by 60% to 1.3%". This constitutes a significant emerging thread of evidence wherein dynamic results lag behind steady-state results by proportions exceeding traditional expectations. Seven phases (also called "driving cycles"), consisting primarily of highway driving comprise the structure of the transient simulations.

The simulation results are partially validated through comparisons to manufacturer data, with further experimental validation necessary against a vehicle-mounted system. While the authors do include water among the working fluid candidates for the study, the preferred fluids are filtered down to acetone and ethanol before ethanol is eventually settled on based on acetone's low flash point. The weighted steady-state performance of two cooling configurations (with and without a dedicated WHR radiator) and four WHR configurations (exhaust-only, EGR-only, exhaust + EGR (serial), and exhaust + EGR (parallel)) suggests that the absence of a dedicated WHR radiator reduces the ORC performance by 11–15%, whereas the additional complexity of the parallel arrangement only adds 4–5%. Importantly, the weighted steady-state performance is shown to overestimate the dynamic performance significantly (a "performance criterion" of 3.46% for dynamic, versus 6.59% for a steady state). As a silver lining, the low system performance is partially due to the lack of optimized system control and component designs.

Cipollone et al. [144] completed an experimental research effort with ORC–WHR using an INEVCO N67 engine. Based on ideal cycle performance, R245fa (Genetron®) is the working fluid, although R1233zd (Solstice®) is considered a strong candidate as a future environmentally friendly replacement. Exhaust heat recovery occurs under increasing engine loads up to 100 kW, for which the exhaust contains 80 kW of heat, of which 55 kW transfers to the working fluid. Running at medium-high duty, the net efficiency is 3–4% with electrical power generation of around 2 kW. While making conclusions, the authors suggest that, considering the unit's mass of 60 kg and increased backpressure, in the context of the overall system cost, the ORC power itself is less valuable than the CO_2 emissions avoided. The authors use a relatively aspirational payback period (termed a "return of investment") goal of 2 years.

Ren et al. [145] offer a unique efficiency-boosting concept of using a solar energy incident on the vehicle to power an ORC system. Although integration into an ICE–WHR system is not described, and the energy density is low, researchers should consider this effort an example of utilizing non-ICE sources that are available.

Arsie et al. [146] cite progress in WHR implementation for heavy-duty applications outpacing passenger car applications as a motivation for their study on incorporating a basic ORC (with R123) into a compact car with a turbocharged SI engine. A "grey-box model" structure uses correlations and detailed heat exchanger equations to study heat exchanger performance, providing calibration for a simpler "black-box model" that could support real-time operation. Component calibration is achieved through comparisons with literature data, including for the system's scroll expander. Control of the ORC is achieved with dual PI controllers (pump frequency and expander rotational speed) as the powertrain–ORC model ran over the NEDC and WLTC driving cycles, with around a 4% reduction in CO_2 emissions over each. The NEDC simulation was perhaps more demanding, as warm-up behavior (especially during the first 500 s) had to be calibrated using test rig data.

Agudelo et al. [147] also advance the research base on passenger vehicle ICE–WHR, for which the powerplant is a diesel engine and exhaust WHR occurs at the outlet of the diesel particulate filter. The study was not specific to ORCs, instead focusing on WHR on an actual vehicle affixed to a dynamometer run over the NEDC with ambient temperatures of -7 °C and 20 °C. These two temperatures represent different modes of engine operation, as extreme cold temperatures allow efficient engine breathing and reduced EGR rates. Over the entire cycle, which takes approximately 1150 s, of which the first 200 s is the primary warm-up period, the authors estimate that a thermoelectric generator (not an ORC) with 4% efficiency could produce useful work at a rate of 2.6–6% of the fuel exergy (offering fuel savings of 8–19%).

Lemmens [148] offers another article on ORC cost estimation, noting that many ORC articles provide cost estimates without accompanying accuracy figures, perhaps in part due to the ORC cost literature only recently expanding. Additional estimated costs (versus system size) appear on a log–log basis plot (see Figure 12), yet most of the data pertain to system sizes larger than typical ICE–WHR ORCs. Costing techniques typically follow chemical engineering strategies, with many efforts employing correlations from textbooks, such as Turton et al. [149]. According to the study's findings, authors should consider "factorial estimation techniques such as the module costing technique" for the best results. In addition, module costs are not exhaustive and require further costing associated with vehicle integration and installation.

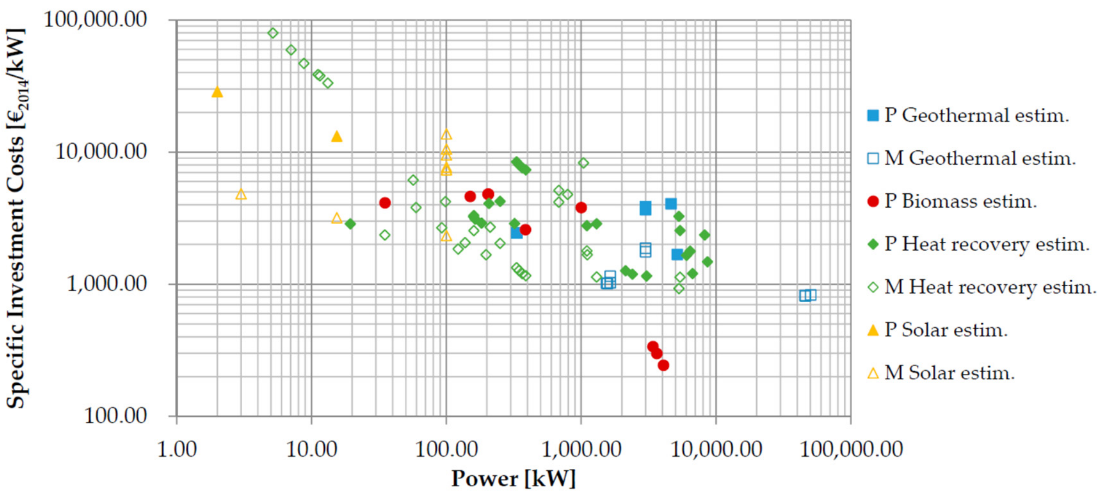

Figure 12. Estimated costs for ORC modules (M) and projects (P) from the available literature [148].

Peralez et al. [150] bolster the literature on vehicle ORC control by describing the control of an ORC recovering heavy-duty diesel exhaust heat through both modeling and experimentation. A point of emphasis is attaining superheated conditions at the evaporator exit to protect expanders without significantly heating beyond that point since superheating reduces fluid flow rates and reduces ORC output. Modeling-wise, the basic ORC (using R245fa) is represented with moving-boundary heat exchanger models and algebraic pump–turbine models. In the experimental setup, the pump is a positive displacement machine, and the expander is a kinetic turbine. To regulate superheating, the authors modulate the pump speed. An evaporator bypass valve aids with evaporator pressure control, while a turbine bypass valve prevents non-superheated flow from entering the turbine. The turbine speed is called out as a parameter available for future optimization of power production. When controlling only the superheat (by altering the pump mass flow rate), the authors enhance the gain-scheduled PID approach by adding a dynamic feedforward term. Through the novel addition of an exhaust bypass, the authors control the evaporator pressure by "combining an additional non-linear controller with an implicit extended Kalman filter (EKF) for wall temperature estimation".

Based on experiments, the superheat controller limits error to 1.9 K, versus 10 K without the dynamic feedforward term. The authors note that tracking system operation is complicated by using the evaporator bypass valve, and applying the EKF to the moving boundary evaporator is computationally intensive, leading to the design of an observer. While further experimental work is planned, the open-loop model has an 8 K superheat error compared to 5 K with the observer.

Desai and Bandyopadhyay [151] compare steam and organic Rankine cycles for power generation based on solar thermal input, which is not directly related to ICE–WHR, yet the authors make important observations about the broader trends of the period. Building on earlier ORC work [152], the authors recognize an increased availability of ORC units and increased cost competitiveness, especially for small–medium-scale production and low temperatures (<150 °C). The authors consider medium–high temperature ranges (>250 °C) to be an area in which both steam and organic Rankine cycles are worth considering. This trend of ORCs becoming competitive at higher source temperatures appears throughout the decade.

Dai et al. [153] offer a thermal stability study focused on using hydrocarbon working fluids in supercritical ORCs, noting that heat recovery from higher temperature sources is sometimes simulated using fluids vulnerable to degradation under evaporator conditions. The authors focus on the mechanism of degradation based on free radicals from the work of Rice and Herzfeld [154], following the steps of initiation, propagation, and termination. A single pathway (out of many) is shown below for N-pentane.

$$nC_5H_{12} \rightarrow C_2H_5\cdot + nC_3H_7\cdot(Initiation) \tag{14}$$

$$\begin{aligned} C_2H_5\cdot &\rightarrow C_2H_4 + H\cdot \\ nC_3H_7\cdot &\rightarrow C_2H_4 + CH_3\cdot(Propagation) \end{aligned} \tag{15}$$

$$CH_3\cdot + H\cdot \rightarrow CH_4\,(Termination) \tag{16}$$

The provided example illustrates the rationale for measuring methane production in the experimental setup using gas chromatography (GC).

Included in the study are the following fluids: n-hexane, n-pentane, isopentane, cyclopentane, n-Butane, and isobutane. All the fluids were obtained from commercial suppliers with purities above 99.87% and tested in a reactor that was evacuated of air to a pressure below 10 Pa. Testing in the stainless-steel reactor apparatus takes place for 24 h, all at 4 MPa (pressure has little influence on decomposition). Fluids are tested at temperatures from 240 °C to 320 °C in 20 °C increments. Three fluids first show decomposition at 280 °C (n-Hexane, isopentane, and cyclopentane), one at 300 °C (n-pentane), and two at 320 °C (n-butane and isobutane). Some of these conditions produce hydrogen in addition to methane, and cyclopentane produces unknown substances larger than cyclopentane (showing recombination). An additional aspect of the work is the introduction of common piping component materials (copper and aluminum) during n-pentane testing to observe any effects on decomposition. No appreciable difference is observed at 280 °C; however, both aluminum and copper increase the decomposition mass fraction by over 10% during a 70 h test at 320 °C.

Ren et al. [155] explore the use of an ORC to recover heat from the oil of a hydraulic retarder in a hydraulic braking system of a commercial vehicle. This unique heat source adds 80–125 °C heat at a rate around 1 kW, keeping the oil cooler and improving braking consistency. Based on the test bench results, fluctuations in oil temperature were reduced by 87%, causing the authors to pursue further study. One lens through which to view this work is by recalling an insight of Erlandsson et al. [147], in which the authors suggest that the numerous heat exchangers in modern vehicles may benefit from consolidation towards two larger heat exchangers, one for a low temperature and one for a high temperature. Gao et al. [156] further demonstrate the use of the working fluid flow rate as a successful mechanism for "cooling the hydraulic retarder transmission oil". Among the key outcomes of the study is the authors' finding that an increased evaporator area produces higher evaporation pressures, allowing larger pressure drops across the expander and consequently enabling more efficient expander operation.

Gimelli et al. [157] perform multi-objective optimization using the (evolutionary) genetic algorithm MOGA (Multi-Objective Genetic Algorithm) II to maximize the electrical efficiency of a large (1 MW) regenerative ORC plant while minimizing the heat exchanger surface area. Interestingly, the authors observe the potential to use surface area as an indirect proxy for system cost since heat exchangers are the primary cost drivers of the system, not the pump and turbine. Thermodynamic cycle modeling and optimization follows the approach of Ferrara et al. [158], and with octamethyltrisiloxane (MDM) as the working fluid, the authors obtain 1000 "DOE solutions," which are filtered down to 40 feasible solutions (26 on the Pareto optimal plus an additional 14 nearby). The "decision variables" number five: the maximum and minimum cycle pressure, the regenerator efficiency, the amount of superheating out of the evaporator, and the amount of subcooling out of the condenser. As previously mentioned, the "objective variables" number two: electric efficiency and total heat exchange area, which ranges from 14.1 to 18.9% and from 446 to 1079 m^2, respectively.

2.6. 2017

Tocci et al. [159] share numerous (data-supported) insights on small-scale ORCs in a "techno-economic review". The authors describe the maturity of the MW-scale ORC market in contrast with the kW-scale ORC market [160], where specific costs (€/kW) for small systems are a persistent challenge. Energy market dynamics, including electricity prices and incentives (especially in the European Union, such as feed-in tariffs), are given as context for ORC viability. Regarding WHR from commercial ICEs, the authors cite the cost concerns of Cavazzini and Dal Toso [161]. Fortunately, the costs of ICEs are relatively low (<€1000/kW) [162], even down to 5 kW [163], potentially allowing slightly elevated costs for the ORC (keeping the combined ICE–ORC cost manageable). Various cost-effective heat exchanger designs are cited from the literature, including helical coil, shell-and-tube, and plate geometries. Taken in combination, existing heat exchanger research and the availability of commercial heat exchangers and pumps cause the authors to focus on expanders and working fluids, with an emphasis on economics (e.g., fluid costs and expander costs in specific size ranges). Along these lines, the authors believe a "reduction in cost of high-speed electric generators together with the use of working fluids that do not require an unreasonably high turbine rotational speed could rapidly make small-scale ORC units cost-effective". Ultimately, the authors also believe additional government incentives are key catalysts for ORCs in the 1–100 kW range. As a specific example, the authors' "business case 2" explores stationary ICE–WHR via ORC, which is not considered "renewable energy" and thus is unable to qualify for various incentives.

Merrett et al. [164] address the accuracy shortcomings of some 1D ORC simulation tools with the development of a Mentor Graphics FloMASTER™ model, validated against the findings of Teng et al. [165] and Park et al. [166]. To enable comparisons, component parameters from published studies are carried over the new model. Mechanistically, engine speed and load feed into the regenerative superheated ORC model traversing the ESC driving cycle. A representation of the ORC and FloMASTER™ model is presented in Figure 13. Across all ESC test points, key thermodynamic properties of temperature and pressure vary by no more than 1.7% and 1.26%, respectively, between the simulation and literature values. To achieve this accuracy, the only calibration that occurs is for the recuperator.

Kim and Kim [167] continue the thread of ORC turbine design previously visited with Sauret and Gu [126], explicitly aiming to learn from those authors and develop a "rigorous preliminary design" tool that is not reliant on the ideal gas equation, average state properties, or performance charts. Part of the authors' motivation comes from the relatively small number of studies describing ORC turbine design compared to the mature field of steam turbine design. In particular, researchers lack access to design tools for which the desired ORC turbine efficiency is input, and the resultant geometry is output. For example, Sauret and Gu state a target efficiency of 90%, whereas meanline analysis finds 76.8%, and CFD analysis indicates an 83.5% efficiency. The analytical code developed by Kim and Kim is named RTDM; it features "the SST turbulent model with the automatic wall treatment" to accurately predict separation and employs the Aungier Redlich Kwong equation of state. To enable performance comparisons, the effort uses Sauret and Gu's design conditions and working fluid (R143a). Upon detailing the geometric parameters for the RTDM code (including an author-selected flow coefficient and blade loading coefficient), the authors accurately project the radial-inflow turbine performance and express interest in exploring more design conditions in future studies.

Mid-year, the US Department of Energy (DOE) issues an update on using ORCs for WHR from reciprocating engines, gas turbines, and industrial processes [168], incorporating project results from Guillen and Zia [169]. Among the innovative aspects of the study was the use of direct evaporation for hot exhaust gases, rather than a secondary heat transfer loop. Challenges to direct evaporation are the potential exposure of the fluid to temperatures sufficiently high to produce thermal degradation and the chance that a fluid leak could occur towards a hot surface, causing a fire hazard. Additional challenges are the stated cost of ORC systems above 2500 USD/kW, meaning "most of the available waste heat recovery opportunities are not economically viable".

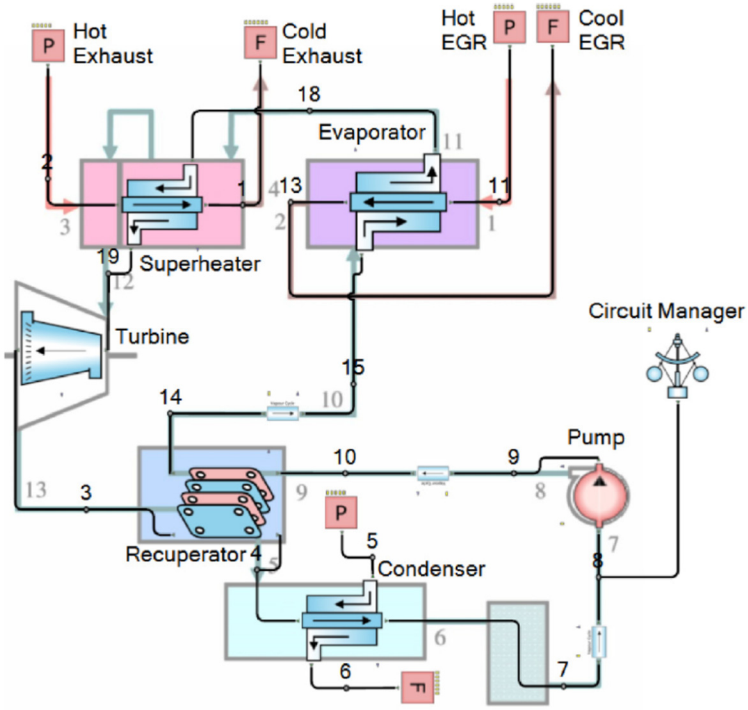

Figure 13. Regenerative superheated ORC schematic with a FloMASTER™ model [164]. Reprinted with permission from Merrett, SAE Technical Papers; published by SAE International 2017.

Subramanian [170,171] of Eaton focuses directly on the ORC affordability challenge with a project aimed at developing a cost-effective heavy-duty engine ORC producing an efficiency improvement of ≥5%. A distinguishing element of the approach is the use of engine coolant as the ORC working fluid and the vehicle radiator as the ORC condenser, with additional heat recovery from the engine's EGR and exhaust. Initially, the author plans to design a roots expander and heat exchangers for coupling to the 13-L PACCAR HD diesel engine; however, cycle parameters (such as the 12-bar maximum evaporating pressure and quality of 0.47) cause the author to consider alternative expanders. Ultimately, the system's estimated potential fuel savings of ~1.5% falls short of the 5% target, with the limited maximum temperature of the working fluid (240 °C) causing a ~1% penalty, the EGR inlet temperature limitation causing a ~1.5% penalty, and the limited heat rejection capacity of the engine radiator providing another ~1% penalty. While the discontinuation of the project after phase 1 prevents a more detailed costing analysis, the system design itself is suggestive of a potential for significant cost savings.

Rech et al. [172] take a predominantly open approach to studying ORC systems for marine applications with four Wärtsilä dual fuel diesel generators (DFDGs). The authors consider single-stage, two-stage subcritical, and two-stage supercritical ORC configurations, all using R245fa. With an EES model for each of the three systems set up to calculate the optimal conditions for key engine operating points, the authors obtain maximum system performance values of 6.5% for single-stage and 12.6% for two-stage supercritical, with annual work production of 1665.8 MWhrs and 2306.6 MWhrs, respectively. So, while the two-stage supercritical system is more efficient, the system must operate in single-stage mode for most engine conditions, leading to less dramatic cumulative energy production improvements. Dynamic modeling in the study occurs with MATLAB Simulink. Based on an examination of the control literature, the authors monitor drum conditions and use PID controllers to dynamically adjust pump speeds. While the single-stage and two-stage supercritical system are controllable with a single controller monitoring a single hot drum, the subcritical system monitors two separate hot drums (low pressure and high pressure) to control the two pumps.

Kuboth et al. [173] highlight the use of experimental results to calibrate and refine the accuracy of ORC computer models using a small-scale (1 kW) test rig targeting industrial WHR, which the authors consider a possible gateway to open further small-scale ORC opportunities. The recuperated ORC using R365mfc is simulated in Aspen Plus, and it features plate heat exchangers, a gear pump, and a scroll expander. The test rig is highly instrumented, including oil and refrigerant mass flow sensors (including a Coriolis flow meter for the refrigerant) and a power meter in addition to temperature, pressure, and differential pressure sensors. Due to time limitations, the authors focus on implementing model improvements for heat exchangers via "empirically determined correlations for the heat transfer rates" and retain the original pump and expander efficiencies (volumetric and isentropic). Using two operating points corresponding to condenser temperatures of 45 °C and 65 °C, the authors find differences between simulation and experimental values, averaging 8.4% for enthalpy and 4.1% for pressure. When the original simulation is replaced with the semi-empirical simulation model, the average difference between simulation and experimental across the two operating points is reduced to 1.0% for enthalpy and 1.7% for pressure.

Another US DOE update [174] follows the completion of a government-funded study, in this case based on the work of Dieckmann [175] targeting the use of scroll expanders for industrial WHR. The expander in the study aims to be compact and cost-effective while also offering simplicity and efficiency, a set of characteristics admirably balanced by the manufacturer (TIAX). For the target heat source conditions between 400 and 1000 °F, the scroll expander can be manufactured from common (non-specialized) materials to reduce material costs and operate in the practical 1200–3600 RPM range. In the laboratory, the duration of the final 20 kWe "scalable scroll expander" testing exceeds 200 h, yielding efficiencies ~50–75% with R245fa as the working fluid. Despite the scroll expander showing minimal wear during these tests, an expander failure occurred after 98 h of full system testing. This precipitates expander upsizing that allows a slower operation at 1800 RPM, after which the system is installed at an industrial facility, for which 21 h of automatic operation are reported. Based on field operation data for volumetric flow, the author deduces that internal leakage of the expander has increased due to the conditions near the inlet of the scroll, where fluid cooling causes some condensation into liquid.

Di Battista and Cipollone [176] published additional experimental work on ORC–WHR from an INEVCO N67 turbocharged diesel engine (6.7 L), further illustrating the lessons available through physical testing. For this effort, the authors use five different load conditions with the engine operating at 1500 RPM, using a basic ORC with R245fa as the working fluid and a single-stage impulse axial turbine expander rated for 7 kW at 80,000 RPM. Particular care is taken to track the energy efficiency of each stage of power production, from the thermal energy of the exhaust to the power production of the turbine, to the electrical power from the variable speed generator and AC/DC converter. The maximum electrical power delivered via the system is 2.5 kW, after roughly 60% of the mechanical turbine energy is lost through the conversion to 24 VDC electrical power (for battery storage). This inefficient energy conversion, combined with difficulties controlling the turbine and modest turbine efficiencies (around 50–55%), cause the authors to express interest in alternative expanders. During the research project, the sizable amount of heat rejection necessary from the condenser leads the authors to formally evaluate the addition of a regenerator to the cycle, limiting the amount of low-temperature heat rejection necessary. This additional component is found to produce another 10–15% output beyond the 2.5–3% efficient system tested experimentally.

Li et al. [177] focus especially on the benefits of a fin-type "preheater" (regenerator) within a "micro vehicular ORC" for ICE–WHR, evaluating the performance of a cycle system with and without a "preheater". The authors use an experimental preheater test setup with 300 °C engine exhaust, as well as a simplified computer model, in the work. With R123 as the working fluid, the authors find a "total heat efficiency" of 23.1% with the preheater and 10.8% without, a significant disparity partly driven by a significant calculated increase in heat exchange within the evaporator. For context, the net ORC output is stated to be 1.74 kJ with the preheater and 1.24 kJ without.

Also, 2017 included a review of on- and off-road diesel-powered vehicles by Lion et al. [178]. The primary focus of the article is "reporting an overview of the considered technology with particular focus on commercial vehicles". Instead of tackling the wide range of ORC efforts, the focus is characterizing the operation of on- and off-road diesels across different applications. Recalling the regulations of Section 1.3.3, the article addresses differences in emission limits between on- and off-road diesels, and differences between city and highway profiles. Importantly, the article wonderfully tabulates profiles of different research efforts, listing each work's ORC configuration, working fluid, component selection, etc. Other broad works include a brief review by Arefin et al. [179] and an important dissertation by Guillame [180].

2.7. 2018

Jiménez-Arreola et al. [181] discuss the thermal fluctuations present in WHR streams used for steam Rankine or ORC power production, primarily focusing on industrial waste heat streams but also covering ICEs, naming those two "the most suitable waste heat sources for power generation". A consideration of published values allows engine exhaust to be characterized as varying between temperatures of 400–900 °C, with mass flow and temperature fluctuations occurring over time scales of seconds to minutes. Road vehicles are an especially dynamic WHR target. Citing another set of studies, ORCs are said to experience moderate efficiency drops at a partial load, making ORCs less sensitive to heat source fluctuations than steam Rankine cycles; however, ORCs are susceptible to working fluid degradation at high temperatures. A unique part of the study is the classification of technologies to manage waste heat fluctuations, branching from the initial two categories of "stream control" and "thermal energy storage". Potential methods of stream control include bypass valves and stream dilution, for which bypass valves divert a portion of the waste heat flow (or working fluid flow) around the evaporator and stream dilution occurs by mixing the waste heat stream with a separate flow stream (e.g., cool air). Alternatively, thermal energy storage includes various sensible heat storage methods (e.g., oil loop, water tank, and molten salt) and latent heat storage methods (e.g., steam accumulator and phase change material). As shown in Table 3, stream control methods are simpler (i.e., better control complexity rating) and less energy-efficient (i.e., worse energy use potential) than thermal energy storage, especially latent storage in phase change materials, and the cost is lower (i.e., better capital cost rating; recall that "-" means a weakness, rather than a lower value).

Rijpkema et al. [182] conduct a comparative study on four alternative power cycles to find the best candidate for WHR from an HDDE. The four candidate cycles are (1) organic Rankine cycles (ORCs), (2) transcritical Rankine cycles (TRCs), (3) trilateral flash cycles (TFCs), and (4) organic flash cycles (OFCs). Compared to the configurations of Peris et al. [5], the organic Rankine cycle here is identical to the basic organic Rankine cycle (BORC), whereas the three other cycles have no exact corollary. The transcritical Rankine cycle merely lacks the regenerator component when compared to the transcritical regenerative organic Rankine cycle (TRORC) of Peris et al. [5]. For the trilateral flash cycle and organic flash cycle, the cycles do not involve performing heat addition to cause working fluid phase change (as occurs in all cycles of Peris et al. [5]). As shown in Figure 14, the TFC simply expands a saturated liquid, whereas the OFC uses a throttling valve to drop the pressure ($3 \rightarrow 4$) and a flash tank to separate the working fluid phases ($4 \rightarrow 4'', 4'$).

Table 3. Benchmarking comparison of technical options for managing thermal fluctuations in heat streams using rating codes of neutral (0), strengths (+), and weaknesses (−) [181]. Reprinted with permission from Jiménez-Arreola et al., Applied Thermal Engineering; published by Elsevier 2018.

Technology	Fluctuation Removal	Additional Volume/Weight	Implementation Effort	Control Complexity	Capital Cost	Efficient Energy Use Potential
Stream Control—Heat Source Bypass	+	−	−	−	− −	−
Stream Control—Heat Source Dilution	+	−	−	−	−	− −
Stream Control—Working Fluid Flow Control	−	0	− −	− −	−	+
Sensible Heat Storage	−	− −	−	− −	− −	+ +
Latent Heat Storage	+ + +	− − −	− −	− − −	− − −	+ + +

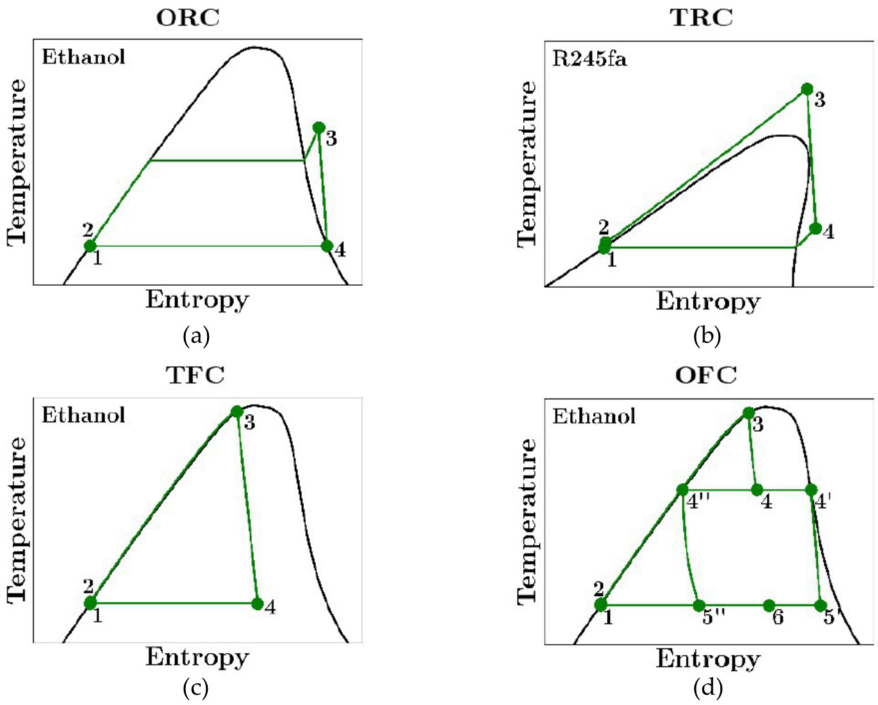

Figure 14. Four cycle alternatives considered for HDDE–WHR: (**a**) organic Rankine cycle, (**b**) trilateral Rankine cycle, (**c**) trilateral flash cycle, and (**d**) organic flash cycle [182]. Reprinted with permission from Rijpkema et al., SAE Technical Papers; published by SAE International 2018.

For each cycle, of the 122 working fluids considered, 56 pass initial screening and proceed into simulations, which are performed using a Modelica code (with Dymola solvers and CoolProp refrigerant properties). Consistent throughout the analysis are the "same boundary conditions and working fluids" for each cycle, with consistent component performance metrics (e.g., isentropic efficiencies) and practical limitations (e.g., condensing pressures at or above atmospheric maximum pressures of 100 bar).

After ruling out WHR from the CAC due to its low energy and exergy content, WHR is simulated on three separate heat sources (engine coolant, EGR, and exhaust) during the operation of a 13 L (six-cylinder) diesel engine at 100 kW and 1100 RPM (between the A25 and A50 ESC modes). As expected, the ORC outperforms other cycles for engine coolant WHR, whereas certain ORCs and TRCs both show strong performance for EGR and exhaust WHR. In the end, the authors prefer the ORC for its lower maximum pressures; because although the authors allowed pressures to 100 bars, a preference was stated to stay below 60 bars and ideally below 20 bars. High-performing ORCs for each source generated

1.5 kW from coolant heat using acetone, 2.5 kW from EGR using cyclopentane and 5 kW from exhaust using methanol (see Figure 15).

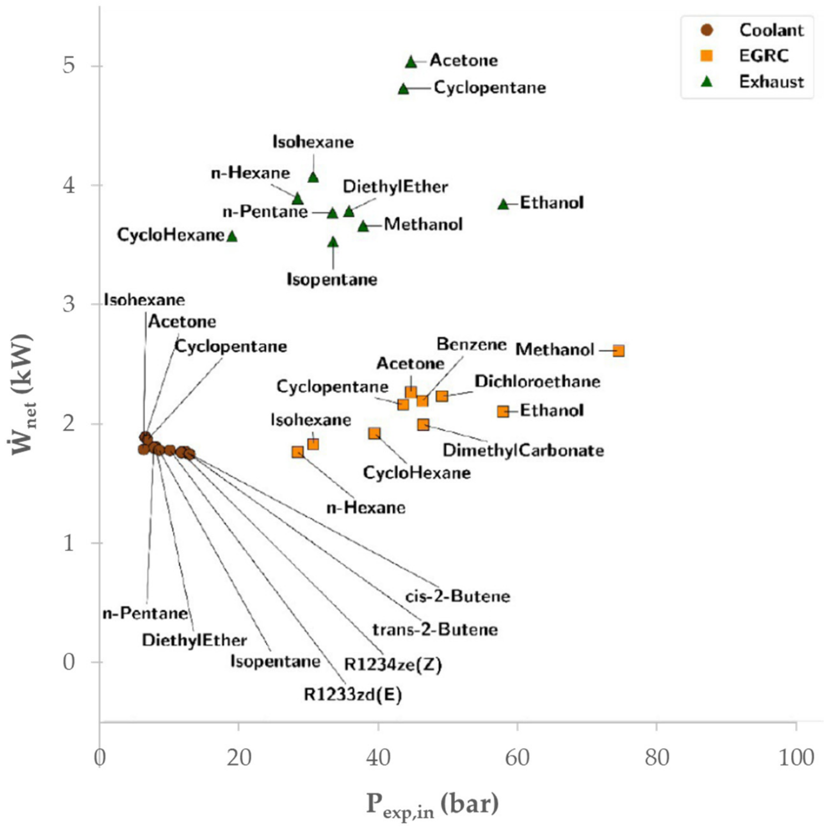

Figure 15. ORC output and maximum pressure for the 10 highest-performing working fluids from each engine waste heat source (engine coolant, engine EGR cooler, and engine exhaust) [182]. Reprinted with permission from Rijpkema et al., SAE Technical Papers; published by SAE International 2018.

Joshi et al. [183] also compare cycles, choosing to compare steam Rankine cycles to two ORCs (R245fa and R134a) for WHR from a diesel generator. The study considers engine generator outputs from 0 kW to 80 kW in 10 kW increments, with the systems being designed primarily for the upper-end conditions (60–80 kW). (Engine generators, in contrast to vehicle ICEs, run at or near maximum output for a significant portion of operating time). As part of their component design, the heat exchangers were crossflow shell-and-tube heat exchangers, with a staggered tube bundle arrangement sized to suit each upper-end condition (80 kW, 70 kW, and 60 kW). Based on simulation results, the steam Rankine cycle performs very well at the highest engine generator outputs yet drops off significantly at lower outputs, with no production at 10 kW or 20 kW. Ultimately, the authors prefer the 20% additional power at 80 kW from the steam Rankine, considering the performance gap above R245fa at 11.2% and R134a at 7.0%. However, concluding remarks emphasize the value of fully accounting for the control needs and weather challenges associated with any working fluid choice.

Kanchibhotla et al. [184] (the same trio of researchers that authored [183]) perform an analogous study on a smaller engine generator with a rated capacity of 26.57 kW. Experimentally determined engine exhaust conditions serve as inputs to the Rankine cycle models for determining potential fuel economy improvements, with water and three organic fluids (R113, R124, and R245fa) being simulated. As before, the authors generate heat exchanger design parameters for particular engine operating points, including the rated power; however, they mention the low flow rates and exhaust temperatures of the relatively small engine generator at low load conditions as an additional motivation for designing this system around a lower operating point. Adjusting the design to suit

21.48 kW did not enable energy production at 5.28 kW for water, yet the principle of making minor sacrifices in top end performance to enable wider WHR operation remains important. At the rated power, water reduced the BSFC by 12.2%, whereas R245fa was the leading organic fluid at 9.6%. Despite the organic fluids performing relatively better in this study, the authors did once again conclude that the system "should be designed with water as the working fluid". In fact, Kanchibhotla and Bari [185] pursue a study with a steam Rankine cycle from the same Toyota 13B naturally aspirated diesel engine with an additional modeling study considering more engine conditions (35), finding support for designing the steam Rankine system for the rated engine power.

Dumont [186] also investigates a steam Rankine cycle, targeting the smaller application of gasoline passenger car engine WHR. ORC researchers may be interested to know that a 1.5 kW axial turbine expander with inlet conditions of 50 °C superheat performed with a maximum isentropic efficiency of 41.5%, falling between a "real scroll expander" at 30% and an "optimal" scroll expander at 60%. The study considers "turbine exhaust temperatures" of 32 °C (ideal), 60 °C (matching a moderate engine coolant temperature), and 90 °C (a high engine coolant temperature).

Chatzopoulou and Markides [187] offer the first study intensively focused on simultaneously designing an ICE and ORC to function optimally together. Structurally, the work is oriented towards developing a simulation tool capable of simultaneous optimization of an ICE and ORC, rather than considering the two units separately. This simultaneous optimization leads to interesting realizations about employing engine operation that may hinder engine performance slightly in order to realize greater gains in ORC performance. For example, the engine exhaust from simultaneous (ICE and ORC) optimization often remains slightly higher, reducing engine expansion stroke power yet allowing the ORC to operate more efficiently.

The study includes three different combined heat and power engines (for stationary cogeneration) named according to the kilowatts of rated power: (1) CHP-160, (2) CHP-230, and (3) CHP-2500. A (subcritical) recuperative ORC was designed for working fluids common in commercial offerings of the time—R245fa, R152a, R1233zd, R1234ze, and R1234yf—as well as four selected hydrocarbons: butane, pentane, hexane, and toluene. Simulations across five operational "cases" for engines, with the energy and component-efficiency-based MATLAB ORC model, provide energy flow, exergy destruction, and efficiency results for the many combinations of engine, case, and ORC working fluid. The optimized engine parameters include the valve size, lift, and timing, while the optimized ORC parameters include the working fluid flowrates, degree of superheat, and evaporation/condensation pressures. The authors observe the most improvement in the medium-sized engines, in which, in one comparison, the addition of the ORC to an unmodified engine improves the system efficiency by 11%, whereas the simultaneously optimized system improves the system efficiency by 21%. These efficiencies are shown to exceed the typical published values for ICE–ORC systems, in large part due to the integrated design approach. As a portion of the overall power output, the ORC contributes 4–15%.

Park et al. [188] conduct a data-driven study examining experimental data from 200+ ORC publications, spanning all types of heat sources (e.g., geothermal, biomass, and ICE–WHR). This work is made possible by a recent increase in publications featuring experimental results, hovering around 10% between 2014 and 2018. While the work focuses on power outputs ≤50 kW, only source temperatures ≤300 °C are considered, excluding most ICE–WHR from engine exhaust. Researchers may still be interested to know that the top research "streams" are design/modeling/analysis (37%), expander/turbine (15%), and optimization (14%), with the next streams being dynamics and controls, both around 3%. The experimental data also yield interesting findings with respect to correlations, such as a weak correlation between the temperature difference ($T_{Hot\ Sink} - T_{Cold\ Sink}$) versus ORC efficiency yet a slightly stronger correlation between Carnot efficiency and ORC efficiency; the average system achieved 43.6% of the Carnot maximum.

Kraljevic et al. [189] describe using hybridization and a steam Rankine cycle to improve vehicle efficiency in advance of 95 g CO_2/km fleet limits in the EU in 2021, potentially with further reductions of 15% by 2025 and 30% by 2030. This important context is based on the Worldwide Harmonized Light Vehicles Test Procedure (WLTP), and it includes the Worldwide Harmonized Light Vehicles Test Cycle (WLTC), replacing the outgoing NEDC. A notable element of the Rankine cycle system is the inclusion of a steam accumulator for managing dynamic fluctuations of the heat source, as well as a bypass valve controlling the portion of exhaust directed towards the Rankine cycle. The authors calculate the most benefit occurring through hybridization, reducing fuel consumption by 10.3%, whereas the steam Rankine system reduced fuel consumption by 1% when initially cold and 4.9% when already hot (at the start of the test cycle).

Hoang [190] offers a lengthy and wide-ranging topical review of recovering heat from diesel engines using organic Rankine cycles. The author addresses emissions regulations and sources of waste heat from diesel engines, offers brief "case studies" mostly focused on recent publications, provides a narrative commentary on working fluid selection (stylistically complementary to the guidelines of Hærvig et al. [191]), outlines ORC modeling (including costing fundamentals), and opines on different ORC configurations and components. Attributes of each alternative component design, such as different pump and expander types, are "gathered and presented". This arrangement is beneficial in helping researchers who are unfamiliar (or less familiar) with ORCs see a literature-grounded perspective on the current state of knowledge in different topical areas.

Lu et al. [192] published a book chapter on vehicle WHR using ORCs, offering some smaller-scale complementarity to the previous (diesel engine-based) article. Although the article does cover historical works pertaining to the development of the technology, the book chapter format also causes the bulk of the article to concentrate on the fundamental concepts of Rankine cycle operation from engine waste heat and the candidate expander technologies, with a generous treatment given to positive displacement expanders. In this way, the authors are not attempting to publish something novel in the sense of capturing the most advanced recent discoveries in the research area, instead emphasizing the clear and accessible presentation of fundamental concepts with a few concluding remarks on the technological "barriers" and prevailing sentiments on the best practices for ORCs for vehicle ICE–WHR.

Galuppo et al. [193] evaluate two competing condensation arrangements for an ORC performing ICE–WHR in a mild-hybrid heavy-duty truck. The first is termed indirect condensation, which accomplishes working fluid heat rejection via a water loop with a separate radiator. By contrast, the second architecture simply uses a working fluid radiator with additional forced convection via an electric fan. Using an internal simulation tool developed by Volvo in the MATLAB Simulink environment, the two configurations are simulated over two highway-based road cycles, the LCG (Lyon-Chambéry-Grenoble—French) and FK (Frankfurt Koblenz—German). Under both condensation arrangements, the ORC produces electrical power for charging a 48V hybrid battery pack. Two competing superheat control strategies are tested, including (1) a multi-model PID (MMPID) controller using "37 first order plus time delay (FOPTD) models" from previous work [194] and (2) a single fixed PI controller using cycle-averaged FOPTD parameters. Chief among the results is the superior performance of the indirect condensation system and the MMPID controller since the direct condensation system's fan consumed up to 500 W of the ORC's power benefit and the MMPID controller kept superheat fluctuations/error significantly lower ($\pm$10 K instead of $\pm$20 K). Further work on subcooling control and the optimization of fan power demand vs. condensation pressure is ongoing, as are the implementation and optimization of a (confidential) overall energy and SOC-level controller.

Alshammari et al. [195] provide an overview of ORC expanders for automotive applications, covering many aspects of expander selection and operation. The study reports on four positive displacement expander types (scroll, screw, piston, and rotary vane) and two turbomachine expander types (axial and radial). For each type of expander, insights are drawn from a strong collection of previous efforts, and formal comparisons are made based on efficiency and power output (see Figure 16). As previously mentioned, turbomachinery clearly operates at higher specific speeds, skewing towards use with larger systems and perhaps slightly higher efficiencies than positive displacement expanders.

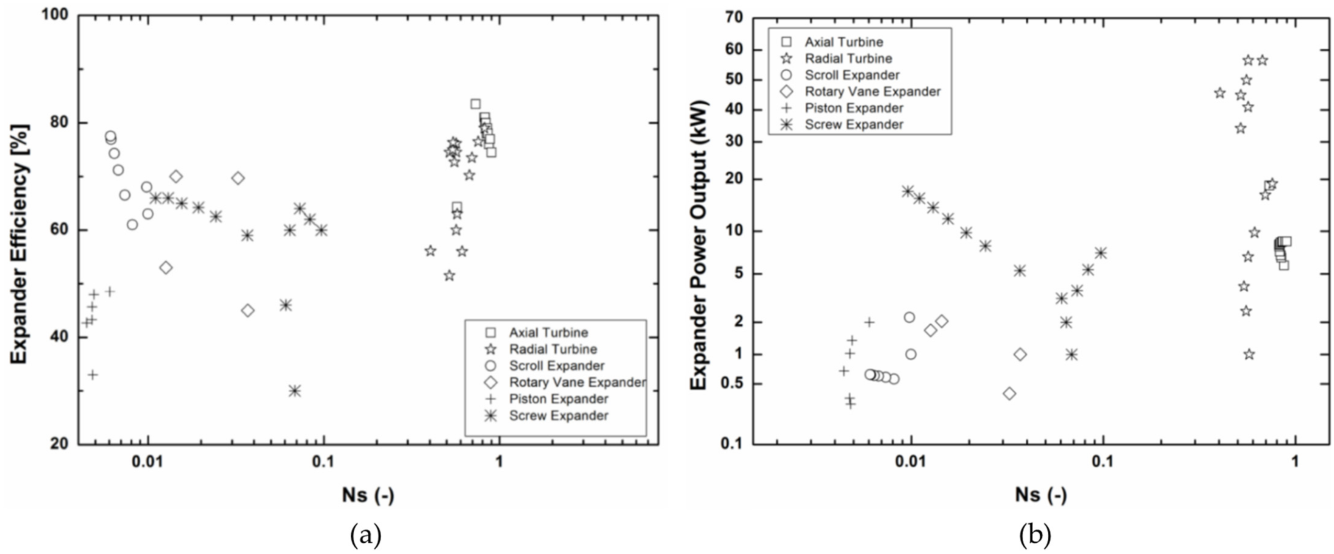

Figure 16. Comparison of ORC expander types for ICE–WHR, including (**a**) expander efficiency across specific speeds and (**b**) expander power output across relative speeds [195].

Moreover, costing analysis is provided, as shown by Figure 17. The basis of these figures is the size-averaging of cost estimates generated via correlations designed for industrial ORC–WHR, rather than specifically applying to vehicular ORC expanders. To the authors' credit, "confidence concerns" based on limited data are cited multiple times within the conclusions.

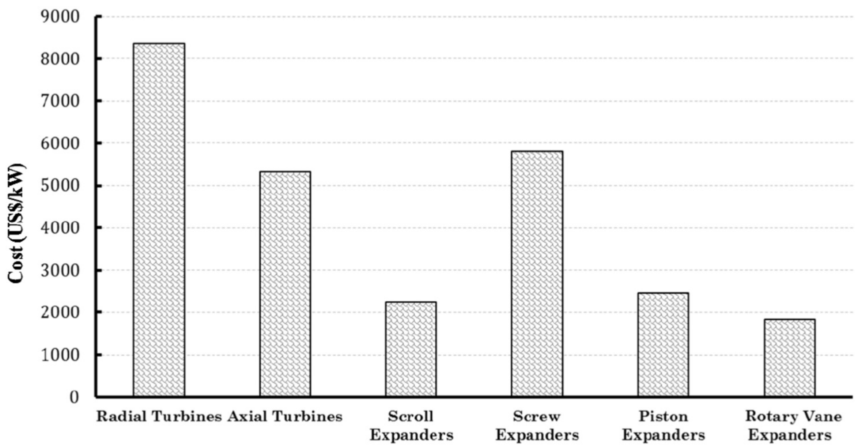

Figure 17. Specific cost comparison between different ORC expander technologies [195].

Kermani et al. [196] give a "superstructure synthesis" for industrial ORC–WHR, with predominantly secondary connections to ICE–WHR. Those interested in the ORC reviews across different industries or pursuing genetic optimizations can benefit from the article (including the appendices).

2.8. 2019

Feng et al. [197] experimentally investigate the impact of the lubricating oil ratio (x) on the performance of a 3 kW ORC. This pivotal parameter is defined as the lubricating oil mass flow rate ($\dot{m}_{lub}$) relative to the sum of the working fluid (R123) mass flow rate $\dot{m}_{wf}$ and the lubricating oil flow rate.

$$x = \frac{\dot{m}_{lub}}{\dot{m}_{wf} + \dot{m}_{lub}} \tag{17}$$

While this parameter often represents the portion of the mixed flow that is lubricating oil, the oil is not always intentionally mixed with the working fluid. A common practice is modifying a scroll compressor to work as a scroll expander, and in these cases, some efforts inject lubricating oil into the working fluid (to limit friction and leakages). Sometimes, an oil separator is included to split the working fluid and lubricant oil, yet in other cases, the authors focus on the impact of increasing the lubricant oil ratio (mixed with working fluid). Lubricant oil ratios of "1.2%, 3.1%, 5.0%, 6.7%, and 9.0%" are examined with different degrees of superheat (5 °C, 10 °C, and 15 °C). Aside from some erratic behavior for the cycle with 5 °C superheat, pump power consumption generally decreases slightly with increasing lubricating oil content, whereas the ORC electrical power output decreases at a near-linear rate. Pertinently, the study uses a heat source of 130 °C, so higher temperature ORCs would have additional considerations around thermal degradation of the lubricant oil. Still, proper lubrication to promote effective and long-lasting component performance is an important aspect of system development addressed in the study.

Chatzopoulou et al. [198] further advance the literature on the off-design operation of ORCs with a CHP investigation using R1233zd (among other fluids) powered using a 1250 kW engine. The study is one of only a few utilizing a piston-cylinder expander, which is shown to perform well during the off-design operation of the ICE, even offering increased isentropic efficiency with certain deviations from nominal engine output (at times exceeding 80%). Performance maps are generated for the heat exchangers, expander, and the entire ORC with four fluids (R1233zd, R245fa, pentane, and toluene). As an illustrative example, the R1233zd ORC operates at 77% of capacity when the engine is down to a 60% load. This part-load ORC performance exceeds other studies, which is a significant finding.

Liu et al. [199] also investigate off-design performance in a study focused on the influence of working fluid charge (undercharged, normally charged, and overcharged). More specifically, the "theoretically rated charged amount" for the 3 kWe ORC prototype is 33 kg, the undercharged amount is 30 kg, and the overcharged amount is 36 kg. The system receives heat from the exhaust of a natural gas burner, yet the temperature level is down in the 100–200 °C range with a 150 °C design value. Using a diaphragm pump and a scroll expander, the ORC performs at a maximum calculated efficiency of 6.57% and a maximum measured efficiency of 4.97% for the overcharged scenario, whereas the normally charged and undercharged efficiencies were comparable (skewing lower). Simulation results for the normally charged system slightly exceed the undercharged system, whereas the experimental results show little difference. The one experimental difference for the undercharged system is the detection of cavitation in the pump, a concerning observation partially attributable to a low condenser fluid level.

The US Department of Energy's Office of Energy Efficiency and Renewable Energy [200] reports on the need to develop a different type of CHP system made possible via the higher-temperature operation of ORCs. The paper suggests increasing the temperature of ORC heat rejection to match building heating needs (~185 °F), rather than using engine waste heat for building heating,. The project, commenced in 2018, involves a high-pressure expander based on a screw compressor from BITZER. Moreover, the effort involves ElectraTherm, a company with significant ORC experience [201].

Rathod et al. [202] attempt to overcome the challenge of controlling ORCs downstream of heavy-duty diesel engines with a look-ahead model. The authors include measurements from a 13 L heavy-duty diesel engine and an ethanol-based ORC, for which experimental data aid in validating physics-based ORC component models. Forecasting vehicle speed from road topography and V2V (vehicle-to-vehicle) connectivity, two non-linear model predictive control (NMPC) methods undergo evaluation, one with future exhaust conditions and one without. For the interior states of the ORC, estimation occurs using an extended Kalman filter (EKF). The NMPC with future exhaust conditions marginally outperforms the NMPC without future exhaust conditions, with the main benefit being more stable operation of the ORC pump.

Imran et al. [203] published an optimization-themed study on using an ORC for HDDE–WHR, employing a genetic algorithm to maximize power output while minimizing cost, volume, and mass. The system being optimized recovers exhaust heat from a 13 L Euro 6 diesel engine without EGR explicitly designed for two cases: condensation temperatures of 40 °C and 60 °C. Building on the authors' significant experience with thermo-economic optimizations, historical thermo-economic optimizations are outlined to provide context for the study's choices of objective functions and decision variables. Based on the historical context, the study is unique in using mass and volume minimization as additional criteria. Specifically, the objective functions are net power, mass, volume, and cost, with decision variables of evaporator temperature, degree of superheat, and evaporator pinch point temperature difference.

Seven working fluids (ethanol, MDM, MM, pentane, R1233zd, R245fa, and RE347mcc) receive consideration for the ORC, which has the configuration of a basic ORC (BORC) with a storage tank before the pump and bypasses for the evaporator and expander. Simulation results for nominal and off-design conditions center on the Pareto frontier optimal solutions, with pentane generally producing high net work at the expense of having the highest total cost (see full results in Figure 18). For example, ethanol produces 10–15% less power at roughly half the cost of the pentane system. (The mass is similar, while ethanol volume is significantly lower for a 40 °C condensation temperature). The authors also highlight the challenge with reducing the condensation temperature from 60 °C to 40 °C since the system with the lower condensation temperature only produces 22% more power (8.53 kW to 10.94 kW) with a corresponding cost increase of 46% (€4552 to €8527).

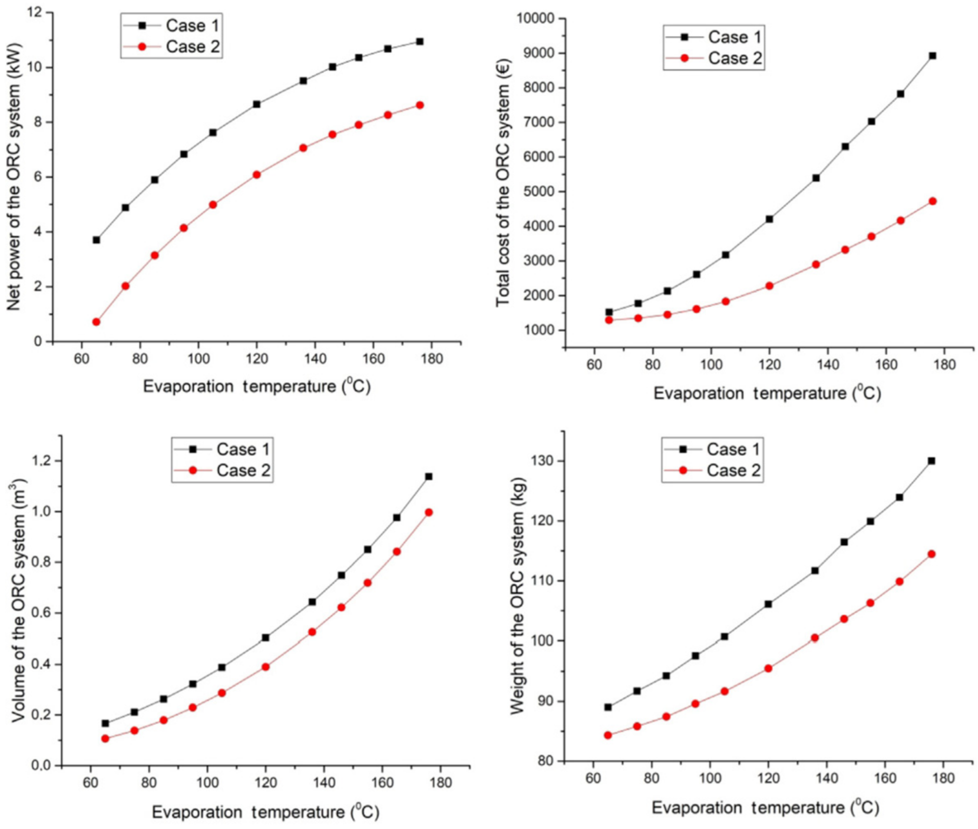

Figure 18. Variations in objective functions across evaporation temperatures for two condensing temperature cases (Case 1 = 40 °C and Case 2 = 60 °C) [203]. Reprinted with permission from Imran et al., Energy; published by Elsevier 2019.

Dai et al. [204] reviews published literature on the thermal stability of organic fluids for ORCs, which is characterized as the "primary limitation for working fluid selection and system design". These working fluid stability subject matter experts have published studies on hydrocarbons [153] and HFCs [205] for supercritical ORCs, with the latter showing HFC236fa to exhibit stability above 375 °C. Although the article is not focused on ICEs, engine exhaust is obviously a challengingly high-temperature source that researchers are interested in capturing. The authors enumerate the fluid categories and offer a highly useful tabulation of the temperature limitations for 30+ common fluids. This type of narrowly focused review can combine with other thermal stability efforts [206] to bolster the informational basis for subsequent working fluid selection works [207].

Simpson et al. [208] advance the development of ORCs for CHP–ICE systems, potentially helping systems qualify for the CHP Quality Assurance (CHPQA) program in the UK. Unique to the study is the provision of techno-economic optimization through simulations based on "real historical electricity and heating demand for thirty energy-intensive buildings at half-hour resolution". For a basic ORC (BORC) configuration, the model uses low-cost tube-in-tube heat exchangers and a piston-cylinder expander. Continuing an ongoing research thread, the authors consider one configuration in which engine coolant preheats the ORC working fluid and heats the building, whereas another uses a higher ORC operating range so that both the ORC condenser and engine coolant heat the building. For the E375 CHP–ICE, the second configuration costs 1750 GBP/kW with broad benefits across the different buildings studied, with "discounted payback periods between 3.5 and 7.5 years".

De Servi et al. [209] detail the design of a high-temperature turbine for use as an expander in a small-scale ORC, citing evidence that conventional radial turbine designs and the "design rules" used to develop them are not well matched to organic working fluid expansion characteristics (e.g., pressure ratios). The effort lays the groundwork for a 10-kW radial-inflow mini-turbine through CFD-based design and loss analysis, with future experimental plans to integrate the mini-turbine into a siloxane MM ORC system in the Organic Rankine Cycle Hybrid Integrated Device (ORCHID) testing facility.

Ekström et al. [210] report Volvo's progress in developing an ORC for ICE–WHR to fit in the "vehicle tunnel" of two passenger vehicles (an XC90 SUV and an S90 sedan). In-house expertise influences the decision-making for the "demonstrator project," including the development of a three-cylinder axial piston expander connected in place of the A/C compressor on the engine belt drive system. Using ethanol as the working fluid, the ORC recovers exhaust heat from a 2.0 L spark-ignition engine with direct injection across a total of 10 simulated system variants (plus the engine alone). Some systems include electrical-only or mechanical-only coupling, a choice enabled by the vehicle's 48 V hybrid battery pack, while others have an electromechanical coupling. After ruling out several heat exchanger geometries, the heat recovery heat exchanger features twisting helical coils between the exhaust pipe and an additional outer pipe. Although the practical realization of the system is the main focus of the article, further information on the control system is provided by Ripjkema et al. [211] in a separate work. Substantial agreement between experimental and calculated reductions in fuel consumption was present at engine power >20 kW, with reductions in fuel consumption growing from around 5 to 7%. The physical ORC performance dropped off precipitously between engine powers of 18 kW to 23 kW, a phenomenon underestimated with the simulation model.

Pantaleo et al. [212] use numerical simulation and optimization to give the most detailed comparison of reciprocating-piston and single/two-stage screw expanders for use as ORC expanders for ICE exhaust WHR. Citing the global study of Tartiére and Astolfi [213], the authors mention that 65% of the ~400 MW of ORC–WHR capacity worldwide is WHR from engines and turbines. (Geothermal is still the dominant type of ORC system at 2000+ MW). This establishes an existing and growing market, with the small system size suggesting the need for efficient volumetric-type expanders. The source of exhaust gases is a 185-kW natural gas cogeneration engine of Centrica (formerly ENER-G), named E185. A total of 18 working fluids are considered with the piston, single-stage screw, and two-stage screw expanders. One series of simulations optimizes for maximum ORC power, while a second (thermo-economic) optimization targets minimum specific cost. Ethanol and acetone stand out as the two fluids producing the most power and requiring the lowest specific costs, and while the reciprocating-piston expander has the lowest specific costs, the authors lament the lack of technological maturity of such expanders. Based on the authors' analysis, an acetone ORC with a piston expander could have a specific cost of 1630 €/kW and offer a payback period of 4 years for an (avoided) electricity price of 0.13 €/kWh.

Xu et al. [214] provide an ORC–WHR review restricted to HDDE applications, styling the article as a guide to ORCs and diesel WHR, which is a unique approach. The article offers "preliminary guidance for people who are not familiar with the HDD ORC–WHR system and for the people who start developing the HDD ORC–WHR system". As such, most of the coverage is topical (e.g., system architectures, heat exchanger selection, expander selection, working fluid selection, challenges and solutions for ORC optimization, and control strategies), with a smaller portion describing full ORC works (e.g., simulation and experimentation).

2.9. 2020

Köse et al. [215] study the use of gas turbine exhaust to generate power, for which the "triple combined cycle" includes a steam Rankine bottoming cycle recovering the higher temperature exhaust before an ORC recovers heat further downstream. This type of system is clearly capable of protecting organic fluids from direct exposure to exhaust temperatures, and it benefits from the relatively high efficiency of steam Rankine cycles, yet the system complexity suggests that the first-line engine applications would need to be relatively large. The gas turbine–steam Rankine cycle–organic Rankine cycle (GT–SRC–ORC) system attained a thermal efficiency of 47.65% with R141b as the ORC working fluid, which roughly parallels an ICE–SRC–ORC system (or even an ICE–ORC–ORC with separate ORCs to recover mid-high- and low-temperature WHR).

Talluri et al. [216] experimentally use Tesla turbines as expanders for micro-ORCs. The authors choose the working fluid R1233zd and mention industrial waste heat and ICE exhaust as small-scale waste heat sources. Unique to the study is the use of organic fluids with Tesla turbines, in contrast to most ORCs using different expander types and most Tesla turbines using air. Despite the experimental device having a modest shaft efficiency around 10% and an adiabatic efficiency around 30%, the effort develops a 2D computer model that agrees with experimental results, and various losses (e.g., disk edge losses) are accurately characterized.

Bari and Loh [217] design and optimize a steam Rankine cycle for WHR from a very large 1.1 MW CAT G3516 diesel genset. A main objective of the study is to design the Rankine cycle by selecting operating parameters and sizing components, allowing the weight of the system to be tracked while evaluating the system against a target efficiency improvement of 10%. While the authors acknowledge that further work is necessary to finalize the design, system performance is improved by "approximately 12.06%".

Le Brun et al. [218] build on the work of Simpson et al. [208] by examining the potential of ORCs for engine WHR within CHP systems, specifically for UK supermarkets. An intermediate water loop receives heat from the "jacket water" and "flue gases," which is used for building heat and heating the organic fluid. A unique aspect of the study is that the ICE and ORC design varies between 30 different supermarkets, ranging from 2000 m^2 to 6000 m^2, fitting a store size range for which a payback period of 6 years is attainable. Through single and multi-case studies, payback periods down to 4 years are found, yet the ICE itself is the primary economic driver of the system, and some supermarkets require long payback periods of 9 years.

Dickson and Damon [219] provide an annual merit review (AMR) update on SuperTruck II program efforts (initiated in October 2016) by Cummins/Peterbilt (following a final SuperTruck I update in 2015 [101]), in particular pertaining to progress towards the powerplant target of 55% brake thermal efficiency. WHR remains a key component of the overall powerplant, recovering heat from coolant, EGR, exhaust, and charge air. At the time of reporting, the final turbine build was complete, and final optimization was in progress. Due in large part to the COVID-19 global pandemic occurring at the time, the authors expected a three-month delay in achieving the 55% target due to “supplier and financial constraints delaying project”.

A broader DieselNet [220] article following the AMR, covering the progress of all five SuperTruck II teams (Daimler, Volvo, Cummins/Peterbilt, Navistar, and PACCAR), explicitly notes that the COVID-19 pandemic will likely cause project delays, such that projects cannot realistically be completed by the initial completion date. For the first four teams, that date is late 2021, whereas PACCAR joined the project later and has a 2022 completion date. Among the notes on Daimler is a statement that the existing cyclopentane-based ORC will be replaced with a “phase change engine cooling system with a piston expander” in which the working fluid is a 60/40 water/ethanol mixture. Expander inlet conditions are anticipated to be 50 bar and 305 °C, achieving a significant 3.5% BTE improvement. Volvo’s WHR efforts are listed as “simulated—validation ongoing,” and PACCAR has plans for a 51% BTE engine with the remaining 4% from WHR.

Imran et al. [221] supplement their multifaceted optimization-based article [203] from the previous year with a review of “dynamic modeling and control” methods for ORCs. This review is predominantly a narrative-style critical review that very effectively describes the modeling and control landscape for ORCs, going well beyond previous efforts in the area. Due to the dynamic operation of most engines, ICE–WHR applications constitute a majority of the review’s references. Recognizing heat exchangers as the primary drivers of dynamic system response, emphasis is placed on clearly distinguishing moving boundary and finite volume modeling strategies. Interestingly, roughly equal portions of the tabulated literature studies employ the moving boundary method as the finite-volume method. By independently modeling each section of the heat exchanger according to changes in fluid phase (e.g., vapor, liquid–vapor, and liquid), the moving-boundary method captures the dominant physics with minimal computational effort. Alternatively, the finite volume method offers a discretized solution through the heat exchanger, with additional accuracy at the expense of the computational burden. These attributes are illustrated through highlighting the doctoral work of Desideri [222]. The article also tabulates the alternative methods for pump and expander modeling, which essentially pare down to performance maps and isentropic efficiencies. Although the range of control strategies is significantly more complex, most research efforts are either PID (proportional integral derivative)-based or MPC (model predictive control)-based with a handful of variations within each category. Based on the review, ORC dynamics are sufficiently complex to cause PID-based methods to underperform MPC-based methods, “especially improved model predictive controllers”. Based on the depth and thoughtfulness of the overview, researchers should consider this article a first-line resource.

Dumont et al. [223] provide a review of Carnot battery technology, furthering the discussion of thermal energy storage and its possible integration with Rankine cycles and heat pump cycles. Although the Carnot battery systems typically receive heat from resistance heaters or heat pumps, certain engine and automotive efforts are mentioned in the review, such as the work of Di Cairano et al. [224]. Like the review of Imran et al. [221], the authors effectively synthesize existing research in an area with disparate works. For instance, sensible heat storage is shown to cost 0.1–10 USD/kWh and be capable of storage times from days to months at temperatures around 500 °C. This large price range is shown to depend on the storage medium, system size, temperature, environment, etc., yet at the same time, sensible heat storage is shown to offer comparable temperatures to latent heat storage at lower specific costs, and it offers higher temperatures than chemical heat storage at lower costs [225]. In an emerging field, the authors also offer a perspective on the need for additional development in simulations, model designs with strong part-load performance, and experimental demonstrations of performance and control, accompanied by full cost reporting.

Singh et al. [226] detail the use of a dual-loop Rankine cycle for WHR from a large Scania D13 engine. The authors consider 10 working fluids (acetone, cyclopentane, DME, ethanol, methanol, MM, Novec 649, R1233zd(E), R1234ze(Z), and water) for the separate loops, one recovering coolant heat and the other recovering exhaust heat. As a basis of the WHR study, an instrumented engine provides engine coolant and exhaust properties, enabling the development of a "quadratic multi-linear regression model". Simulating the heat recovery and power generation across 41 operating points motivates the selection of cyclopentane for engine coolant WHR and methanol for engine exhaust WHR. Subsequent WHTC simulations, over 1800 s, show the fuel consumption reductions possible with the dual-loop system, ranging from 5.3 to 9%. The primary driver of the system's productivity is the temperature of the engine coolant, as simulations illustrate a marginal (~0.5%) benefit at 80 °C compared to a ~4% benefit at 140 °C (see Figure 19).

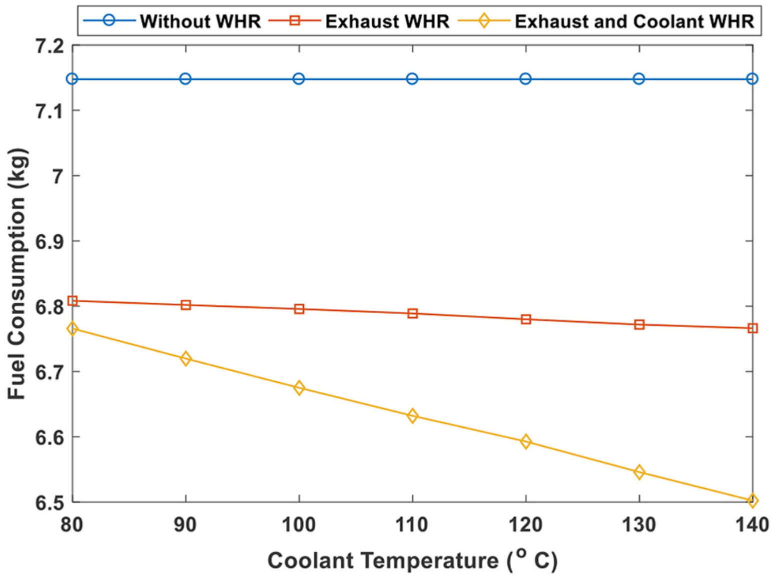

Figure 19. Fuel consumption of Scania D13 engine alone, with exhaust WHR, and with exhaust and coolant WHR [226]. Reprinted with permission from Singh et al, SAE Technical Papers; published by SAE International 2020.

Bin Wan Ramli et al. [227] embrace the trend towards vehicle drivetrain electrification and offer a study on ORC use for ICE–WHR within hybrid drivetrains of passenger vehicles. Simulating in GT-SUITE with the integration of ORC solutions and validated GT-Power engine models, the benefits of the ORC are considered over the NEDC and WLTC, finding a fuel consumption reduction of 1.0% and 1.2%, respectively. Further improvements are expected through the use of different component designs, and presumably through more advanced strategies for battery SOC (state of charge) management.

2.10. 2021

Ren et al. [228] published an article associated with the International Refrigeration and Air Conditioning Conference from the previous year, centering on a 1 kW experimental ORC designed for a maximum heat source temperature of 500 °C. A detailed investigation of the prototype Air Squared scroll pump and scroll expander is the main objective of the study, with the remainder of the ORC constructed from an available microtube evaporator and microchannel condenser, using R245fa as the working fluid. Although different heat source types are mentioned, the study recovers heat from ICE exhaust. The primary results for the novel scroll pump include a maximum pressure ratio of 7.052 and maximum isentropic efficiency ~50% (with most operating points between 30–40%). Regarding the scroll expander, experiments demonstrate a maximum pressure ratio of 5.928 and maximum isentropic efficiency of 60.9% (at pressure ratio 4.03 and rotational speed 3600 RPM), along with a maximum power output of 1.111 kW. Fluid dynamics in the microtube/microchannel heat exchangers generated "intermittent two phase or slug flows," a physical phenomenon identified for further investigation.

Li et al. [229] also published an expander-focused study, exploring the concept of flooded expansion as a means to improve ORC efficiency. Several historical efforts have considered a type of flooded expansion for scroll expanders [230,231], yet this study belongs to a smaller group focusing on screw expanders [232]. The theoretical basis for flooded expansion lies with an isothermal expansion, mimicking an efficiency advantage of the Carnot cycle. However, instead of maintaining a constant temperature via an external heat transfer into the fluid, flooded expansion injects a separate fluid with high thermal capacity during expansion to prevent a temperature drop. For the study, the working fluid is R1233zd(E), and the secondary fluid is SAE20W50 oil, which offers lubrication to the single screw expander while also helping maintain temperature. The regenerative ORC experimental system operates by heating Therminol 66 with an electric heater to a maximum temperature of 340 °C, with "various pressure ratios (3.3–4.1) over the expander and flooding ratios (0–0.3)," across a total of 142 steady-state operating points. The authors observe an expander output increase up to 9.1% with slightly diminished expander inlet conditions and state an interest in testing higher pressure and flooding ratios to prevent overexpansion losses in the future. As a performance-focused effort, seemingly among the first to report the performance of a single screw expander under flooded expansion conditions in an ORC, the authors note "scientific" and "practical" areas for future work. For instance, the authors note the possibility of making computer models based on the experimental results generated in the laboratory and recognize the practical challenge posed by economically performing isothermal expansion using two different fluids.

Alvarez Ojeda and Munoz [233] offer a design methodology for an ORC performing WHR from the exhaust of a small car engine. An overall scheme with three modules is employed, consisting of (1) an ORC model module, (2) an optimization module seeking minimum volume and maximum power, and (3) an interface model transferring data between modules. In line with other multi-objective optimizations, the Pareto-optimal solutions are the target, and in the present work, a range of true physical parameters appear in the ORC model (e.g., expander scroll displacement, parallel pipe evaporator geometry, and piston pump volumetric efficiency). Moreover, for the system architecture, an intermediate thermal oil loop receives heat from the engine exhaust and loses heat to the working fluid. Despite the methodology being intentionally presented as an independent literature contribution that could be adapted to other systems, the authors do generate the Pareto set of optimal solutions, offering 331–1835 W within volumes 0.0139–0.074 m^3.

Kaczmarczyk [234] continues a trend of detailed studies on volumetric expanders with a multiple scroll expander study in the context of a wood-biomass-fueled ORC. The study considers serial and parallel configurations of multiple scroll expanders, with first- and second-law efficiencies both being ~8% greater for the parallel configuration. The 1 kW scale of the ORC in the study is suitable for small engines, whereas the thermal oil temperature at 179–182 °C skews towards the lower temperature range of engine exhaust and above-normal engine coolant temperatures.

Meijer and Grover [235] report the PACCAR team's SuperTruck II progress, based on their unique start date of October 2017 and subsequent disruption due to the COVID-19 pandemic (resulting in a 9-month extension of the third budget period). At mid-year 2021, a major focus of the team is WHR, for which PACCAR attributed 40% of the engine efficiency improvements necessary to reach 55% BTE. Among the available details are the use of a dual-loop ORC with "dual-core two-stage charge coolers" and a "dual-core tailpipe boiler" tailored to the engine. Regarding ORC–WHR, the team notes significant costs associated with high-efficiency WHR systems, yet the ORC is being pursued along with mild hybridization. With hybrid powertrain and initial WHR testing complete, the engine and ORC 55% BTE demo is planned for 2022.

Smague et al. [236] contributed to the International Seminar on ORC Power Systems by detailing a design methodology for heavy-duty truck engine coolant WHR using an ORC. The engines of particular interest use "low carbon content fuels (biofuel or hydrogen)," and the strategy forming the basis of the design methodology is "cost-driven". Hydrogen engines are considered to have a similar engine cooling circuit heat load compared to ordinary diesels. Polymer electrolyte membrane fuel cells (PEMFCs) are also considered, offering higher efficiency (~60%) and rejecting most of the remaining heat to the cooling circuit. For the vehicle and ORC simulator built in Simcenter Amesim, heat exchanger calculations use the "steady-state efficiency NTU method," the pump is assumed to have fixed efficiency, and the turbine performance is calculated as a function of the pressure ratio. Fluid properties are obtained with REFPROP for the NOVEC 649 working fluid. System performance is based on the VECTO long-haul cycle, in which the ORC produces an additional 1–2% of power for the vehicle. The payback time target of the authors is 2 years, and although the general costing outline is provided, the price of hydrogen at the time of the study (>10 €/kg) was significantly different from future projections (4–6 €/kg), leading the authors to tentatively suggest that a split cooling radiator architecture coupled with PEMFCs could be profitable. Also, enhanced ORC performance is expected by means of an optimized turbine and enhanced radiator.

Falbo et al. [237] also use a low carbon content fuel in their study, specifically waste vegetable oil (WVO) biodiesel, within a small-scale CHP system using an ICE topping cycle with an ORC bottoming cycle. The system features thermal energy storage and operates transcritically using the engine's exhaust heat, with one of three working fluids (cyclohexane, decane, and toluene). Constructing a model based on the ORC components, such as the stainless-steel plate heat exchangers (evaporator and condenser), the authors prefer toluene. As a result of the system design, with an ORC cost of 6000 €/kW and a TES cost of 1000 €/m^3, the estimated payback time of 8.4 years caused the system to be adopted for a 20,000 m^3 commercial center.

Chen et al. [238] provide perhaps the first detailed application of "life-cycle assessment" (LCA) to an ORC for ICE–WHR for a CHP system, using LCA as an evaluation method for CHP systems, along with two additional evaluation methods (an equivalent emissions method and a modified techno-economic method). The system under evaluation is a natural-gas-fueled ICE with DHW heating from three sources (engine coolant, residual exhaust heat, and an ORC condenser) and electrical power from two sources (an engine generator and ORC). The system also features a heat pump and uses cyclohexane as the ORC working fluid. Five different processes are included in the LCA, and the associated emissions are calculated for each, as shown in Table 4.

Table 4. Life-cycle assessment overview [238].

Process	Title	Description
1	Material Acquisition	Emissions from obtaining the raw materials necessary for building components and system and obtaining fuel.
2	Component Construction	Emissions from manufacturing individual system components, such as running the equipment and factories.
3	System Construction	Emissions from assembling individual components into a functioning overall system.
4	System Operation	Emissions ongoing due to operation of the system, including any chemical fuel or electrical power use.
5	End of Life Material Recovery	Emissions from returning an expired system to its final state, such as recycling and disposal.

One way of distinguishing between the different methods is to recognize that the life-cycle method tallies primary energy consumption, while the equivalent emissions method gives overall pollutant emissions totals, and the modified techno-economic method gives costs for each system function (electricity generation, domestic hot water heating, space heating water, and cooling). The dominant source of energy consumption and emissions for the system is shown to be natural gas consumption (e.g., 27,080 tons of CO_2), and electricity is the cheapest system output (0.145 USD/kWh). It should also be noted that numerous other LCA methods exist [239], and these types of research rely on previous findings (e.g., the material acquisition energies of Jing et al. [240] and ORC costs of Boyaghchi et al. [241]).

Tian et al. [242] published an effective vision-style article on Rankine cycle (not restricted to ORC) ICE–WHR, sharing several valuable insights. One of the proposals is the use of "thermodynamic perfection" as a metric to ease comparisons between WHR systems and heat sources. Essentially, the method requires dividing the actual performance by the ideal performance, resulting in a percentage of thermodynamic perfection achieved. Based on the literature surveyed, basic Rankine cycles can achieve 54.1% of thermodynamic perfection, whereas cascading and dual-pressure Rankine cycles can reach 62.3%. Additionally, four "research directions" are offered, (1) active working fluid design, (2) cycle configuration design according to energy quality, (3) integrated research across "component, system, and energy management," (4) "advanced coordinative control" [242].

2.11. 2022

Dickson and Meilke [243] update Cummins/Peterbilt progress for the SuperTruck II program, indicating the completion of the engine, WHR system, and 48V mild-hybrid system for final testing. The engine + WHR system achieved 55% BTE on the dyno; however, some optimization over the test route remains to be accomplished. A major advance reported in the annual merit review is the integration of the turbine expander and gearbox with the vehicle transmission, and final testing is planned for the month following the review (July 2022).

Bond and Li [244] give a similar update from Volvo, indicating progress towards an estimated 55.5% BTE powertrain, for which WHR is the largest contributor of efficiency gains. More specifically, the engine coolant ORC undergoing testing at the University of Liège should improve BTE by ~1% and the engine exhaust ORC undergoing testing at the Southwest Research Institute should improve BTE by ~2% (estimated). So, while Volvo has more work to do before final testing, the group continues to progress and ICE–WHR is essential to the group's 55% BTE approach.

Villeneuve et al. [245] of Daimler report the 52.9% BTE achieved with cyclopentane WHR with the phase change cooling (PCC) WHR system showing a 4.4% BTE improvement in simulations, with system construction complete, and testing is "pending but not scheduled". The 60/40 ethanol/water mixture replacing the traditional coolant system is designed to vaporize in the cylinder head and superheat in the EGR cooler. Along with the WHR system, the team also reports future opportunities for ground and vehicle aerodynamics.

Ping et al. [246] share an optimization strategy for a vehicle-based ICE-ORC powerplant, taking into account not only the waste heat source influence but also "the system influence laws of different operating parameters". In simple terms, the authors develop a "vehicle engine-ORC combined system model" using sub-modules that are capable of accounting for various influences, including the vehicle design, road conditions, and driver behavior. The computer model is built in GT-Suite and interfaces with Simulink and MATLAB, providing a digital representation of a four-cylinder turbocharged diesel and a basic ORC with R245fa. Optimization occurs with NSGA-II (non-dominated sorting genetic algorithm II) through the generation of the Pareto frontier and optimal point selection using TOPSIS (technique for order preference by similarity to an ideal solution). Importantly, the simulation tool is developed with the practical target of real-time operation on board a vehicle, and it shows an ability to recognize efficient operating points across different driving cycles, accounting for dynamic waste heat conditions and corresponding ORC lag times.

Basing their view on the GE 7FDL diesel and railroads in Brazil, dos Santos Juvencio et al. [247] give a perspective on evaluating locomotive applications for potential ORC–WHR,. Both engine coolant and exhaust heat are considered, either for independent production or within a "preheating" configuration in which the working fluid undergoes sequential heating processes, first from the engine coolant and then from the engine exhaust. Thus, six potential systems are evaluated with an EES model, two with coolant (R134a and R245fa), two with exhaust (R141b and R245fa), and two with sequenced coolant + exhaust (R141b and R245fa). In the energy-based computer model, the authors chose subcooling of 3 °C, superheating of 5 °C, and a pinch point temperature difference of 5 °C. As a practically focused evaluation study, the authors generate specific ORC costs and use a typical operating schedule to calculate fuel savings, for which a sensitivity analysis shows the internal rate of return for different fuel prices. Although the systems with exhaust WHR (exhaust alone and preheating) provide a positive IRR through the fuel value range, no system is able to attain the goal of 17% IRR to provide a desirable payback period.

2.12. 2023 (Partial)

Bari and Randhawa [248] continue the Australian development of ICE–WHR with steam Rankine cycles, using an EES model to develop an optimal exhaust WHR system from a 25.5 kW diesel engine. Configuration-wise, the Rankine system has a water tank, shell-and-tube heat exchangers (including separate superheater), and a steam turbine expander. Operationally, the system is tested in pursuit of the "maximum efficiency and least cost," with evaporating pressures between 10 and 40 bars and a condenser pressure of 0.3 bars and with pinch point temperature differences above 20 °C. Itemized costing suggests the turbine and controls are the dominant contributors to the 14,440 USD system cost; however, the system offers a modest payback period of 4 years and 5 months under the expected operation.

Cottin et al. [249] of Stellantis and the Université d'Orleans study the potential for Rankine cycle WHR from passenger car engines with series hybrid electric vehicle (SHEV) drivetrains, noting the inherent advantage of mechanically decoupling power production from power use. The authors consider a water Rankine machine (WRM) for high-temperature WHR and an organic Rankine machine (ORM) for low-temperature WHR, both of which benefit from the stabilized operation of SHEV engines (even with multiple operating points). Furthermore, the authors' WHR system architecture also includes an "inverted Rankine machine" for removing heat from the vehicle cabin (evaporation at ~20 °C by car air and condensing at ~15 °C with ambient air). Primarily, the WRM receives exhaust heat into the evaporator and rejects heat into the engine coolant, while the ORM receives engine coolant heat into its evaporator and rejects heat into the ambient air. Based on the dynamic programming control method, the system is able to manage the state of charge of the battery pack using the inputs of engine speed and torque. Fuel consumption is, on average, reduced by 19.4% on the WLTC and 21.5% on a highway cycle.

Downstream of the completion of the SuperTruck II program, a number of press releases are issued via OEMs [250,251], and articles are written by industry observers. Following the attainment of 55% BTE by Cummins/Peterbilt in 2022, Navistar/International details a 55.2% BTE and 16 mpg from their latest SuperTruck in a press release [250]. Based on commentary from an industry expert (Roeth), it seems that this efficiency was obtained without incorporating a thermal bottoming cycle [252], a shift from Navistar's 2021 AMR [253]. (The final vehicle has significant aerodynamic gains and trailer modifications (lightweighting and a solar roof)). Roeth's article states that "all the SuperTruck I and SuperTruck II teams spent a lot of time, effort, and money working on full-scale waste-heat recovery only to find that it is really too complex and too expensive to commercialize". This unique framing dismisses key research results from the SuperTruck II program itself, for which Cummins achieved the 55% BTE target using WHR [243]. As Fischer puts it, "Cummins reported that due to the (waste heat) recovery system and other improvements, the SuperTruck II engine achieved 55% brake thermal efficiency (BTE)" [254]. Moreover, it is unclear whether any team other than Navistar/International achieved 55% BTE during the SuperTruck II program without WHR, and some industry experts believe the Navistar/International truck would have achieved 57–58% BTE with WHR.

3. Review Summary

Commensurate with the large number of research works published over roughly the past decade, several research themes clearly emerged through Section 2. These trends are individually highlighted here and contextualized based on the external factors of Section 1.3.

During the 2012–2014 time frame, the enhanced simulation capabilities facilitated the rapid expansion of computer-based ORC studies. Integration occurred between coding programs (e.g., MATLAB), fluid property software (e.g., REFPROP, CoolProp), engine models (e.g., GT-Power), and vehicle models (e.g., LVS). These powerful tools allow researchers to quickly expand from steady-state simulations across engine operating points to pseudo-steady simulations over driving cycles like the World Harmonized Light Vehicles Test Procedure (WLTP) cycle and eventually dynamic simulations. Especially prevalent are studies on thermo-economic (or techno-economic) optimization, which definitively show that system design and operation for the minimum specific cost (price/kW or price/kWh) is often more desirable than maximum thermal efficiency. As examples, some longer-chain hydrocarbon working fluids have high thermal efficiency but increase system costs due to requiring larger components, while turbine expanders may not offer enough performance to justify the initial investment, especially for small-scale systems. This style of analysis also spawns investigations into a diverse group of heat exchanger geometries and features, including shell-and-tube, plate, tube-in-tube/tubular, spiral, helical, radiator, microtube, microchannel, radiator, finned, and baffled.

In that same period, fuel costs were persistently elevated, engine emissions regulations were tightened on vehicles (e.g., Euro VI [255]), fleet fuel economy standards grew (e.g., China Stage 1 [256] and US CAFE [257]), and HCFC refrigerant use was restricted. On top of research findings supporting long-haul trucks with heavy-duty diesel engines as promising candidates, these external factors fuel additional interest in ORC–WHR from HDDEs. Considering the size of these vehicles and engines, along with the array of waste heat streams present [84], particularly as larger amounts of EGR and EGR cooling become ubiquitous, researchers consider systems with wide-ranging complexity, from basic ORCs (BORCs) to double regenerative ORCs (DRORCs). Working fluids also become more complex, as zeotropic mixtures are aggressively pursued for their potential to match the thermal slide of heat sources through cycle evaporators [79,115]. Also supporting this shift are thermal degradation concerns and flammability concerns; pure organic fluids are typically vulnerable to degradation when exposed to engine exhaust temperatures [129], and natural refrigerants have excessive flammability ratings unless sufficiently mixed with a retardant fluid [130]. While important research contributions come from around the world, the expansive growth of the automotive industry in China and proactive environmental policies of Europe appear to drive research in those regions, whereas US efforts appear to expand more modestly.

During 2015–2017, fuel prices shifted downward, and the regulatory environment was decidedly more relaxed. Global GDP begins to steadily grow, and car sales climbed to a peak in 2017 [258]. In this type of environment, at least in the US, light-duty truck sales expand. ICE applications still target HDDEs, such as the SuperTruck program in the US (for which engineers from many countries are involved), yet market penetration is essentially restricted to stationary ICEs and very large applications like transport ships. At the same time, ORC capacity worldwide exhibits significant growth, largely in connection with expanded renewable energy infrastructure, but also due to research advancements in managing higher operating temperatures, component design improvement, and the control of "improved" Rankine cycles (e.g., regenerative ORCs) [50,259]. While researchers continually advanced the thermodynamic understanding by expanding the use of second law (entropy- and exergy-based) criteria [260], practical advancements in costing and transient modeling were also prevalent in these years [139,148].

The cost efforts of the period built on landmark studies of the early 2010s [61], realizing that authors in the field struggle to find sufficient cost data to justify system cost estimates, especially for exhaust WHR ORCs [261]. This is due to the unique economics of vehicle

ORCs; the heat source temperatures are different from geothermal and solar ORCs, with those source temperatures typically between engine coolant and exhaust temperature levels and ICEs skewing towards smaller scales. Stationary ICE–WHR with ORCs is also decidedly different, as the dynamic operation of the engine is less pronounced, and the system benefit is principally electricity savings, rather than fuel savings. These shortcomings in cost information were lessened through efforts in this period through the compilation of module and system costs at different sizes and nicely packaged cost approaches for individual ORC components [148]. At the same time, more capable simulation tools giving accurate projections of fuel savings during dynamic driving conditions with basic control strategies reveal that dynamic performance significantly lags pseudo-steady performance more dramatically than anticipated [214].

The 2018–2020 period demonstrated a lasting interest in ICE–WHR using ORCs, as the conclusion of the SuperTruck program's WHR efforts seamlessly continued into the SuperTruck II program [121]. In some cases, a working physical system demonstrated enough value to justify optimization and integration efforts, whereas in other cases, WHR appeared to be the most promising avenue to achieve 55% BTE. Researchers tackled the dynamic performance problem in several important ways [181]. Most simply, through first- and second-law analysis, investigations into poor dynamic performance led to the adoption of alternative component designs, including pumps, heat exchangers, and expanders [116]. Off-design performance is improved using and designing components that are insensitive to modest working fluid property variations, limiting the performance penalty associated with varying operating conditions. Also, the use of improved model predictive controllers allows the system to operate more closely to the design point [221]. Furthermore, the design points themselves are selected in a more sophisticated manner through the advent of multi-objective optimizations using genetic algorithms [262]. Many of these efforts admirably forged on despite a myriad of challenges and risks due to COVID-19 throughout most of 2020.

Through the portion of 2021–2023 leading up to today, external influences have been chaotic, posing major challenges to the development of physical prototypes. Particularly, production delays, supply chain bottlenecks, labor shortages, and violent wars have caused many ORC system components to be unavailable or unaffordable. These factors, as well as challenges with packaging, control, and costs, contribute to the noteworthy abandonment of ORC systems by some SuperTruck II teams [250]. For the physical prototypes that are evolutions of previous prototypes, several of which are published during this period, researchers are consistently able to reduce or eliminate the weaknesses of the system. Savings in cost, weight, and space occur from one version to the next, accompanied by improvements in on-design and off-design performance. ORCs are integral to Cummins–Peterbilt being the first team to achieve 55% BTE in SuperTruck II; meanwhile, funding sources are unstable, as the SuperTruck III program is focused on electric and fuel cell vehicles [263]. This significant shift in funding, combined with the trend of formally announcing the phase-out of carbon-based ICE powerplants, and governmental ICE bans (and de facto bans) cause the ORC for ICE–WHR future to be cloudy. Without the expansion of research into cost reduction, advanced integration, and thermal management, the significant potential of ORCs for fuel and electricity savings on new and existing ICEs may dissipate.

Related Reading

Perhaps the largest collection of related ORC work deals with renewable energy, especially solar thermal [145,158,260,264,265] and geothermal [73,76,89,103,266] ORC systems. The literature in this area is too extensive to fully include despite some systems (especially solar thermal) being small enough to be comparable to large engines and temperatures being relevant (typically between engine coolant and exhaust). Moreover, significant research is occurring on thermal energy storage in connection to these renewable energy systems, and those advances may be integral in the pursuit of economically viable approaches to buffering dynamic waste heat source conditions (e.g., engine exhaust) and protecting working fluids from thermal degradation through exposure to overly high heat source temperatures.

4. Results

Key results from the literature review can be organized based on the topics highlighted in Section 3. While the details of the studies presented in Section 2 are crucial and necessary for understanding the scope and design of each study, this section gives a highly streamlined method of post-review accounting for researchers who may be interested in individual portions of the review.

4.1. Working Fluid Selection

Zeotropic mixtures, such as an 88/12 molar mixture of isopentane/cyclohexane, are shown to improve ORC performance significantly at lower heat source temperatures (12.3% at 150 °C) and modestly at medium source temperatures (5.5% at 250 °C) [79]. Binary mixtures of hydrocarbons may improve performance while still not controlling flammability. Hydrocarbon flammability is valuably tempered by mixing hydrocarbons with retardants (e.g., benzene/R11 [130]) and other refrigerants (e.g., isopentane/R245fa [91]), and alcohol flammability is ameliorated using water (e.g., methanol/water [71]). Aside from natural refrigerants like hydrocarbons, other fourth-generation working fluids are also worthy of consideration. Several studies consider R-1233zd [176,187,203], an HFO with a higher critical temperature than R1234yf (439.6 K vs. 367.85 K) and a GWP 99% lower than R245fa. However, the present period is one of considerable transition in refrigerants, with increasing numbers of patented mixtures being explored. As such, broad working fluid selection studies with updated selection criteria continue to be valuable, such as [87,111,191,207]. Most selection criteria are well established, yet the methods of reporting are improving; for instance, the maximum thermal stability is slowly maturing from predominantly being considered a percentage of the critical temperature to having specific experimental evidence based on detecting decomposition products. In other cases, such as condensing pressure, researchers are becoming more attuned to the practical challenge of sub-atmospheric condensation and the performance penalty associated with increasing condensing temperatures to avoid sub-atmospheric pressures. These are merely a small glimpse into the ongoing challenge facing researchers; a greater understanding of the myriad of working fluid requirements leaves a narrow group of practically viable fluids, most of which have characteristics that significantly inhibit efficient cycle operation.

Alongside burgeoning cost control efforts targeting cycle components, a small group of researchers also cite the increasing importance of working fluid costs, especially as working fluids feature more complex molecules and mixtures. Economies of scale, particularly in connection to the HVAC industry, are becoming more favorable to the production of zeotropic mixtures (e.g., R-454b) and pure HFOs (e.g., R-1234yf), although researchers should expect a period of elevated prices due to high demand. Production processes are relatively mature for these working fluids, although in certain cases, there are alternative "routes" to produce the same working fluid, which may involve different raw material inputs and chemical processes [267].

4.2. Computer Simulation Tools

Improvements in simulation tools are made possible through the significantly improved computational power of localized PCs during the decade. Of the surveyed works, the use of external supercomputing capacity is not prevalent, so local computing resources correlate with simulation potential. CFD programs are especially useful in heat exchanger [268] and turbine design [126]; for instance, some authors use SolidWorks for 3D modeling and ANSYS for subsequent analysis. The most widely used simulation tools are self-generated codes and fluid property software (e.g., REFPROP [269]), but increasingly prominent are vehicle simulations over driving cycles [83,227]. Certain innovations are currently taking place in this regard, such as the expansion of VECTO (vehicle emissions calculation tool) to include ORC–WHR [270] and the definition of a more realistic long-haul US driving route [271]. Some authors have demonstrated the integration of engine models (e.g., GT-Power) with other models/codes via Simulink [246]. While these tools are not substitutes for real-world experimentation, authors seek the experimental validation of computer models throughout the literature.

Computer models with artificial intelligence and other technologies like vehicle-to-vehicle communication [202] are essential for real-time control of ORCs in dynamic operating environments. Along these lines, researchers are also moving towards integrating ORC controls with EMSs (engine management systems). That being said, this vision of ORC system development requires persistence and long-term commitment, which have not yet been realized. Let us elaborate on this challenge. Relatively few computer models have been experimentally validated at the ORC level, and even fewer at the ICE + ORC or ICE + ORC + vehicle level. Fortunately, these modeling architectures are well established; however, authors have predominantly chosen not to share or contribute towards open-source modeling codes, leaving researchers facing a choice between adopting commercial code frameworks and enduring long research startup times. Authors should always consider doing a major service to the research community by sharing, at a minimum, any source code not contractually restricted by IP (intellectual property) or NDA (non-disclosure agreement) clauses. By doing so, other authors can adapt and advance the codes towards on-board real-time operation and, upon publication, reciprocate the openness to code transparency.

4.3. Costs

Component and system cost correlations are available in the literature, but researchers must exercise caution for several reasons. Many costing efforts are based on chemical engineering-plant-style fundamentals [124], use industrial components [195], and/or vary widely in heat source and system size [139,148]. So, while the cost literature has significantly grown and improved over the last decade, researchers will still encounter wide-ranging estimates on ORC system costs for ICE–WHR. This large variability applies to cost estimates for individual components and to relative costs between individual components. In certain studies, heat exchangers dominate costs [157], while expanders and controls drive the costs in other studies [248]. A certain amount of variability can be attributed to different component designs (e.g., positive displacement vs. turbine); however, it is also difficult to find consensus on the cost impact of different component designs. Perhaps one of the more discerning approaches is taken by Pantaleo et al. [212], and one of the most widely encompassing costing studies is by Chen et al. [238]. Another important thread of study is cost reduction, for which Subramanian [170,171] explores alternative arrangements (e.g., using engine passages and an existing radiator). Relatively few studies focus on small-scale system cost in the way of Tocci et al. [159]. Among the more optimistic studies, several ORC cost estimates are around 2000–3000 €/kW, whereas others estimate 10,000 + €/kW.

Although the publications of the past decade have included cost information at a greater rate than earlier decades, variations in cost estimates are not the only challenge facing researchers. One possible remedy is an increase in itemized cost reporting and more detailed component design descriptions. For example, product numbers and descriptions for control system components would allow researchers to explore avenues for cost reduction in expensive systems, as well as critically reviewing the suitability of cheaper systems. Similarly, authors are justified in reporting the cost of one-off custom components; however, estimates or quotes for unit production costs for batch production are more relevant to payback analysis.

4.4. ORC Components

Less numerous than overall ORC studies are efforts focused on specific components. Among expander studies, publications examine turbines [126,186,209], scroll expanders [174,175,228,272], screw expanders [128,212,232], piston expanders [198,212,273], Tesla turbines [216], and variable expansion ratio expanders [274]. Similarly, heat exchanger (HX) studies address plate HXs [275], microchannel HXs [276], tube-in-tube HXs [208], helical HXs [159,210], and shell-and-tube HXs [268].

Component-specific findings show slightly higher efficiencies for turbine expanders, on average [195], especially axial turbines. All other types of expanders are capable of efficiencies above 70%, although the ability to achieve that efficiency threshold is less common for piston and rotary-vane expanders. For small systems in particular, the scroll expanders typically offer affordability and low sensitivity to fluctuating heat source conditions (tolerating wet expansion). Displacement-type expanders are also typically more efficient than turbines during off-design operation [198,199], and their lower rotational speeds make mechanical coupling more practicable. Compact heat exchangers are much more prevalent and affordable than one decade ago, assuaging packaging concerns yet still awaiting the cost benefits that arrive after longer-term production. Often, ORC designers are faced with a size-vs.-cost tradeoff; in the same way, they face a size-vs.-performance tradeoff.

A trend apparent in the body of component research is broadening the discussion from efficiency, cost, and size towards optimizing internal geometric parameters (or ratios), improving off-design performance, and handling fluctuations in working fluid inlet conditions. Even more recently, authors are simultaneously optimizing multiple components and reporting traditionally undervalued performance metrics (e.g., warm-up time for heat exchangers).

4.5. Control

As observed by Xu et al. [214], numerous literature efforts report ORC performance improvements achieved via better system control. The methodologies employed to achieve control vary widely, as indicated in the important study of Imran et al. [221]. For simpler ORC configurations and less dynamic operation, pump speed control with relatively ordinary PID controllers can suffice. Alternatively, dual-loop ORCs with unsteady engine operation may require improved model predictive controllers, pump and valve control, fluctuation dampening technologies (e.g., tanks, phase change materials), vehicle-to-vehicle communication, and real-time management via AI.

As previously mentioned, dynamic engine operation without sophisticated and well-tuned control strategies leads to dramatic reductions in ORC output. Efforts to reduce the magnitude of this effect occupied researchers for much of the decade and largely explain the persistent lack of commercially deployed ORCs for WHR from mobile ICEs. Another aspect of the growing prominence of control hardware in ORC–WHR systems is the increased number of experimental research efforts, especially those tied to driving cycle operation, for which ORCs often need to be equipped with valving and additional flow pathways that protect system components (e.g., overly hot exhaust and wet working fluid conditions).

4.6. ORC Performance

Many literature studies report dashboard performance metrics like ORC thermal efficiency, BSFC reduction, ORC net power, etc. Recently, % BSFC reduction has been reported over a certain driving cycle; the WLTP has replaced the NEDC. Others incorporate costs to report a payback period or specific cost. Although no perfect metric exists to override the necessity of studying the experimental methodology utilized in each research effort, Tian's use of "thermodynamic perfection" [242] does ease comparisons.

One mode of progress during the past decade is adding additional clarity on the objective of using ORCs for ICE–WHR. What is the goal? Moving beyond finding a working fluid to provide high thermal efficiency operation at a steady state, the actual goal is an economically viable system based on reducing real-world fuel consumption and emissions. The value proposition is a system that saves a certain amount of fuel (measurable monetarily) and saves a certain quantity of emissions (measurable in various ways, such as tons of CO_{2e} (equivalent), and some locales have a monetary conversion).

Xu et al. [214] find a disparity in performance between simulations (0–60 kW) and experiments (0–14 kW), although again, the trend is towards better controls allowing dynamic performance to more closely match the potential performance suggested with simulations. Along these lines, authors have increasingly accounted for the lost performance during cold start conditions, the drop in expander performance during off-design operation, and the lost heat recovery potential associated with bypass loops designed to prevent working fluid decomposition. As selection of key numerical results of the past decade is as follows:

- The Cummins/Peterbilt's SuperTruck team quickly achieved a 3.6% improvement in BTE with parallel loop WHR by 2015 [101].
- Daimler's SuperTruck II team replaced a cyclopentane-based ORC with a "phase change engine cooling system with a piston expander" using a 60/40 water/ethanol mixture, projecting a 3.5% BTE improvement [221].
- Volvo's SuperTruck II team reports a ~2% BTE improvement via exhaust WHR and a ~1% BTE improvement via coolant WHR [244].
- Payback periods for ORCs for ICE–WHR are represented by Shu et al. [124] at 2.8–4.5 years and Simpson et al. [208] at 3.5–7.5 years. Most studies that include payback periods involve CHP systems.

4.7. Alternative Engines/Fuels

At this point, there are no ICE–WHR studies available targeting the performance of ORCs from HCCI engines, although the prospect of biodiesel HCCI engines (aside from WHR) is discussed by Riyadi et al. [34]. Also, there is a near absence of studies on WHR from ICEs burning hydrogen, CNG, biodiesel, and other alternative fuels. Abedin et al. [32] offers perhaps the most useful overview of the impacts of alternative fuels on engine energy balances, accompanied by a discussion of low heat rejection engines, serving as a general starting point for ORC research.

Specific studies on hydrogen essentially consist of the design-focused coolant WHR work of Smague et al. [236], leaving researchers with the rough preliminary strategy of projecting ORC performance based on hydrogen engine energy studies [10–12,277]. While the general indication is that hydrogen engines are slightly more efficient and have similar waste heat characteristics, further study is warranted. Essentially, the same reasoning applies to CNG engines, for which relatively little ORC–WHR research is available despite being more widely used than hydrogen ICEs.

Among biodiesel studies, Falbo et al. [237] examine waste vegetable oil biodiesel in a fairly large and complex CHP system (compared to vehicle ICEs). Surprisingly few other studies consider biodiesel, so considering the wide range of feedstocks available for biodiesel fuel (which impact the waste heat characteristics) and the frequent need to blend biodiesel with conventional diesel, there are wide gaps in the available research.

5. Discussion

It is immediately apparent from this examination of the recent literature that each year provides a wealth of new and impressive research. Innovations have drawn ORCs closer to adoption for HDDE-powered vehicles, to the point where systems are available online [278]. Perhaps the most obvious validation of the ORC for ICE–WHR concept is the mainstream adoption for stationary power and ship applications, showing viability in the absence of strict packaging limitations and dynamic engine operation. As mentioned previously, compact heat exchanger designs, improved controls, and optimized positive displacement expanders are important steps towards broader adoption.

The research field appears to be lacking affordable and efficient ORC components on the small scale because no systematic reason prevents these ORCs from being cost-effective. The larger-scale ORC–WHR taking place with industrial applications (and small CHP systems) should spread down to the smaller scale, providing more market overlap. The production of fourth-generation refrigerants has ramped up significantly, such as R1234yf being deployed in automotive A/C systems, yet affordable fourth-generation refrigerants with higher critical temperatures (to allow more efficient operation and prevent thermal degradation with exhaust WHR) are essential for ICE–WHR.

There are still relatively few studies exploring WHR from ICE's burning alternative fuels, with only a handful in the past decade. Vehicle-sized ORC component costs are still difficult to price due to the limited data available and challenges applying correlations from other industries. Moreover, almost no studies have attempted to give a detailed environmental account of the benefits of using ORCs for ICE–WHR. Those benefits could provide a firm basis for additional research funding, and a sound critique of governmental policies, by recognizing the widespread lack of incentives for this green technology. Industry, academia, and governments will need to collaborate to create the conditions for the proliferation of ORCs for ICE–WHR.

A simple example is US-based long-haul trucks, for which ORCs have been shown to realistically improve vehicle fuel economy by 3% (with near-term potential of 5%). These vehicles travel an average of around 100,000 miles/year with a high-end fuel economy of 10 miles/gallon, burning 4.50 USD/gallon diesel fuel. Adding the WHR system saves 300 gallons of fuel each year, saving 1350 USD, and preventing the emissions from burning 300 gallons of diesel. Getting that type of system on the road with reasonable subsidies brings the 5% performance closer through economies of scale and real-world refinement. Ideally, these single-digit gains are combined with total-vehicle efficiency strategies and the use of environmentally friendly gaseous fuels (e.g., renewable natural gas).

6. Conclusions

As a final encapsulation of the most salient findings of this review study of the past decade of research on using organic Rankine cycles for internal combustion engine waste heat recovery, the following conclusions are formally drawn:

- The published literature on ICE–WHR using ORCs continues expanding rapidly, and the past decade is further evidence of lasting interest.
- As research efforts mature, the challenges facing the technology become increasingly apparent, especially with regard to developing a system that simultaneously offers affordability, performance, and durability.
- Although not openly shared, the simulation capabilities of researchers have expanded to complex assemblages of ORC, ICE, and vehicle models within control and communication frameworks, showing the capabilities of researchers to integrate thermodynamic properties, computational fluid dynamics, and other commercially available software packages into sophisticated (and highly capable) model architectures.
- Surprisingly few efforts have specifically investigated the potential of using ORCs for WHR from HCCI engines and engines burning alternative fuels (hydrogen, CNG, ethanol, biodiesels, etc.).
- Hardly any studies have seriously investigated the environmental impacts of using ORCs for ICE–WHR, from basic aspects like fuel savings, emissions prevention, and reduced thermal pollution to more thorough environmental impact assessments.
- Further studies are also urgently needed on the topics of cost reduction and cost estimation across various types of systems (stationary, mobile, CHP, and with thermal storage).
- Long-term collaborations like SuperTruck II, especially combining the resources of academia, industry, and government, are necessary to incentivize the deployment of ORCs to real-world vehicle fleets.

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Appendix A. Early (Pre-ICE–WHR) ORC History

Citing the history of steam engines authored by Galloway (with an appendix by Hebert) [279], Colonna et al. recently offered a brief survey of the origins of organic Rankine cycles (ORCs) dating back to 1826 [280]. Galloway's book attributes the commencement of organic fluid use (instead of steam) to lower boiler pressures, alleviating the need for costly specialized operators, as well as the simplicity of multi-use fluids (e.g., naphtha as fuel, lubricant, and working fluid). Attributed to F. W. Ofeldt in the US [281,282] and A. F. Yarrow in Britain (according to [280]), these early external combustion engines powered small boats, receiving heat from the combustion of fuel and rejecting heat to the navigable waters. Later designs, circa the early 1900s, amassed thermal energy using a solar collector for use in pumping water, as described by J. T. Pytlinski [283]. This concept was effectively championed and advanced by L. D'Amelio at the University of Naples through the early-mid-20th century, expanding ORC use into solar ORC power plants and geothermal ORC power plants [284–287] and developing working fluid selection criteria [288].

Eventually, external combustion Rankine cycle engines were developed for space and military applications, including a mercury Rankine power system described by Snoke and Mrava in 1964 [289]. The 1960s also saw clear movement towards vehicle applications, which in the US is primarily documented in the publications of the Society of Automotive Engineers (known as SAE before becoming SAE International). Like NASA (National Aeronautics and Space Administration) technical papers and US EPA (Environmental Protection Agency) publications, SAE publications are sometimes omitted by non-US authors; in the same way, US authors unintentionally fail to recognize all the governmental and professional organizations' publications from other countries.

Table A1 provides a list of publications germane to the development of ORCs yet without being specifically ICE–WHR. This flurry of efforts arose from the development of emissions regulations and the associated research funding programs that were launched, as outlined in Appendix B.

Table A1. List of SAE publications on ORCs before the work of Patel and Doyle [2].

Year	Author(s)	Description	Reference
1969	Bjerklie and Sternlicht	Comparison of steam Rankine and organic Rankine engines to existing Otto engines.	[290]
1969	Bjerklie and Luchter	Characterization of an "ideal working fluid" for development by the chemical industry.	[291]
1969	Degner and Velie	Development of a silent organic Rankine cycle system with monisopropylbiphenyl for electrical power in tactical military applications.	[292]
1970	Doyle et al.	Description of "developments in small reciprocating" ORCs using Monsanto Cp-34 as the working fluid.	[293]
1970	Bjerklie and Sternlicht	Description of ORC applications outside of prime movers, including bottoming cycles.	[294]
1970	Lodwig	ORC receiving heat from exhaust gas of gas turbine generator for military tactical use. (Milestone in the use of exhaust gases for WHR).	[295]
1971	Barber et al.	Vehicle prime mover turbine-gearbox design and testing, with experiments using supplied "organic vapor".	[296]
1973	Reck and Randolf	ORC bus engine design and prototype testing.	[297]
1973	Morgan et al.	Vehicle prime mover ORC development by Thermo Electron Corporation. (Forerunner to the seminal publication by Patel and Doyle [2]).	[298]
1974	Patel et al.	Further detail on ORC as a prime mover with reciprocating geometry.	[58]
1974	Hodgson and Collamore	Rankine cycle prime mover development and preprototype testing using AEF-78 or water as a working fluid.	[299]

Note: Publications exclusively featuring steam Rankine cycles have been excluded, such as those by Kitrilakis and Doyle [300], Burkland [301], Gerstmann and Pompei [302], and Luchter and Mirsky [303].

Appendix B. Overview of Early (1940s–1970s) External Influences

In 1969, near the end of this period, Bjerklie and Luchter indicate that the serious development of ORCs "has been underway for about 20 years," without establishing a "widespread application" [291]. The next year, according to Doyle et al., "during the period 1925–1965, relatively little technical activity was concentrated on Rankine cycle systems for motive power" [293]. On the surface, these two statements seem to be in opposition, yet the authors are using different approaches to signal the shift towards investigating ORCs for automotive applications. Bjerklie and Luchter [291] refer to substantial previous work developing ORC cycle technologies, irrespective of application, while the "cycle is now being evaluated for several automotive uses". Similarly, Doyle et al. [293] focus on the "motive power" application, referring to automotive applications in the midst of emissions legislation more narrowly focusing on the automotive sector.

Appendix B.1. Proliferating US Government Regulations

Legislation designed to address air pollution problems, especially in cities, played a major role early in the 1940s–1970s period. US laws have the largest influence on early ICE–WHR efforts and form the basis of this sub-section; however, Appendix B.3 surveys the broader global regulatory landscape. In the US, this influence begins unassumingly with the passage of the California Air Pollution Control Act of 1947, which "authorized the creation of Air Pollution Control Districts (APCD) in every county of the State" [304]. At the national level in the US, pollution had been studied for decades, leading the Eisenhower administration to ultimately enact the Air Pollution Control Act of 1955, providing federal "support and aid" for technical research while recognizing "rights of the States and local governments in controlling air pollution" [305]. The Clean Air Act of 1963 [306], (p. 19), not to be confused with the London Clean Air Act of 1956 [307] focused on dark chimney smoke, expands the funding of research and instantiates a federal role for pollution cleanup and response [308]. Furthering this motivation are the Clean Air Act Amendments of 1970 (often simply called the Clean Air Act), announcing that the government (eventually through the EPA, established that same year) would publish "national primary and secondary ambient air quality standards" (NAAQS). Rounding out this period-specific summary of overarching US regulations, the Clean Air Act Amendments of 1977 categorized different sectors (e.g., power, transportation) for independent regulation [309].

More specifically on vehicles, the 1970 Clean Air Act Amendments set emission limits for 1976 model year vehicles [298], driving research on Rankine cycle engines, in which the expansion process can be separated from the combustion process. Around this time, oil prices underwent significant turmoil, leading Congress to adopt fuel economy standards (Corporate Average Fuel Economy—CAFE) in 1975 [310,311]. Assisting with the governmental support of low emission engine development is the Division of Advanced Automotive Power Systems Development, a part of the EPA [299], the National Pollution Control Administration (NAPCA) [296], and the Alternative Automotive Power Systems (AAPS) program [303], with additional efforts by individual states (predominantly California) and companies.

Appendix B.2. Volatile Oil Market Dynamics

Hall's 1981 article also identifies a motivation for ICE–WHR as the volatile oil market dynamics of the 1970s in the United States and worldwide [312], especially the oil embargo of 1973–1974 and the OPEC price hike of 1979 [310,313]. During the embargo, oil prices shot "from $3 per barrel to $12 per barrel," causing price alarm on top of the availability concerns accompanying decreased production [310]. The "100-percent OPEC-inspired price hike" of 1979 [312] occurred downstream of the first ORC for ICE–WHR efforts; however, secondary research efforts were undoubtedly motivated by the totality of the oil price volatility of the 1970s.

Appendix B.3. Global Trends in Research Drivers

As mentioned in Appendix B.1, initial research efforts were geolocated in the United States, largely driven by regulatory (Appendix B.1) and oil (Appendix B.2) factors; however, research on ORCs for ICE–WHR quickly spread across the developed world. The oil market and regulatory environment are unique to each region, so rather than attempting to elucidate that complex history, a limited set of salient points are given here for basic context.

Through the 1970s, and in earlier decades, many countries observed increasingly poor air quality and in response adopted air quality standards and/or emissions limits, while Europe notably imposed high fuel taxes [314]. Regarding oil, China, Europe, and Japan endured the oil embargo in a similar way to the US, making oil market dynamics a research driver in those countries as well. In the late 1970s and through the 1980s, many countries passed or greatly expanded environmental regulations. China, for example, began establishing a broad "Environmental Protection Law" in 1979 [315].

Oil crises were unequally felt in other regions, though, as oil-exporting nations (e.g., countries in the Middle East and the Soviet Union) were relatively unaffected. In the Soviet Union, much of the 20th century involved a decades-long series of modest regulations, leading to the formation of the more centralized Union Committee for Environmental Protection in 1988 and subsequently the Russian Federation passing the Law on Environmental Protection in 1991 [316]. This law centered on "pollution charges" and "environmental funds".

Less-affluent countries face a range of challenges, counteracting the intrinsic air quality benefits of having less industrialization and vehicle-based transport. In regions with little personal wealth, industry wields significant influence, and even healthy industries may not be capable of financing the systems necessary to satisfy ambitious governmental regulations [317]. Certain developing countries have demonstrated fruitful success with voluntary approaches; however, the results do not show voluntary approaches to be definitively the best approaches [318].

Appendix C. Compilation of ORC for ICE–WHR (and Related) Reviews

Given the present systematic review's consideration of research articles and review articles, this article performs a type of meta-analysis. In the latter case, some reviews have a specific focus based on waste heat sources (e.g., heavy-duty diesel engines), a certain time period, certain ORC topics (e.g., working fluid decomposition), and so on. Accordingly, this appendix tabulates the individual review efforts that contribute to a meta-analysis of the past decade of research on ORCs for ICE–WHR.

Table A2. Compilation of Review Articles with Relevance to ORC–WHR from ICEs.

Year	Author(s)	Description	Reference
2012	Lopes et al.	Component-focused ORC review covering different component designs (e.g., different heat exchangers, expanders, etc.).	[319]
2012	Saidur et al.	Review of WHR technologies for ICE exhaust, various methods.	[80]
2012	Sprouse III and Depcik	Review of ORCs for ICE–WHR, precursor to the current article.	[1]
2013	Abedin et al.	ICE energy balances for different alternative fuels.	[32]
2013	Jadhao and Thombare	Review of exhaust gas WHR from ICEs, various methods.	[48]
2013	Quoilin et al.	Survey of ORC applications across industries and challenges.	[87]
2014	Song et al.	Review of using scroll expanders in ORC systems.	[272]
2015	Aghaali and Ångström	Turbocompounding for ICE–WHR.	[45]
2015	Colonna et al.	Vision article covering broad ORC uses, especially electricity.	[280]
2015	Delgado and Lutsey	Vision article on long-haul truck efficiency for 2020–2030.	[36]
2015	Karvonen et al.	Review of all ICE–WHR technologies with patent analysis.	[320]
2015	Lecompte et al.	Review of ORC configurations for WHR, electricity-focused.	[6]
2016	Rahbar et al.	Review of small-scale ORCs, applications, and expanders.	[321]
2016	Zhai et al.	Perspective on categorizing WHR sources for ORCs.	[322]

Table A2. *Cont.*

Year	Author(s)	Description	Reference
2016	Zhou et al.	History and future of vehicle WHR using Rankine cycles.	[323]
2017	Arefin et al.	Review of all ICE–WHR technologies.	[179]
2017	Bronicki	History of all ORC systems.	[259]
2017	Lion et al.	Review of heavy-duty diesel WHR using ORCs on- and off-highway.	[178]
2017	Tartière and Astolfi	Review of market for ORCs worldwide.	[213]
2017	Tocci et al.	Techno-economic review of small-scale ORCs.	[159]
2018	Alshammari et al.	ORC expander technologies for ICE–WHR.	[195]
2018	Jiménez-Arreola et al.	Review of challenges and strategies for managing ORC heat source fluctuations.	[181]
2018	Mahmoudi et al.	Review of "recent" ORC–WHR efforts, including from ICEs.	[324]
2018	Shi et al.	Review of "modified" ORCs for ICE–WHR.	[325]
2019	Dai et al.	ORC working fluid thermal stability review.	[204]
2019	Xu et al.	Review of ORCs for HDDE–WHR with introductory content.	[214]
2020	Loni et al.	Industrial WHR with ORCs, recent results, and outlook.	[326]
2021	Oyedepo and Fakeye	Review of WHR as technology for sustainable energy.	[327]
2021	Savitha et al.	Literature review of low GWP refrigerants.	[328]
2021	Tian et al.	Vision article for Rankine cycle WHR from ICEs.	[242]
2022	Dahham et al.	Recent thermal efficiency improvements of ICEs.	[31]
2022	Joshi	Review of engine efficiency and emissions for ICEs.	[28]
2022	Kuah et al.	Bibliometric study of WHR covering 1991–2020.	[37]
2023	Balazadeh Meresht et al.	Recent efficiency advancements in long-haul trucks.	[329]
2023	Wieland et al.	Vision article for ORC power systems.	[330]
2023	Wieland et al.	Survey of the ORC power systems market.	[50]

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