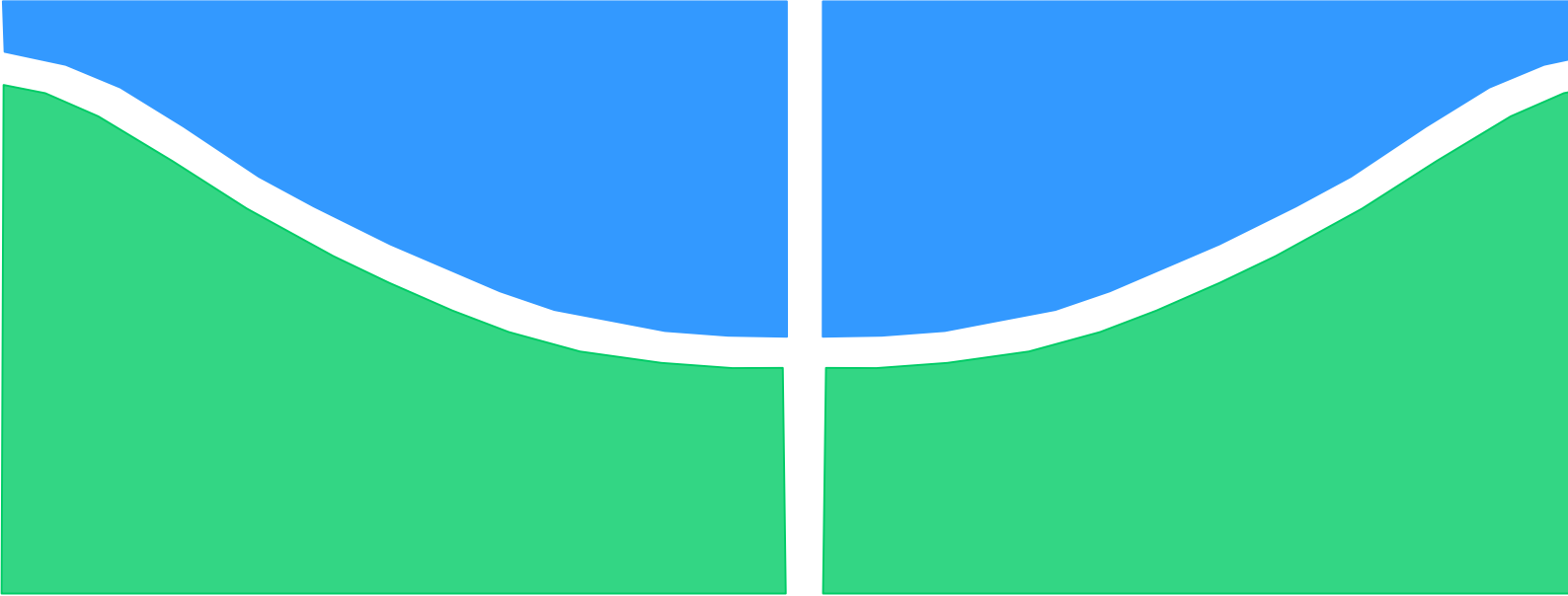




University of Brasília (UnB)
Faculty of Technology
Graduate Program in Mechanical Sciences

Thermodynamic and Exergy Analysis of a Carnot Battery Combined Cycle for Ethanol-Based Range Extenders

Author: Lucas Amorim Pereira Gomes
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Brasília, DF
2026

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ABSTRACT

The decarbonization of heavy-duty transport requires innovative solutions capable of combining energy efficiency, operational flexibility, and the use of renewable fuels. In this context, range-extended electric vehicles (REEVs) powered by biofuels represent a promising transitional pathway. This study proposes and evaluates a novel Carnot Battery Combined Cycle (CBCC) architecture for an ethanol-based range extender, integrating an Otto-cycle internal combustion engine (ICE), a phase change material (PCM)-based thermal energy storage system, and a wet-steam Rankine cycle for waste heat recovery. A thermodynamic and exergy-based model was developed in Engineering Equation Solver (EES), incorporating combustion analysis, exhaust-energy characterization, thermal storage, and bottoming-cycle performance. A genetic algorithm was applied to optimize the expander inlet pressure, expander outlet pressure, and degree of steam superheating, while the PCM charging and discharge times were fixed at 1800 s and 300 s, respectively. The engine model predicted 69.38 kW of brake power, 47.92 kW of exhaust thermal power, and 27.56 kW of exhaust physical exergy, corresponding to an exergy-to-energy ratio of approximately 57.5%. In the directly coupled configuration, the Rankine cycle was constrained by the instantaneous recoverable exhaust-heat rate and produced approximately 2.615 kW of thermodynamic net power, 2.484 kW of mechanical shaft power, and 2.36 kW of electrical output while the engine was operating. In contrast, PCM-based thermal storage enabled temporal decoupling between heat recovery and power conversion by accumulating thermal energy during 1800 s of engine operation and releasing it during a 300 s discharge period. Under these conditions, the optimized PCM-assisted Rankine cycle produced approximately 15.69 kW of thermodynamic net power, 14.91 kW of mechanical shaft power, and 14.16 kW of electrical output after generator losses. When evaluated over the complete charge–discharge operating cycle, the overall efficiency increased from 39.13% for the engine alone to approximately 40.53% for the integrated CBCC architecture. The results demonstrate that PCM-based thermal storage does not increase the total energy or theoretical exergy available in the exhaust stream. Instead, it modifies the temporal distribution of the recovered energy, converting a relatively low continuous heat rate into a higher intermittent power output and thereby improving operational flexibility and dispatchability. The proposed architecture contributes to the development of sustainable heavy-duty transport technologies by combining renewable fuels, hybrid electric propulsion, thermal energy storage, and thermodynamic optimization.

Keywords: Range Extender, Waste Heat Recovery, Carnot Battery Combined Cycle, PCM.

Análise Termodinâmica e Exergética de um Ciclo Combinado com Bateria Carnot para Extensores de Autonomia a Etanol

RESUMO:

A descarbonização do transporte pesado exige soluções inovadoras capazes de combinar eficiência energética, flexibilidade operacional e o uso de combustíveis renováveis. Neste contexto, os veículos elétricos de autonomia estendida (REEVs) alimentados por biocombustíveis representam uma rota de transição promissora. Este estudo propõe e avalia uma nova arquitetura de Ciclo Combinado com Bateria Carnot (CBCC) para um extensor de autonomia a etanol, integrando um motor de combustão interna (MCI) ciclo Otto, um sistema de armazenamento de energia térmica baseado em material de mudança de fase (PCM) e um ciclo Rankine com vapor úmido para recuperação de calor residual. Um modelo termodinâmico e exergético foi desenvolvido no *Engineering Equation Solver* (EES), incorporando análise de combustão, caracterização da energia dos gases de exaustão, armazenamento térmico e desempenho do ciclo inferior (*bottoming cycle*). Um algoritmo genético foi aplicado para otimizar a pressão de entrada no expensor, a pressão de saída no expensor e o grau de superaquecimento do vapor, enquanto os tempos de carga e descarga do PCM foram fixados em 1800 s e 300 s, respectivamente. O modelo do motor previu 69,38 kW de potência efetiva (*brake power*), 47,92 kW de potência térmica de exaustão e 27,56 kW de exergia física de exaustão, correspondendo a uma razão exergia-energia de aproximadamente 57,5%. Na configuração diretamente acoplada, o ciclo Rankine foi limitado pela taxa instantânea de calor recuperável da exaustão e produziu aproximadamente 2,615 kW de potência líquida termodinâmica, 2,484 kW de potência mecânica no eixo e 2,36 kW de saída elétrica enquanto o motor estava em operação. Em contrapartida, o armazenamento térmico baseado em PCM permitiu o desacoplamento temporal entre a recuperação de calor e a conversão de potência, acumulando energia térmica durante 1800 s de operação do motor e liberando-a durante um período de descarga de 300 s. Sob essas condições, o ciclo Rankine otimizado e assistido por PCM produziu aproximadamente 15,69 kW de potência líquida termodinâmica, 14,91 kW de potência mecânica no eixo e 14,16 kW de saída elétrica após as perdas do gerador. Quando avaliada ao longo do ciclo operacional completo de carga e descarga, a eficiência global aumentou de 39,13% para o motor isolado para aproximadamente 40,53% para a arquitetura CBCC integrada. Os resultados demonstram que o armazenamento térmico baseado em PCM não aumenta a energia total nem a exergia teórica disponível no fluxo de exaustão. Em vez disso, ele modifica a distribuição temporal da energia recuperada, convertendo uma taxa de calor contínua relativamente baixa em uma potência de saída intermitente mais elevada, melhorando, assim, a flexibilidade operacional e a despachabilidade. A arquitetura proposta contribui para o desenvolvimento de tecnologias sustentáveis para o transporte pesado ao combinar combustíveis renováveis, propulsão elétrica híbrida, armazenamento de energia térmica e otimização termodinâmica.

Palavras-chave: Extensor de Alcance. Recuperação de Calor Residual. Ciclo Combinado com Bateria Carnot. PCM.

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LIST OF SYMBOLS

Symbol	Description	Unit
A	Heat transfer area	m^2
B	Cylinder bore	mm
c_p	Specific heat at constant pressure	$kJ.kg^{-1}.K^{-1}$
$\dot{E}_x$	Exergy rate	kW
$\dot{E}_{x,fuel}$	Fuel exergy rate	kW
$\dot{E}_{x,exh}$	Exhaust gas exergy rate	kW
h	Specific enthalpy	$kJ.kg^{-1}$
h_{fg}	Latent heat of fusion	$kJ.kg^{-1}$
$\dot{m}$	Mass flow rate	$kg.s^{-1}$
$\dot{m}_{air}$	Air mass flow rate	$kg.s^{-1}$
$\dot{m}_{fuel}$	Fuel mass flow rate	$kg.s^{-1}$
$\dot{m}_{st}$	Steam mass flow rate	$kg.s^{-1}$
N	Engine rotational speed	rpm
n	Polytropic exponent	-
P_{RC1}	Expander inlet pressure	kPa
P_{RC2}	Expander outlet pressure	kPa
P_{RC3}	Pump inlet pressure	kPa
P_{RC4}	Pump outlet pressure	kPa
P_{intake}	Intake pressure	kPa
$\dot{Q}_{exh}$	Exhaust thermal power	kW
$\dot{Q}_{exh,recoverable}$	Recoverable exhaust thermal power	kW
$\dot{Q}_{charge}$	Thermal power transferred to PCM	kW
$\dot{Q}_{in}$	Heat input to Rankine Cycle evaporator	kW
R	Gas constant	$J.mol^{-1}.K^{-1}$
s	Specific entropy	$kJ.kg^{-1}.K^{-1}$
S	Piston Stroke	mm
T	Temperature	K
T_0	Reference environmental temperature	K
T_{exh}	Exhaust gas temperature	K
T_{melt}	PCM melting temperature	°C
t_{charge}	PCM charging time	s
$t_{discharge}$	PCM discharging time	s
u	Specific internal energy	$kJ.kg^{-1}$
V	Volume	m^3
V_d	Displacement volume	m^3
V_c	Clearance volume	m^3
W	Work	kJ
$\dot{W}_{net}$	Net Engine Power output	kW
$\dot{W}_{exp}$	Expander power	kW
$\dot{W}_{pump}$	Pump power	kW
$\dot{W}_{net,RC}$	Net RC power	kW
$\dot{W}_{shaft,RC}$	RC shaft power	kW

$\dot{W}_{shaft,RC,direct}$	Direct RC shaft power	kW
$\dot{W}_{electric,RC}$	RC effective electrical power	kW
$\dot{W}_{electric,RC,direct}$	Direct RC effective electrical power	kW
x_2	Steam quality – expander outlet	-
η_{bth}	Brake thermal efficiency	-
η_{ex}	Exergy efficiency	-
η_m	Mechanical efficiency	-
η_{RC}	Rankine cycle efficiency	-
η_{pump}	Pump efficiency	-
η_{exp}	Expander isentropic efficiency	-
$\varepsilon_{Hx,charge}$	Heat transfer effectiveness	-
ΔT_{sup}	Steam superheating	°C
ϕ	Equivalence ratio	-
ψ	Exergy-to-energy ratio	-

ABBREVIATIONS AND ACRONYMS

<i>Nomenclatures</i>	<i>Acronyms</i>
Battery Electric Vehicle	BEV
Carnot Battery Combined Cycle	CBCC
CR	Compression Ratio
Electric Vehicle	EV
Engineering Equation Solver	EES
Greenhouse Gas	GHG
Internal Combustion Engine	ICE
Internal Combustion Engine Vehicle	ICEV
Lower Heating Value	LHV
Organic Rankine Cycle	ORC
Phase Change Material	PCM
Plug-in Hybrid Electric Vehicle	PHEV
Rankine Cycle	RC
Range-Extended Electric Vehicle	REEV
State of Charge	SOC
Thermal Energy Storage	TES
Total cost of ownership	TCO
Waste Heat Recovery	WHR

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1. INTRODUCTION

In 2024, road transport was responsible for emissions of 6.2 Gt CO₂-eq, accounting for 16.4% of global energy-related CO₂ emissions. Road vehicles accounted for 73% of direct transport emissions (IEA, 2025a, 2025d).

Amid growing concerns over emissions and oil dependency, electric vehicles (EVs) have emerged as a primary solution to decarbonize the road transport sector. In 2024, electric heavy-duty trucks comprised just under 3% of worldwide heavy-duty vehicle sales, a figure projected to increase to almost 15% by 2035, mostly driven by China and the European Union. In 2024, electric heavy-duty vehicle sales in China were over 16% of the nation's total heavy-duty truck sales, establishing China as a leader in the electrification of freight transport (IEA, 2025e).

In the context of electricity consumption, electricity demand related to transport grew by 7% between 2023 and 2024, indicating a strong upward trend as the decade progresses. However, in 2024, only 0.3% of global electricity was allocated to the transport sector (U.S. ENERGY INFORMATION ADMINISTRATION, 2025). Hybrid vehicles can help stabilize energy consumption amid the rapid increase in electrified vehicle sales, contributing to a sustainable transition in the transportation sector. Additionally, hybrid vehicles maintain an energy balance through the use of biofuels alongside electric propulsion, offering reassurance to consumers who still face challenges with the availability of a comprehensive vehicle charging infrastructure, especially in regions where charging stations are scarce or unreliable. Moreover, a significant obstacle impeding the adoption of electric vehicles is range anxiety—the apprehension that an EV's battery may lack adequate charge to arrive at the intended destination or fulfill a scheduled trip (Adnan et al., 2017; Tate et al., 2008).

Although battery-electric vehicles (BEVs) markedly diminish greenhouse gas emissions, apprehensions regarding battery production effects, charging infrastructure, and range constraints have sustained interest in hybrid alternatives (VEZA *et al.*, 2023).

It is well established that BEVs emit no tailpipe pollutants (TOGUN *et al.*, 2025). However, their GHG impact depends on the electricity mix used for charging, making energy sources a decisive factor in transport electrification. So, as road transport becomes increasingly electrified, both directly (e.g., EVs) and indirectly (e-fuels), the potential to integrate renewable electricity into the sector is expected to grow (RAVI; AZIZ, 2022).

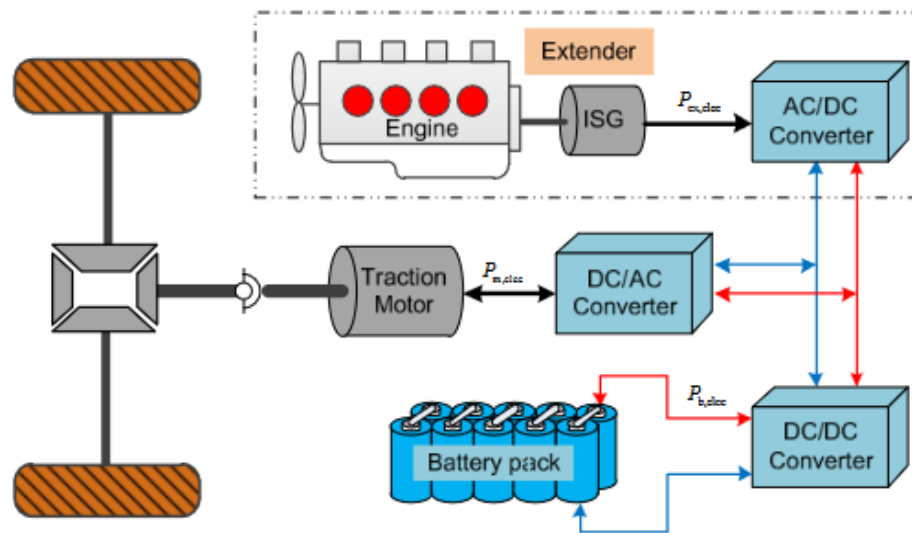
Despite concerns about indirect emissions from BEV charging, BEV sales have surged worldwide. In 2024, nearly half of China's car sales were electric, representing two-thirds of

the global total, while the US reached over 10% (1.6 million units). Europe stagnated due to weaker regulation, but emerging economies in Asia, Latin America, and Africa saw sales grow over 60%, driven by subsidies and low-cost Chinese EVs (IEA, 2025c). Despite global efforts to encourage the adoption of BEVs, sales of Internal Combustion Engine Vehicles (ICEVs) are projected to remain strong for several decades, particularly in developing economies, due to slow vehicle turnover rates; thus, ICEVs are expected to persist in global fleets for many more years (MORFELDT; DAVIDSSON KURLAND; JOHANSSON, 2021)

Thus, plug-in hybrid electric vehicles (PHEVs) offer a balance between electrification and extended range, leveraging an electric propulsion system and an Internal Combustion Engine (ICE) to improve efficiency and reduce emissions (XU *et al.*, 2024).

An extended-range electric vehicle (REEV) operates predominantly as a fully electric vehicle, with propulsion provided entirely by an electric motor. It also incorporates a small internal combustion engine (ICE) that functions solely as a generator to supply additional electrical energy, as illustrated in Fig. 1. From another viewpoint, an REEV can be described as a series hybrid configuration featuring a substantially larger battery pack, typically in the range of 100 to 500 kWh for medium and heavy-duty truck applications.

Figure 1 - REEV architecture applied to medium/heavy-duty trucks



Source: Kalia & Fabien, 2020

As an example, DHL Group and Scania have collaborated on an initiative to develop and evaluate an extended-range electric truck that integrates battery-electric propulsion with a

fuel-powered generator. This concept addresses current limitations in charging infrastructure while supporting the transition to low-emission freight transport. The project is designed to enable operation with approximately 80-90% renewable electricity usage, while still ensuring sufficient range and flexibility for long-haul logistics. The vehicle is being tested in real-world operations between Berlin and Hamburg, highlighting its potential as a practical interim solution for decarbonizing heavy-duty transport as full electrification and charging networks continue to expand (DHL GROUP; SCANIA, 2025).

In this sense, the use of ICE addresses an opportunity to explore biofuels in REEVs. Globally, biofuel use has excellent prospects, considering that demand is expected to increase 23%, to 200 billion liters, by 2028, with renewable diesel and ethanol accounting for two-thirds of this growth, and biodiesel and aviation biofuel constituting the remainder (IEA, 2024b). Thus, low-carbon fuels are critical for decarbonizing the existing fleet (IEA, 2024b), since sustainably produced renewable fuels offer GHG reductions comparable to BEVs powered by renewable electricity (KOROMA *et al.*, 2022) while decreasing in-cylinder pollutants and reducing fossil fuel dependency (SANDAKA; KUMAR, 2023).

One of the motivations for the proposed architecture is the search for improvements in low-emission mobility that support several Sustainable Development Goals (SDGs), particularly SDGs 7 (Affordable and Clean Energy), 11 (Sustainable Cities and Communities), and 13 (Climate Action) (UNITED NATIONS DEPARTMENT OF ECONOMIC AND SOCIAL AFFAIRS, 2024).

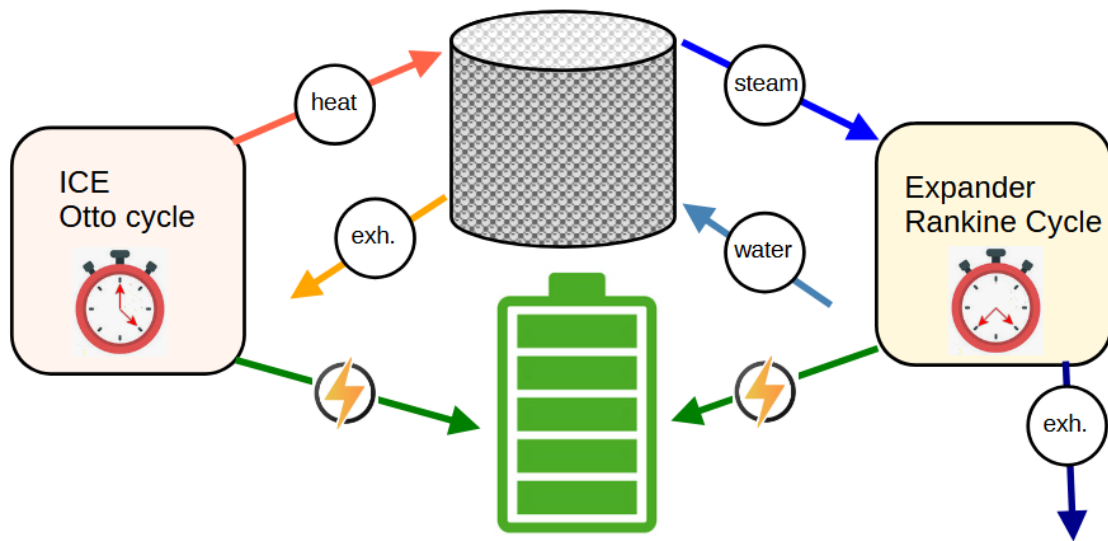
This research aims to evaluate a novel system for extending the range of heavy-duty transport. The system employs a combined power generation cycle. This cycle is activated when the battery State of Charge (SOC) falls below a predefined threshold. For longer distances, REEVs utilize the ICE to keep the battery charged, consuming less fuel than conventional ICEVs for the following two reasons (DELL; MOSELEY; RAND, 2014):

1. The engine of an REEV is significantly smaller than that of a conventional ICEV as it only needs to meet average power demands because peak power is delivered by the battery pack, e.g. accelerations.
2. The engine of a REEV operates at a constant, highly efficient, rotation speed.

Ethanol-fueled ICEs present a particularly advantageous solution for range extension in Brazil, leveraging the nation's established ethanol infrastructure, representing a significant opportunity for investment to decarbonize national transportation. The performance of conventional ICE-based range extenders is constrained by their limited thermal efficiency and significant energy losses through exhaust gases. To mitigate these inefficiencies, this work

proposes a novel range-extender architecture, based on a modified Combined Cycle as shown in Fig. 2. The proposed architecture, herein referred to as the Carnot Battery Combined Cycle (CBCC), integrates an ethanol-fueled ICE, operating on an Otto cycle, with a thermally driven power generation subsystem based on a steam Rankine Cycle (RC). The key innovation lies in replacing the conventional heat recovery unit (HRU) with a Carnot battery, enabling temporal and operational decoupling between the two thermodynamic cycles.

Figure 2 - The Carnot Battery Combined Cycle (CBCC) range extender, featuring a shared electric generator for both the topping and bottoming cycles.



Source: Author's own elaboration.

In this configuration, the thermal energy from the internal combustion engine (ICE) exhaust gases is not transferred directly to the steam cycle. Instead, it is stored in a Thermal Energy Storage (TES) system based on either phase change materials (PCM) or sensible heat storage media, which form the core of the Carnot battery. The storage medium may consist of low-cost materials such as sand saturated with a thermally conductive fluid. During the charging phase, exhaust heat is stored as latent and/or sensible thermal energy. During discharge, the stored thermal energy is released and serves as the heat source for the steam cycle, driving an expander. Steam condensation is carried out in a low-cost direct-contact feedwater condenser.

This architecture introduces a decoupled power generation strategy, in which the ICE and the steam cycle do not operate simultaneously but rather sequentially, depending on the SOC of the thermal storage system. As a result, both subsystems can be designed to deliver comparable electrical power outputs and, therefore, may share a common electric generator.

The proposed system offers some advantages and disadvantages, detailed below:

- Temporal decoupling of heat generation and power conversion, improving operational flexibility;
- Enhanced utilization of waste heat, independent of real-time engine operation;
- Load leveling capability, enabled by the Carnot battery storage;
- Improved overall system efficiency;
- Scalability and modularity, governed by the sizing of the integrated system;
- Heat transfer irreversibilities between the Carnot battery streams;
- Operation of a modified Rankine cycle, which inherently has low thermal efficiency.

This configuration represents a significant departure from conventional combined cycles, where thermodynamic coupling imposes simultaneous operation constraints. By introducing a Carnot battery-mediated interface, the system effectively transforms a traditional combined cycle into a hybrid, time-shifted energy conversion platform, suitable for advanced range-extender applications.

This research is justified by its potential to contribute to a more sustainable and efficient automotive energy landscape, particularly in the context of low-carbon mobility solutions. From a technical and scientific perspective, it investigates the synergistic integration of three key concepts: series hybrid electric vehicles, bioethanol as a renewable fuel, and TES. This combination remains underexplored in the literature, especially in the context of advanced range-extender architectures for heavy-duty trucks.

By introducing a thermal buffer, the proposed system enables a more controllable and dispatchable thermal input and operational flexibility in waste recovery systems. Accordingly, this study is guided by the following research questions:

- RQ1: How does the CBCC affect the overall thermodynamic and exergy efficiency of an ethanol-based range extender compared to a baseline configuration?
- RQ2: How does temporal decoupling enabled by the Carnot battery influence system performance, operational flexibility, and thermal management in the proposed CBCC architecture?
- RQ3: How do the design and operating parameters of the PCM-based TES system affect energy utilization and the operational behavior of the CBCC architecture?

To address these questions, the research tests the following hypotheses:

- H1: The integration of the Carnot battery improves the practical utilization of exhaust-gas energy and increases the overall useful energy delivered by the range-extender

system, while the theoretical exergy content of the exhaust stream remains limited by the exhaust thermodynamic state.

- H2: The performance of the CBCC system is sensitive to the design of the PCM-based TES system, particularly to storage capacity and heat-transfer characteristics, which directly affect energy utilization and operational flexibility.

Building upon the research questions and hypotheses defined above, the proposed architecture adopts a thermodynamic modeling approach to evaluate the proposed CBCC architecture under representative operating conditions of an ethanol-fueled range extender. The analysis is based on mass and energy balances, complemented by an assessment of the exergy content of exhaust gases, in order to quantify the recoverable energy potential and its effective conversion into useful work.

The proposed system is assessed by comparing a baseline configuration, in which exhaust gases are not subjected to decoupled thermal recovery, with the CBCC architecture incorporating a TES system. Particular emphasis is placed on the role of temporal decoupling in enabling controlled heat delivery to the bottoming RC, as well as on the influence of storage capacity and thermal characteristics of the energy storage system.

Through this framework, the study seeks to establish a consistent link between thermodynamic potential, system design, and operational behavior, providing a structured basis to address RQ1, RQ2, and RQ3 within the scope of the developed model.

This dissertation contributes by proposing and thermodynamically evaluating a novel Carnot Battery Combined Cycle architecture that integrates an ethanol-fueled Otto-cycle range extender, TES system, and a steam RC for WHR, to extend the range of electric medium and heavy duty trucks with near zero carbon emissions.

2. LITERATURE REVIEW

This section will present the essential and technical concepts pertinent to this study. It provides a preliminary and conceptual framework for foundational comprehension.

2.1 Range Anxiety and Range Extender Technologies

2.1.1 Range Anxiety

Range anxiety is defined as the driver's concern regarding the limited driving range of BEVs and the risk of not reaching a charging station before battery depletion. This psychological and operational constraint remains one of the primary barriers to the widespread adoption of EVs, particularly in heavy-duty and long-haul applications (IEA, 2025c; LI, W. *et*

al., 2017). Concurrently, range anxiety can profoundly influence the driving and charging habits of EV operators, with significant ramifications for the overall efficiency and stability of the electrical grid (NEUBAUER; WOOD, 2014). Nonetheless, there is insufficient research regarding the relationship between a perceived daily charging routine, demographic and psychological factors, and charging-related variables in relation to range anxiety (ZATSARNAJA *et al.*, 2025).

Although advancements in battery technology have increased energy density and vehicle autonomy, several structural challenges persist, including long charging times, limited charging infrastructure, and operational inflexibility under high-load conditions. These issues are especially critical in freight transport, where route predictability, payload capacity, and uptime are essential (IEA, 2025c).

In heavy-duty vehicles, range anxiety is not only a psychological factor but also a logistical constraint, directly affecting fleet operation, route planning, and total cost of ownership (TCO). As a result, alternative architectures that mitigate range limitations while preserving electrification benefits have gained increasing attention in recent years.

2.1.2 Range Extender Architectures

Range extender electric vehicles (REEVs) are hybrid systems in which an ICE or another onboard energy conversion device is used exclusively to generate electricity, rather than directly propel the vehicle. In these systems, the propulsion remains fully electric, while the range extender acts as an auxiliary power unit (APU) (CHAN, 2007; OFFER *et al.*, 2010).

Several technologies have been proposed and implemented as range extenders:

a) Internal Combustion Engines (ICE-based): The most common solution employs reciprocating internal combustion engines operating under Otto or Diesel cycles. These systems are widely adopted due to their technological maturity, relatively low cost, and fuel flexibility. When decoupled from the drivetrain, ICEs can operate near optimal efficiency conditions, improving fuel conversion efficiency compared to conventional vehicles (GUZZELLA; SCJARRETTA, 2007).

b) Gas Turbines: Gas turbines have been investigated as range extenders due to their high power density, low vibration, and fuel flexibility. However, their lower efficiency at partial loads and high costs limit their practical application in automotive systems (DIXON; HALL, 2014).

c) Fuel Cells: Fuel cell systems, particularly hydrogen-based, represent a zero-emission alternative. However, challenges related to hydrogen infrastructure, storage, and cost hinder their large-scale deployment (STAFFELL *et al.*, 2019).

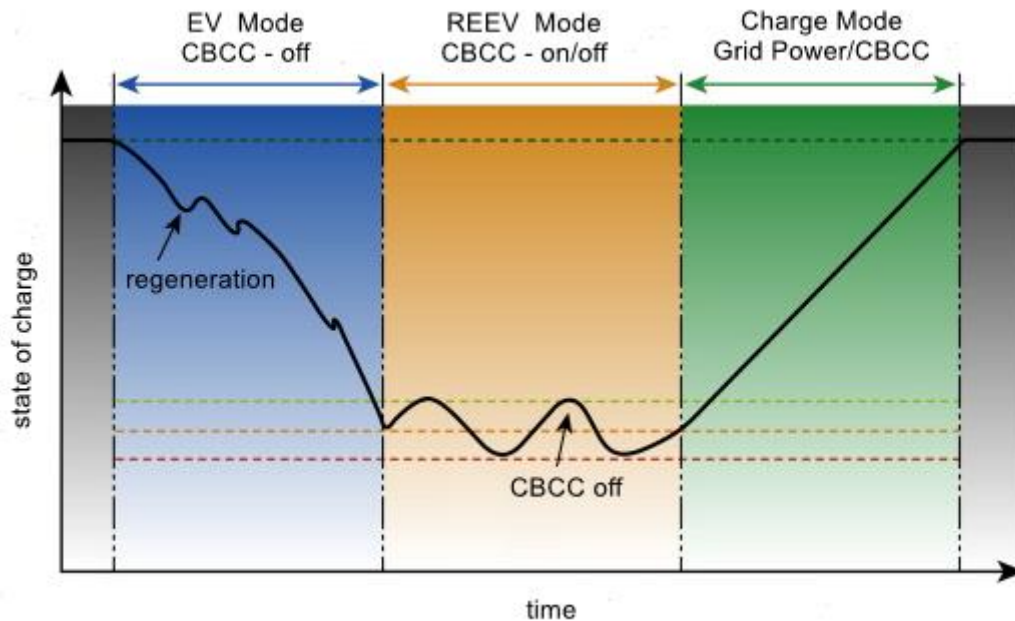
Overall, ICE-based range extenders remain the most viable near-term solution, particularly when combined with low-carbon fuels such as ethanol, as highlighted in the present study.

In conjunction with architectural studies, certain research examines the disparate effects of various control strategies on vehicle performance. Control strategies for range-extended power systems can be categorized into rule-based and optimization-based strategies (Wang *et al.*, 2017).

The rule-based control strategy predominantly establishes various control parameters and fixed thresholds derived from the performance metrics of the power system and the designer's engineering expertise to manage and transition between distinct operating states of the vehicle. The primary control variables encompass the SOC of the power battery, the power demand of the vehicle, and the instantaneous vehicle speed. Rule-based control strategies can be categorized into four types: constant power control strategy, power-following control strategy, multi-point control strategy, and fuzzy control strategy (LARSSON; JOHANNESSON; EGARDT, 2015; LI, J. *et al.*, 2017).

Figure 3 illustrates the power dynamics of a typical battery electric vehicle (BEV) equipped with a range extender. The state of charge (SOC) evolves according to the vehicle's energy management strategy. Initially, in pure EV mode (CBCC off), the battery discharges over time, with small fluctuations due to regenerative braking. When the SOC reaches a predefined lower threshold, the vehicle transitions to REEV mode, in which the CBCC unit is intermittently activated (on/off) to sustain the charge within a bounded range. Finally, in charging mode, the battery is recharged either from the electrical grid or via the CBCC in situations where a charging station is not available, such as in remote areas.

Figure 3 - Dynamics of the state of charge of a REEV architecture.



Source: Adapted from Dell, Moseley and Rand (2014)

2.1.3 Otto-Cycle Engines Applied to Range Extenders

Spark-ignition engines operating under the Otto cycle are particularly suitable for range extender applications due to their operational flexibility and compatibility with renewable fuels such as ethanol. Ethanol presents a high octane number and high latent heat of vaporization, enabling operation at elevated compression ratios (CRs) and improved knock resistance (HEYWOOD, 1988).

Furthermore, the employment of ethanol as a renewable alternative fuel, especially in nations such as Brazil with robust ethanol production infrastructure, presents potential for diminishing carbon emissions and improving engine efficiency (BENAJES et al., 2019). In range extender configurations, the ICE is decoupled from transient driving conditions, allowing operation under quasi-steady and optimized regimes. This enables improved brake thermal efficiency compared to conventional automotive applications (GUZZELLA; SCIARRETTA, 2007).

Thakur et al. (2017) reviewed the influence of key operating parameters, including equivalence ratio, intake pressure, and compression ratio (CR), on the performance of spark-ignition engines, highlighting that their optimization can significantly improve thermal efficiency. In addition, Leone et al. (2015) demonstrated that the high octane rating of ethanol enables the use of higher compression ratios, leading to further improvements in engine

efficiency. Furthermore, ethanol-fueled Otto engines present additional advantages in the Brazilian context, including the widespread availability of renewable fuel and reduced life-cycle greenhouse gas emissions, reinforcing their suitability for sustainable hybrid propulsion systems.

2.1.4 Thermal Storage Systems

Thermal energy storage (TES) systems enable the temporal decoupling between heat availability and energy demand, thereby improving the utilization of intermittent or discontinuous thermal resources. Depending on the storage mechanism, TES technologies are generally classified into sensible heat storage, latent heat storage, and thermochemical storage. Sensible heat systems store energy by increasing the temperature of a solid or liquid medium, whereas latent heat systems exploit the enthalpy associated with a phase transition. Thermochemical systems store energy through reversible chemical reactions and generally offer higher energy densities, although they involve greater material and system complexity (DINCER; ROSEN, 2013; SHARMA *et al.*, 2009).

Sensible heat storage is technologically mature and commonly employs water, thermal oils, molten salts, rocks, concrete, ceramics, or packed-bed materials. The amount of energy stored in these systems depends on the mass of the storage medium, its specific heat capacity, and the allowable temperature variation. Consequently, sensible storage generally requires a relatively large temperature range or storage volume to achieve high storage capacities. Nevertheless, its operational simplicity, comparatively low material cost, and absence of phase-transition-related degradation make it attractive for industrial and solar thermal applications.

Xu *et al.* (2017), for example, investigated a sensible TES material consisting of sand saturated with a thermally conductive fluid. The objective was to improve the heat-transfer performance of conventional low-cost solid media while avoiding some of the thermal-expansion limitations associated with concrete storage systems. Experimental tests and a validated one-dimensional transient enthalpy model indicated that fluid-saturated sand provided more effective thermal charging and discharging than dry sand. When evaluated for a 600 MW concentrated solar power plant operating between 400 and 500 °C, the proposed medium showed favorable energy storage and extraction characteristics. The use of low-cost materials and a relatively simple construction was also identified as a potential economic advantage. Although this concept does not employ latent heat storage, it illustrates the importance of

combining storage capacity, heat-transfer performance, operating temperature, and cost when selecting a TES technology.

Latent heat thermal energy storage employs phase change materials (PCMs), which absorb or release considerable amounts of thermal energy during a phase transition. Solid–liquid PCMs are the most widely investigated because they usually provide high latent heat while avoiding the large volume variations and containment difficulties associated with liquid–gas transitions. Since melting and solidification occur within a comparatively narrow temperature range, PCM-based systems can store and deliver heat at an approximately constant temperature. This characteristic allows higher energy storage density and improved temperature control compared with conventional sensible storage systems operating over the same temperature interval (AGYENIM et al., 2010; Zalba et al., 2003; SHARMA et al., 2009).

PCMs have been extensively investigated in solar thermal systems, concentrated solar power plants, industrial waste heat recovery, building thermal management, refrigeration, electronics cooling, battery thermal control, and vehicle thermal management. Their principal advantage is not only the amount of energy stored per unit mass or volume, but also their ability to reduce the temporal mismatch between heat generation and heat demand. In waste heat recovery applications, this capability is particularly relevant because the thermal source may fluctuate considerably with the operating condition of the primary process or engine (MIRÓ; GASIA; CABEZA, 2016).

PCM candidates are commonly classified as organic, inorganic, and eutectic materials. Organic PCMs, including paraffins and fatty acids, generally exhibit good chemical stability, limited supercooling, and congruent melting. However, they often have relatively low thermal conductivity, moderate latent heat per unit volume, and limited suitability for high-temperature applications. Inorganic PCMs, including salt hydrates, molten salts, metals, and metallic alloys, may provide higher volumetric energy density and broader high-temperature applicability. Their practical implementation, however, may be affected by corrosion, phase segregation, supercooling, chemical instability, or compatibility problems with containment materials. Eutectic mixtures combine two or more components in proportions selected to obtain a specific melting temperature and may offer relatively sharp phase transitions and tunable thermophysical properties (KENISARIN, 2010; MOHAMED et al., 2017).

The selection of a PCM must therefore consider more than its latent heat and melting temperature. The phase-change temperature should match the temperature level of the heat source and the thermal requirements of the downstream process. Other relevant parameters include latent heat of fusion, specific heat, density, thermal conductivity, volume change during

phase transition, vapor pressure, degree of supercooling, phase segregation, chemical stability, cycling durability, flammability, toxicity, corrosion behavior, availability, and cost. Compatibility between the PCM, heat exchanger, encapsulation, and containment materials is particularly important for high-temperature systems, since chemical reactions and corrosion can reduce service life and compromise operational safety (Zalba et al., 2003).

Despite their favorable energy density, many PCMs exhibit low thermal conductivity. This limitation can produce slow melting and solidification rates, large internal temperature gradients, incomplete utilization of the storage medium, and reduced charging and discharging power. Therefore, the performance of a PCM storage system is determined not only by the intrinsic properties of the material but also by the design of the heat exchanger and the thermal interaction between the heat-transfer fluid and the PCM. Kalapala and Devanuri (2018) emphasized that operating conditions and geometric parameters, including heat-transfer-fluid temperature, mass flow rate, tube arrangement, heat-transfer area, PCM layer thickness, and exchanger configuration, strongly influence melting time, solidification time, thermal effectiveness, and overall storage performance.

Several heat-transfer enhancement techniques have consequently been investigated. These include extended surfaces and fins, metal foams, expanded graphite matrices, highly conductive particles, nanoparticles, heat pipes, multiple tubes, encapsulation, and cascaded arrangements using PCMs with different melting temperatures. Although these techniques can improve thermal power and reduce charging and discharging times, they may also decrease the mass or volume fraction occupied by the PCM, increase pressure losses, add weight, raise manufacturing cost, or introduce additional thermal contact resistances. The thermal design therefore involves a compromise between storage density, heat-transfer rate, compactness, system mass, complexity, and cost (KALAPALA; DEVANURI, 2018).

Encapsulation is another important design aspect. Macroencapsulation facilitates handling and replacement of the storage material and can increase the heat-transfer surface, whereas microencapsulation can improve material distribution and reduce leakage. Nevertheless, the encapsulation shell introduces additional thermal resistance and may be subjected to mechanical stresses caused by repeated volume changes. Long-term thermal cycling tests are thus necessary to evaluate changes in melting temperature, latent heat, chemical composition, structural integrity, and heat-transfer performance.

For high-temperature waste heat recovery, inorganic salts, salt mixtures, metals, and metallic alloys are often more appropriate than conventional organic PCMs. High-temperature PCMs can be selected to match the temperature of combustion gases and to supply heat to a

Rankine cycle at a relatively stable temperature. However, their use requires careful evaluation of corrosion, containment, thermal expansion, thermal stability, and compatibility with heat exchanger materials. Moreover, a high latent heat value alone does not guarantee superior system performance; the effective recoverable energy also depends on the heat exchanger effectiveness, the temperature difference available during charging and discharging, heat losses, and the fraction of the PCM that effectively undergoes phase change.

In internal combustion engine applications, waste heat recovery has predominantly been investigated through directly coupled Rankine or Organic Rankine cycles. In such configurations, exhaust-gas heat is transferred directly to the working fluid while the engine is operating. Although direct coupling avoids the intermediate storage stage, the Rankine subsystem remains dependent on the instantaneous exhaust mass flow rate and temperature. Transient vehicle operation may therefore result in unstable evaporation conditions, variable working-fluid flow rates, reduced expander efficiency, and complex control requirements. The relatively low and highly variable power available from the bottoming cycle can also make the selection and operation of pumps, expanders, heat exchangers, and electric generators more challenging.

Applications of PCMs in vehicles have primarily focused on engine warm-up, cabin heating, battery temperature control, electronics cooling, and thermal buffering of individual components. By contrast, the integration of a high-temperature latent heat storage unit between an ethanol-fueled internal combustion engine and a Rankine cycle for electrical power generation remains comparatively underexplored. In this arrangement, the PCM does not create additional exhaust energy or increase the theoretical exergy available from the engine. Instead, it accumulates exhaust heat over an extended charging period and releases it during a shorter discharge period, thereby increasing the instantaneous thermal power supplied to the Rankine cycle.

This temporal decoupling represents the principal functional contribution of the thermal storage subsystem considered in the present study. It allows the engine and the bottoming cycle to operate under different temporal regimes and potentially enables the Rankine subsystem to operate closer to a selected design condition. As a result, the storage unit may reduce the direct dependence of the Rankine cycle on transient exhaust conditions, increase operational flexibility, and permit intermittent operation at a higher power level. Nevertheless, these benefits must be assessed together with the additional mass, volume, thermal losses, heat exchanger requirements, cost, and control complexity introduced by the PCM storage system.

2.1.4.1 Heat Transfer Between Gaseous Streams and PCM-Based Thermal Storage Systems

The integration of phase change materials into waste heat recovery systems requires an effective thermal interface between the heat source and the storage material. In systems driven by hot gaseous streams, such as internal combustion engine exhaust gases, heat is transferred sequentially by convection from the gas to the heat-exchanger wall, conduction through the wall and containment structure, and conduction and natural convection within the PCM during melting.

The thermal performance of a gas-to-PCM storage system is therefore governed by several coupled resistances. On the gas side, the heat-transfer rate depends on exhaust-gas temperature, mass flow rate, specific heat capacity, velocity, flow regime, pressure drop, and heat-exchanger geometry. On the storage side, the main controlling parameters include PCM thermal conductivity, phase-transition temperature, latent heat, thickness of the PCM layer, encapsulation geometry, heat-transfer area, viscosity and the evolution of the solid–liquid interface.

Agyenim et al. (2010) emphasized that the main limitation of latent heat thermal energy storage systems is frequently not the storage capacity of the PCM, but the rate at which thermal energy can be transferred during charging and discharging. The relatively low thermal conductivity of many PCMs can lead to long melting and solidification periods and incomplete utilization of the storage volume. Kalapala and Devanuri (2018) similarly concluded that heat-exchanger geometry, heat-transfer-fluid temperature, mass flow rate, tube arrangement, PCM thickness, and the use of heat-transfer enhancement techniques strongly affect charging time, storage power, and overall system performance.

During the initial stage of PCM melting, heat transfer is predominantly controlled by conduction through the solid material. As melting progresses, buoyancy-driven natural convection develops in the liquid PCM and may significantly increase the heat-transfer rate. However, this behavior is strongly dependent on exchanger orientation and geometry. During solidification, natural convection becomes progressively weaker, and conduction generally becomes the dominant heat-transfer mechanism.

Because gaseous heat-transfer fluids generally have lower density and volumetric heat capacity than liquids, gas-to-PCM systems often require relatively large heat-transfer areas, extended surfaces, or compact channels to obtain adequate thermal power. Fins, metal foams, graphite matrices, heat pipes, multiple tubes, and cascaded PCM arrangements have been

investigated to reduce thermal resistance and improve PCM utilization. Nevertheless, these techniques may increase pressure drop, component mass, manufacturing complexity, and cost.

Experimental investigations confirm that no single heat-transfer efficiency can be universally applied to PCM storage systems. Medrano et al. (2009) evaluated different commercial heat exchangers operating as PCM thermal storage units and showed that the average charging and discharging powers varied substantially with exchanger type and operating conditions. Promopattum et al. (2017) demonstrated experimentally and numerically that the dynamic response of an air–PCM cross-flow heat exchanger depends on PCM transition temperature, airflow conditions, and exchanger geometry.

Studies specifically involving engine exhaust gases also demonstrate a strong dependence on the adopted configuration. Johar et al. (2016) developed a shell-and-tube latent heat storage unit using erythritol and exhaust gases from a stationary compression-ignition engine. The reported maximum charging efficiency was approximately 69.5%, while the amount of stored energy varied with engine load and exhaust conditions. Khader et al. (2024) experimentally integrated PCM compartments into an exhaust-gas heat exchanger for a micro-combined heat and power system and observed a substantial increase in the duration of useful heat delivery. These studies confirm the technical feasibility of exhaust-gas-driven PCM storage but also show that system performance remains highly configuration-specific.

Several definitions of effectiveness and efficiency are used in the literature. In the conventional effectiveness–NTU approach, heat-exchanger effectiveness represents the ratio between the actual heat-transfer rate and the maximum thermodynamically possible rate. For a PCM undergoing phase change at an approximately constant temperature, the maximum heat-transfer rate is commonly referenced to the difference between the inlet heat-transfer-fluid temperature and the PCM melting temperature. The resulting effectiveness depends on the overall heat-transfer coefficient, heat-transfer area, fluid heat-capacity rate, and instantaneous phase-change state.

Other studies define charging efficiency as the ratio between the energy stored in the PCM and the energy supplied by the heat-transfer fluid during the charging period. These definitions are not directly interchangeable. For example, experimentally reported charging efficiencies may include heat losses to the environment, incomplete melting, sensible heating of the container, and thermal inertia of auxiliary components, whereas classical heat-exchanger effectiveness is primarily associated with the temperature change of the fluid.

Reported charging efficiencies for PCM systems commonly span a broad range. Zhang et al. (2020), for example, reported charging efficiencies between approximately 76.7% and

90.2% for an enhanced PCM storage system under different inlet-temperature conditions. Amin et al. (2012) obtained optimized energy-storage effectiveness values in the range of approximately 68–75%. These results indicate that values close to 0.80 are achievable in suitably designed systems, but they should not be interpreted as universally representative of all PCM heat exchangers.

For the simplified thermodynamic model developed in the present work, a nominal charging effectiveness of 0.80 was adopted. This parameter represents the fraction of the temperature-constrained recoverable exhaust heat that is effectively transferred to the PCM. The value should therefore be interpreted as an overall charging effectiveness factor rather than as a rigorously calculated effectiveness obtained from a detailed ε -NTU or UA-based heat-exchanger model.

The adopted value represents a high-performance but non-ideal heat-transfer configuration. Its use is consistent with the upper portion of the range reported for enhanced PCM storage systems, although direct experimental validation would be required for the specific exhaust-gas and LiCl–LiOH configuration proposed in this study. To address this uncertainty, the influence of charging effectiveness is evaluated through sensitivity analysis.

Therefore, the thermal performance of the proposed storage system should not be attributed exclusively to the latent heat of the selected PCM. Practical operation will depend on the combined effects of PCM properties, gas-side convection, exchanger geometry, thermal contact resistance, heat losses, and phase-change dynamics. Future studies should determine the required heat-transfer area, overall heat-transfer coefficient, exhaust-side pressure drop, PCM temperature distribution, and effective charging and discharging rates through detailed numerical and experimental investigation.

2.1.5 State of the Art: Series Hybrid Trucks with Biofuel-Powered Range Extenders

The decarbonization of road freight transport is characterized by a dual and parallel trajectory: the rapid advancement of Battery Electric Trucks (BETs) and the persistent search for transitional solutions for segments where full electrification faces structural barriers. The ultimate goal for zero tailpipe emissions is BETs, but their widespread use in long-haul freight is limited by high initial costs, short range, payload penalties due to large battery packs, and the need for ultra-high-power charging infrastructure. Developing economies and specific logistical operations face particularly acute challenges, which create a critical niche for advanced hybrid architectures.

Most recent research on hybrid electric trucks (HETs) focuses on parallel or power-split arrangements and energy management systems to improve fuel efficiency. A significant yet underexplored area exists within series hybrid systems that utilize ICEs only as small, high-efficiency generators (range extenders) fueled by low-carbon biofuels. By separating the engine from the drivetrain, this technology allows the engine to work at its best Brake Specific Fuel Consumption (BSFC) level all the time, which makes the process of turning fuel into electricity more efficient. This technology, when integrated with a sustainable fuel such as hydrous ethanol, can realize significant Well-to-Wheel (WTW) greenhouse gas (GHG) reductions by utilizing current gasoline distribution networks and circumventing the grid reliance and mineral criticality linked to BETs.

The author of this study confirmed this gap in the research through bibliometric analysis, published in a review article (GOMES *et al.*, 2025) This observation indicates that the integrated application of these technologies remains a relatively unexplored research area. Although the technical viability of individual components is well-established, a holistic assessment of their combined techno-economic and environmental performance, especially for medium- and heavy-duty truck applications, is lacking. The review article main findings were:

- Series hybrids allow for a significant downsizing of the battery pack, reducing the vehicle's cost, weight, and lifecycle environmental burden associated with battery production;
- The use of alcohols (e.g., ethanol) in dedicated generator engines benefits from their high octane number and heat of vaporization, enabling higher CRs and improved thermal efficiency in optimized combustion cycles (e.g., Atkinson or Miller cycles);
- The potential to integrate ORC systems or thermoelectric generators to recapture waste heat from the ICE generator exhaust and coolant presents a further avenue to increase the system's overall energy efficiency by 10-15%, a critical factor for total cost of ownership (TCO) competitiveness.

In this context, the present dissertation positions itself at this research frontier. The dissertation seeks to enhance the field by developing and analyzing a novel hybrid vehicle design that employs a hydrous ethanol-powered range extender and incorporates WHR. The study will employ integrated analyses of energy systems, vehicle dynamics, and economic variables to assess the system's efficiency, emissions, and cost-effectiveness, particularly in regions such as Brazil that possess clean electricity and ample biofuel resources. This dissertation seeks to establish a scientifically sound framework for evaluating a practical and

sustainable transition pathway for the freight sector, bridging the gap between existing ICE technology and a completely electric future by resolving the identified gap.

2.1.6 Screw Expanders

Screw expanders are positive-displacement machines that convert the pressure and enthalpy of a working fluid into mechanical power through the rotation of intermeshing helical rotors. Unlike turbines, in which power transfer is strongly associated with high fluid velocities, screw expanders generate work mainly through the pressure forces acting on the rotor surfaces as the confined working-fluid volume increases. Their lower internal fluid velocities make them less sensitive to droplet impingement and erosion, allowing operation with two-phase mixtures and wet steam conditions that would generally be unsuitable for conventional turbines.

This characteristic makes screw expanders particularly attractive for small- and medium-scale Rankine cycle systems, waste heat recovery, geothermal power generation, and other applications involving variable thermal sources. Smith et al. (2011) demonstrated that screw expanders can operate with wet steam and proposed their use for recovering energy from internal combustion engine exhaust gases. According to the authors, the presence of liquid during expansion has a comparatively limited effect on the operating principle and efficiency of these machines because the fluid velocities are substantially lower than those found in turbomachinery. Therefore, screw expanders may tolerate a progressive reduction in steam quality during expansion without the severe blade-erosion problems associated with steam turbines.

Smith et al. (2011) also compared a screw-driven wet-steam Rankine cycle with turbine-driven Organic Rankine Cycle systems for exhaust-gas heat recovery. The analysis considered an isentropic expander efficiency of 75% for the screw expander and indicated that the wet-steam configuration could achieve power outputs and cycle efficiencies comparable to, or higher than, the investigated ORC configurations. Additional advantages included the use of water as a low-cost, non-flammable, and environmentally benign working fluid, lower feed-pump power requirements, and the possibility of direct coupling to conventional generators at relatively moderate rotational speeds.

Despite these advantages, screw expanders are affected by internal leakage, friction, pressure losses, thermal deformation, rotor-profile design, built-in volume ratio, rotational speed, and clearance between the rotors and casing. Their performance may also decrease when the operating pressure ratio differs substantially from the machine's design value, resulting in

under-expansion or over-expansion. Furthermore, the relatively low density of steam at low exhaust pressures can lead to large volumetric flow rates and consequently increase the required expander size. This limitation can be mitigated by selecting a higher condensation pressure, using multiple expansion stages, or adapting the expander geometry to the expected operating conditions.

For the PCM–Rankine subsystem investigated in the present study, the screw expander provides an appropriate alternative to a conventional turbine because it is suitable for relatively low power levels and can accommodate the formation of liquid during expansion. Although the working fluid enters the expander in a superheated state in the proposed model, it may leave the machine as wet steam. The ability of screw expanders to tolerate two-phase outlet conditions therefore supports the adopted Rankine cycle configuration and reduces the need for excessive inlet superheating solely to maintain dry steam throughout the expansion process.

2.1.7 Steam cycles with screw expanders

Positive displacement machines, particularly screw expanders, have emerged as a promising alternative to conventional turbines for small- and medium-scale power generation systems. Unlike turbomachinery, where energy transfer is primarily associated with high fluid velocities and momentum variation, screw expanders operate through volumetric expansion caused by pressure forces acting directly on the rotor surfaces. Consequently, fluid velocities inside these machines are significantly lower than those observed in turbines, reducing aerodynamic losses and sensitivity to two-phase flow effects.

One of the major advantages of screw expanders is their ability to operate under wet-steam conditions without severe performance degradation. The presence of liquid droplets during expansion has a considerably smaller impact on operation compared to conventional steam turbines, which are highly sensitive to blade erosion and flow instability. This characteristic makes screw expanders particularly suitable for waste heat recovery systems operating with moderate temperatures and low-to-medium pressure ratios.

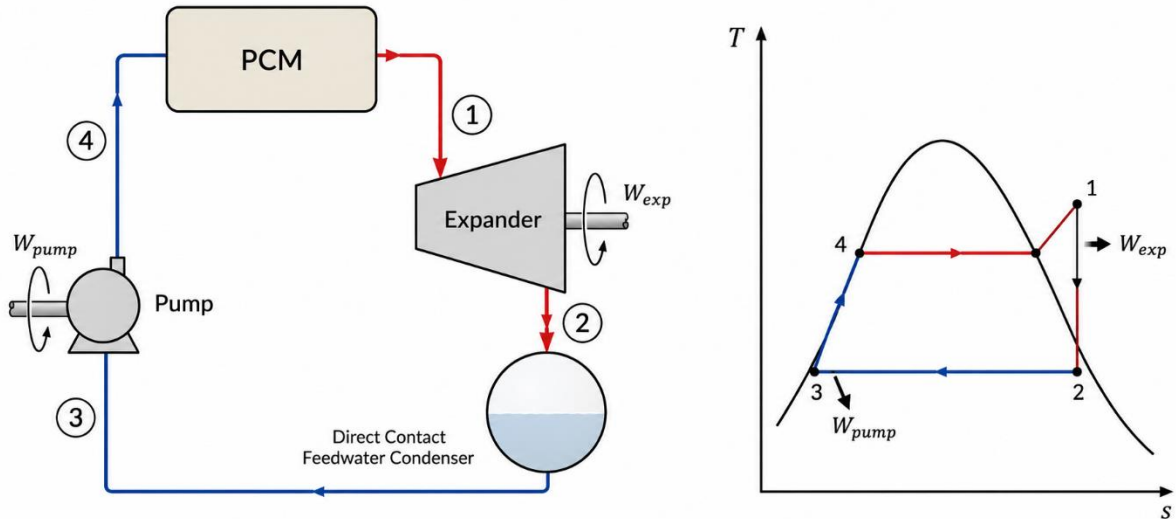
In the present work, the bottoming cycle is modeled as a steam Rankine cycle operating with moderately superheated steam at the expander inlet and slightly wet steam at the outlet, thereby limiting thermal stresses and avoiding unnecessarily high operating temperatures.

Steam presents several advantages as a working fluid for the proposed application. Even at relatively moderate temperatures, water vapor exhibits high specific enthalpy and significant

energy density compared to many organic working fluids. Furthermore, the pumping work required for pressurization remains relatively low due to the liquid-phase compression process.

Another important consideration in steam-based screw expander systems is the condensing pressure. Very low condensing pressures may lead to excessively large volumetric flow rates, resulting in increased expander size. In the proposed configuration, the expander outlet pressure is maintained at atmospheric conditions, representing a practical compromise between maximizing expansion work and ensuring equipment feasibility. The steam condenser is designed as an open vessel operating as a direct-contact feedwater heat exchanger as illustrated in Fig. 4.

Figure 4 - Rankine cycle based on steam screw expanders.



Source: Adapted from SMITH et al. (2011)

Therefore, steam screw expanders constitute an attractive solution for PCM-assisted waste heat recovery systems, particularly in applications involving intermittent operation and thermal energy storage, such as the proposed Carnot Battery Combined Cycle architecture.

Several studies have demonstrated the suitability of screw expanders for wet-steam applications in waste heat recovery systems. According to SMITH et al. (2011), screw expanders can operate efficiently under two-phase expansion conditions due to their positive displacement characteristics and reduced sensitivity to liquid droplet formation compared to conventional turbines.

Additionally, the presence of moderate liquid fractions during expansion may contribute to internal sealing effects between rotor clearances, partially reducing leakage losses and improving volumetric performance under certain operating conditions

The use of moderate superheating in the present work was therefore selected as a compromise between improving vapor quality at the expander outlet and avoiding excessively high operating temperatures and pressures.

Furthermore, screw expanders exhibit favorable operational characteristics for intermittent and transient operating conditions due to their relatively robust mechanical behavior and reduced sensitivity to flow fluctuations. This feature is particularly relevant for the proposed PCM-assisted Carnot Battery Combined Cycle, in which the Rankine subsystem may operate under non-steady thermal discharge conditions associated with the charging and discharging processes of the thermal energy storage system.

3. METHODOLOGY

3.1 General Modeling Approach

This work employs a deterministic thermodynamic modeling methodology to evaluate the performance of an ethanol-fueled internal combustion engine integrated with a latent thermal energy storage (TES) system based on a phase change material (PCM) and a steam Rankine cycle (RC) for waste heat recovery (WHR). The proposed research framework and the energy flow among the engine, PCM storage unit, and Rankine cycle are illustrated in Figure 5. The thermodynamic model was developed based on mass, energy, and exergy balances to quantify the performance of each subsystem and the overall CBCC.

Unlike conventional WHR systems, in which the bottoming cycle operates simultaneously with the internal combustion engine, the proposed CBCC introduces temporal decoupling through latent thermal energy storage. During the charging stage, the internal combustion engine operates for a predetermined period, producing electrical power while transferring part of the exhaust-gas thermal energy to the PCM, where it is stored as latent heat. After the charging process is completed, the engine is switched off and the stored thermal energy is discharged to the steam Rankine cycle, which continues generating electrical power independently of engine operation. This sequential operating strategy enables the concentration of thermal energy into a shorter discharge period, increasing the instantaneous power available to the Rankine cycle while decoupling heat recovery from engine operation. This architecture represents a novel contribution in the context of ethanol-based range extenders.

All simulations were conducted utilizing the Engineering Equation Solver (EES), which facilitates the resolution of coupled nonlinear algebraic equations through thermodynamic property libraries.

The analysis is performed under steady-state conditions. At this level, transient effects, dynamic thermal inertia, and control strategies are not considered. The modeling framework is based on the conservation of mass and energy (First Law of Thermodynamics), neglecting variations in kinetic and potential energy. Accordingly, the governing equations are expressed as:

$$\Sigma \dot{m}_{in} = \Sigma \dot{m}_{out} \quad (1)$$

$$\dot{Q} - \dot{W} + \Sigma \dot{m}_{in} h_{in} - \Sigma \dot{m}_{out} h_{out} = 0, \quad (2)$$

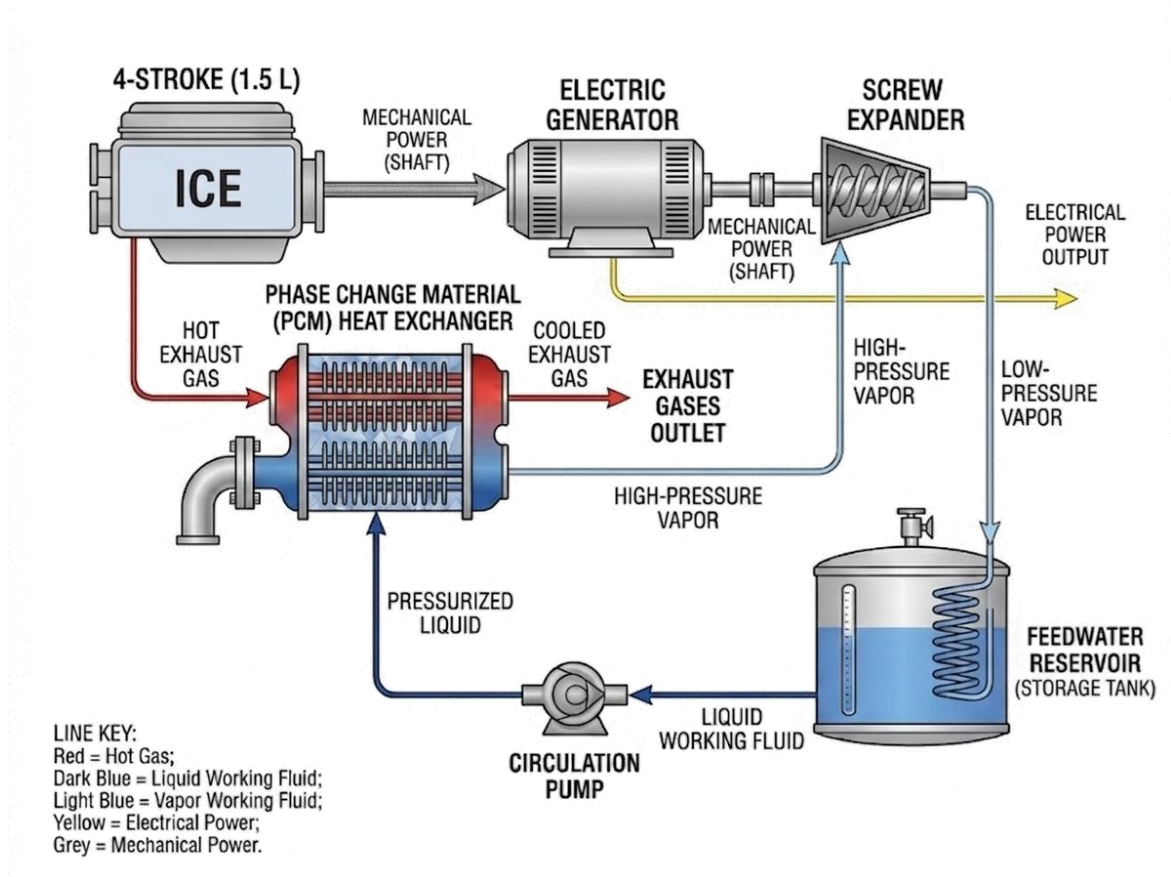
where $\dot{m}$ is the mass flow rate, $\dot{Q}$ is the heat transfer rate, $\dot{W}$ is the work rate, and h is the specific enthalpy. These equations constitute the fundamental balance equations applied to each control volume of the proposed CBCC, including the internal combustion engine, heat exchangers, pump, screw expander, condenser, and PCM thermal storage system. In addition, the Second Law of Thermodynamics is applied through exergy balances to evaluate the quality of energy flows, quantify irreversibilities, and determine the exergy efficiency of the proposed CBCC architecture. The ICE and Rankine subsystems are modeled under steady-state assumptions, whereas a simplified pseudo-transient energy accumulation model represents the PCM storage.

The integrated system is divided into three subsystems:

- Ethanol-fueled Otto-cycle internal combustion engine;
- PCM-based TES system;
- Steam RC with screw expander.

Each subsystem is described in the following sections. Initially, a commercially available ethanol-fueled engine (Toyota 2NR-FBE, 1.5 L) was selected as a representative range extender to establish the boundary conditions for the subsequent development of the proposed CBCC architecture. The engine was thermodynamically modeled using EES to determine its main operating characteristics, including brake power, fuel consumption, exhaust gas temperature, exhaust mass flow rate, and the thermal and exergy potential of the exhaust stream. These results were subsequently used as boundary conditions for the design and analysis of the downstream subsystems, particularly the PCM-based thermal energy storage system and the Rankine cycle for waste heat recovery.

Figure 5 – CBCC - ethanol-fueled internal combustion engine integrated with a latent TES system based on a PCM and a steam RC for WHR.



Source: Author's own elaboration.

In contrast to traditional WHR systems, the TES establishes a temporal separation between energy acquisition and application. The engine delivers heat consistently at a modest rate at high temperature, while the TES facilitates energy storage for subsequent release at an elevated thermal power level. This method allows the RC to deliver mechanical power close to that of the topping cycle.

It is crucial to highlight that the system's performance is contingent upon the heat transfer efficiency of the TES heat exchanger, including thermal conductivity, surface area, and phase-change dynamics when using PCM.

The thermal storage subsystem was evaluated through two complementary levels of modeling. First, an energy-capacity formulation was employed to quantify the thermal energy accumulated during the charging period and the corresponding PCM mass. In this preliminary approach, the storage system is treated as an ideal thermal buffer, in which the energy accumulated in the PCM is calculated as:

$$E_{PCM} = \dot{Q}_{PCM} t_{charge}, \quad (3)$$

where E_{PCM} is the energy stored in the PCM (kJ), $\dot{Q}_{PCM}$ is the charging thermal power (kW), and t_{charge} is the charging time (s). Assuming that the storage process is dominated by latent heat, the PCM mass is determined from:

$$m_{PCM} = \frac{E_{PCM}}{\Delta h_{fus}}, \quad (4)$$

where Δh_{fus} is the latent heat of fusion of the selected PCM (kJ/kg). This preliminary formulation is useful for illustrating the temporal decoupling concept because it relates the energy recovered from the exhaust gases to the storage capacity independently of the heat-transfer limitations. Subsequently, the model was refined by introducing a temperature constraint between the exhaust gases and the PCM. To ensure a feasible heat transfer during the charging process, the exhaust-gas outlet temperature was constrained according to:

$$T_{exh,out} \geq T_{PCM,eff} + \Delta T_{min}, \quad (5)$$

where $T_{exh,out}$ is the exhaust-gas outlet temperature, $T_{PCM,eff}$ is the effective phase-change temperature of the PCM, and ΔT_{min} is the minimum temperature approach adopted in the charging heat exchanger. This constraint ensures that heat can only be transferred while a sufficient temperature difference exists between the exhaust gases and the PCM. Consequently, only the fraction of the exhaust-gas thermal energy above this limiting temperature is considered recoverable for thermal storage. Based on this constraint, the recoverable thermal power is calculated as:

$$\dot{Q}_{rec} = \dot{m}_{exh} c_{p,exh} (T_{exh,in} - T_{PCM,eff} - \Delta T_{min}) \quad (6)$$

The thermal power effectively stored in the PCM is then expressed as:

$$\dot{Q}_{PCM} = \epsilon_{HX} \dot{Q}_{rec}, \quad (7)$$

where $\dot{m}_{exh}$ is the exhaust-gas mass flow rate, $c_{p,exh}$ is the specific heat of the exhaust gases, $T_{exh,in}$ is the exhaust-gas inlet temperature, $T_{PCM,eff}$ is the effective PCM phase-change temperature, ΔT_{min} is the minimum temperature approach imposed in the charging heat exchanger and ϵ_{HX} is the effectiveness of the charging heat exchanger.

This refined formulation was adopted for the final performance assessment because it distinguishes the total thermal energy carried by the exhaust gases from the fraction that can be

effectively transferred to the PCM at the required temperature level. During the discharge process, the thermal power supplied to the Rankine evaporator is obtained by dividing the stored energy by the prescribed discharge time:

$$\dot{Q}_{discharge} = \frac{E_{PCM}}{t_{discharge}}, \quad (8)$$

which represents the thermal power available to the bottoming cycle.

In this formulation, the thermal power available for charging is limited by the exhaust-gas inlet temperature, the effective PCM phase-transition temperature, and a minimum temperature approach in the charging heat exchanger. The refined formulation was used for the final performance assessment because it distinguishes the total thermal energy carried by the exhaust gases from the fraction that can be effectively transferred at the required temperature level.

3.2 Internal Combustion Engine Model

3.2.1 Engine Configuration and Assumptions

A real four-stroke, spark-ignition internal combustion engine operating according to the real Otto cycle at full-load conditions and fueled with hydrated ethanol (C_2H_5OH) was adopted as the ICE model. The Otto cycle was selected due to the high octane rating of ethanol, which enables operation at elevated compression ratios while maintaining knock resistance, thereby allowing the development of a relatively simple engine configuration capable of producing substantial power relative to its size. The model was developed under the assumptions of steady-state operation, ideal gas behavior for both air and combustion products, complete combustion, and a homogeneous air–fuel mixture within the cylinder. Compression and expansion processes were represented using a polytropic approach, assuming negligible heat transfer during these stages. In addition, mechanical losses associated with friction, pumping, and auxiliary components were accounted for through a variable mechanical efficiency formulation based on friction mean effective pressure correlations.

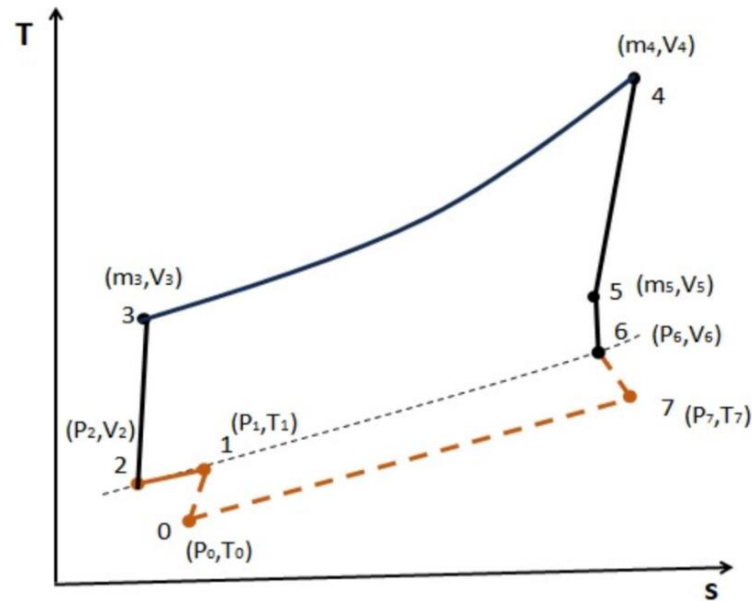
Thermodynamic modeling was based on the solution of a set of equations representing a standard four-stroke Otto cycle, following Martins (2013), as illustrated in Fig. 6. Basically, the cycle is comprised of a polytropic compression (stage 2 to 3), energy addition at near constant volume (state 3 to 4), and polytropic expansion with heat losses to the cylinder wall (state 4 to 5) and cylinder exhausting (state 5 to 6). To close the ideal cycle, heat rejection would takes place after the flue gas expansion in the turbocharger, process 7-1. Processes 0-1 and 6-7

take place in the engine turbocharger and 1-2 in the aftercooler. The working fluid in states 1 to 3 is air, for simplicity and the products of the fuel oxidation comprises the fluid mixture from state 4 up to state 7. Process 2 to 3 is irreversible and the expansion from 4 to 5 accounts for the heat losses to the cylinder walls.

In the model, heat transfer to the wall is significant during expansion (ε_p). Additional engines losses related to the exhaust (ε_{exh}) and intake (ε_{Rl}) are also considered, as suggested by Martin et al. (2015).

The thermodynamic model of the baseline ethanol-fueled range extender engine, implemented in the EES software system, provide performance data of the ICE-cycle and relevant information to design the suggested TES-Rankine WHR system. Appendix 1 includes the EES code for modeling the motor.

Figure 6 - Otto cycle and main thermodynamic states in the temperature and entropy chart.



Source: Adapted from Martins (2013).

3.2.2 Geometric data of the ICE

The proposed model employs a four-cylinder spark-ignition engine operating at a constant rotational speed. For the initial engine modeling, a widely used model in the country was chosen: the Toyota 2NR-FBE engine, a 1.5-liter engine used in the Etios and Yaris models. The primary geometric parameters are specified below:

- Number of cylinders: 4;
- Bore (B): 72.5 mm;

- Stroke (S): 90.6 mm;

The operating parameters are:

- CR: 14;
- Turbocharger pressure: 1.5 bar.
- ϕ (equivalence ratio): 0.8;
- Engine speed: 4500 rpm.

The displacement volume of each cylinder is determined as follows:

$$V_d = \frac{\pi}{4} B^2 S, \quad (9)$$

B represents the cylinder bore, whereas S denotes the piston stroke. The total engine displacement volume ($V_{d,total}$) is calculated by multiplying the displacement of a single cylinder by the total number of cylinders.

The clearance volume is determined from the CR:

$$V_c = \frac{V_d}{CR - 1} \quad (10)$$

The cylinder volume at bottom dead center (V_{BDC}) and top dead center (V_{TDC}) are then given by:

$$V_{BDC} = V_d + V_c \quad (11)$$

$$V_{TDC} = V_c \quad (12)$$

The geometric parameters delineate the control volumes employed in the thermodynamic cycle simulation.

3.2.3 Optimization Procedure – Genetic Method

A genetic algorithm (GA) optimization procedure was implemented in the Engineering Equation Solver (EES) to optimize the thermal energy storage (TES)–Rankine subsystem. In the final formulation adopted in this study, the internal combustion engine was treated as a fixed thermal source, whereas the GA was applied exclusively to the PCM–Rankine subsystem. This

approach isolates the influence of the thermal storage and bottoming cycle parameters on the overall system performance while maintaining the engine operating conditions unchanged.

The optimization procedure was developed to determine the operating conditions that maximize the thermodynamic performance of the PCM-assisted Rankine cycle while satisfying the operating constraints of the screw expander and the thermal energy storage system. The optimization variables considered were the Rankine cycle evaporation pressure, condenser pressure, superheating degree at the expander inlet, and PCM charging time.

The genetic algorithm was configured using a population of 4–5 individuals per generation, 6–8 generations, and a maximum mutation rate of 5–8%. These parameters were selected to provide an adequate balance between computational effort and convergence stability, considering the relatively small optimization problem involving four decision variables. The following optimization variables were selected:

- Expander inlet pressure (P_1);
- Expander outlet pressure (P_2);
- Degree of superheating ($\Delta t_{superheat}$).

These variables were selected because of their strong influence on Rankine cycle efficiency, vapor quality during expansion, thermal storage capacity, and operational flexibility.

The optimization objectives included:

- Maximization of the thermodynamic performance of the PCM–Rankine subsystem ($\dot{W}_{net}$);
- Maximization of Rankine cycle efficiency (η_{RC}).

The optimization process follows the steps below:

1. Initialization: A population of candidate solutions is generated with random values of the optimization variables within predefined physical limits;
2. Evaluation: Each candidate solution is evaluated using the thermodynamic PCM–Rankine model developed in EES;
3. Selection: The best-performing individuals are selected according to the objective function value;
4. Crossover and Mutation: New solutions are generated through recombination and mutation operators, maintaining diversity in the search space;
5. Convergence: The algorithm iterates until convergence criteria are satisfied, such as minimal improvement between generations or the maximum number of iterations.

The optimization ranges adopted in the present study are:

$$800 \text{ kPa} \leq P_1 \leq 1200 \text{ kPa}$$

$$110 \text{ kPa} \leq P_2 \leq 200 \text{ kPa}$$

$$0 \text{ }^\circ\text{C} \leq \Delta t_{\text{superheat}} \leq 20 \text{ }^\circ\text{C}$$

These limits ensure physically realistic operating conditions and prevent excessive moisture formation during the expansion process. The upper limit adopted for the expander inlet pressure was established based on commercially available screw compressor technology. According to technical specifications provided by DENAIR screw compressor systems, pressure levels around 12 bar represent a practical operating limit for standard industrial screw machines. Since the present study considers the possibility of operating a commercially available screw compressor in reverse mode as a screw expander, the optimization range was intentionally restricted to values compatible with existing market equipment. This approach increases the technological feasibility of the proposed system and avoids optimization results that would require custom-designed expansion machinery.

3.2.4 Compilation of results – Genetic Method

In this work, the optimization strategy follows a weighted-function approach, whereby multiple competing objectives are combined into a single scalar objective function through weighting factors. This methodology is widely employed in multi-objective optimization problems because it allows the evaluation of trade-offs between different performance indicators (MARLER; ARORA, 2010).

In the present study, the weighted objective function was applied to the PCM–Rankine subsystem in order to simultaneously maximize electrical power generation and thermodynamic efficiency while maintaining suitable operating conditions for the screw expander and the thermal storage system.

The adopted objective function is expressed as:

$$F = 0.5 \left(\frac{\dot{W}_{RC,net}}{\dot{W}_{ref}} \right) + 0.3 \left(\frac{\eta_{RC}}{\eta_{ref}} \right) + 0.2 \left(\frac{x_2}{x_{ref}} \right) \quad (13)$$

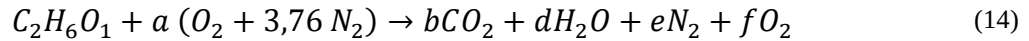
Where:

- $\dot{W}_{RC,net}$ is the net electrical power output of the Rankine cycle;
- η_{RC} is the Rankine cycle thermal efficiency;
- $\dot{W}_{ref}$, η_{ref} and x_{ref} are reference normalization values.

Reference values were selected based on the maximum values obtained during preliminary simulations and were used exclusively to normalize the objective function. The selected weighting factors were chosen to prioritize power generation while simultaneously preserving acceptable vapor quality and operational flexibility. The inclusion of vapor quality in the objective function is particularly important for screw expander operation, since excessive liquid fractions may negatively affect mechanical performance and volumetric efficiency.

3.2.5 Combustion Stoichiometry and Product Composition

Ethanol combustion was represented using a specific function implemented in the EES code. Accordingly, the reaction considered in the model is expressed as follows:



The stoichiometric coefficients are determined by the elemental balance of carbon, hydrogen, and oxygen. The mass fractions of the combustion products are derived from this formulation as follows. The mass fractions are utilized to assess the thermodynamic parameters of the exhaust gases via mixture property relations. The equivalency ratio ϕ (equation 18) denotes the condition of the air-fuel mixture and may vary from 0.6 to 1.0 in spark ignition engines.

3.2.6 Mass Flow Rate Calculations

Although volumetric formulations are widely used in internal combustion engine analysis, they are not adopted in the present work. Instead, the intake process is modeled using mass and energy balances, ensuring thermodynamic consistency with the cycle simulation implemented in EES.

$$\dot{m}_{air} = \eta_v \rho_{intake} V_{d,total} \frac{N}{2} \quad (15)$$

This formulation is widely used in internal combustion engine analyses (HEYWOOD, 1988). However, this approach assumes ideal volumetric filling and neglects thermodynamic effects during the intake process.

In this work, a more detailed approach is adopted based on mass and energy balances, following the methodology presented by STONE (1992). The intake process is modeled as an open system, where the inducted mass is determined from:

$$m_{in} = m_{air} + m_{fuel} \quad (16)$$

Combined with the first law of thermodynamics:

$$m_{in}h_{in} = P_{intake}(V_2 - V_1) + m_2u_2 - m_1u_1, \quad (17)$$

where:

- m_{in} is the mass of the inducted air–fuel mixture during the intake process (kg);
- h_{in} is the specific enthalpy of the inducted mixture at the intake conditions (kJ/kg);
- P_{intake} is the intake manifold pressure (kPa);
- V_1 is the cylinder volume at the beginning of the intake stroke (m³);
- V_2 is the cylinder volume at the end of the intake stroke (m³);
- m_1 is the mass contained in the cylinder at the beginning of the intake process (kg);
- m_2 is the mass contained in the cylinder at the end of the intake process (kg);
- u_1 is the specific internal energy of the cylinder contents at the beginning of the intake process (kJ/kg);
- u_2 is the specific internal energy of the cylinder contents at the end of the intake process (kJ/kg).

Equation 17 allows the intake mass to be determined consistently with the thermodynamic state of the working fluid, improving the accuracy of the model. The separation between air and fuel is performed using the oxidizer-to-fuel ratio (OF), defined as:

$$OF = \frac{m_{air}}{m_{fuel}} \quad (18)$$

Based on the chemical composition of the fuel, the stoichiometric oxidizer-to-fuel ratio (OF_{st}) defines the exact amount of air required for complete combustion.

In contrast, the actual oxidizer-to-fuel ratio (OF_{act}) corresponds to the real operating condition of the engine and is calculated from the target equivalence ratio (ϕ):

$$\phi = \frac{OF_{st}}{OF_{act}} \quad (19)$$

This formulation allows direct control of the air–fuel mixture in the model. In this study, a range of 0.7 to 1.0 was investigated. Lower equivalence ratio favors decreased engine exhaust gas temperature and power at slightly improved thermal efficiency relative to the stoichiometric condition.

During the intake process, fresh air enters the cylinder at the intake pressure and temperature provided by the turbocharging system. The mass entering the cylinder is determined from the energy balance:

$$m_{in}h_{in} = P(V_{BTC} - V_{TDC}) + m_2u_2 - m_1u_1 \quad (20)$$

The mass in the cylinder is given by $m_1 = V_{TDC}/v_1$, for state 1 and $m_2 = V_{BTC}/v_2$ for state 2.

The compression stroke is modeled as a polytropic process:

$$P_3V_3^{n_{comp}} = P_2V_2^{n_{comp}} \quad (21)$$

The polytropic exponent n_{comp} is approximated from the ratio of specific heats of air considering a non-adiabatic process and can be estimated from

$$n_{comp} = f_1 \frac{c_p}{c_v} \quad (22)$$

The parameter f_1 may vary between 0.93 and 0.98, indicating that the process is not perfectly adiabatic. Its value should be determined iteratively, adjusting it until the model predictions closely match the actual engine performance, when such data are available.

The work associated with compression is calculated from:

$$W_{comp} = \frac{P_3V_3 - P_2V_2}{1 - n_{comp}} \quad (23)$$

The combustion process is modeled as a constant-volume heat addition process occurring near top dead center. The energy balance during combustion is given by:

$$Q_{in} = m_4 u_4 - m_3 u_3 \quad (24)$$

The internal energy of the combustion products is evaluated as a mixture of the individual species contributions weighted by their mass fractions.

The expansion stroke (state 4 $\rightarrow$ state 5) is modeled as a polytropic process:

$$P_5 V_5^{n_{exp}} = P_4 V_4^{n_{exp}} \quad (25)$$

The expansion process (state 4 $\rightarrow$ state 5) is modeled as a polytropic transformation:

$$PV^{n_{exp}} = \text{constant} \quad (26)$$

In conventional simplified models, the polytropic exponent is typically assumed based on literature values for burned gases. However, in the present work, the exponent is not fixed a priori. Instead, the effective polytropic behavior emerges implicitly from the combined solution of the pressure–volume relation and the energy balance:

$$0 = W_{exp} + Q_{exp} + m_5 u_5 - m_4 u_4 \quad (27)$$

The cycle energy balance is given by the ratio of relevant parameter to the energy input to the cycle (Q_{LHV}). The relative contribution are giving by the indicated engine efficiency ($\eta_{ind} = W_{ind}/Q_{LHV}$), engine cooling losses through cylinder walls ($\varepsilon_P = Q_{exp}/Q_{LHV}$), losses to the engine exhaust ($\varepsilon_E = Q_{exh}/Q_{LHV}$), and intake breathing losses ($\varepsilon_R = W_R/Q_{LHV}$). Losses from the exhaust (ε_E) are estimated from the relative enthalpy of the state 7 in Fig. 6. For a turbo-charge engine, intake losses (ε_R) are negligible.

As in Martin et al. (2015), cooling losses depend on the charging coefficient (r), mean piston velocity (w), piston diameter (D), and the engine compression ratio (χ),

$$\varepsilon_P = 0.015(rwD)^{-0.2}(\chi^{0.8} + 3\chi^{-0.4}) \quad (28)$$

By combining the heat losses to the cylinder walls with the energy balance during expansion, the polytropic exponent (η_{exp}). can be determined. This approach allows the expansion process to account for heat transfer effects and variations in thermodynamic properties of the combustion products, leading to a more realistic representation of the in-cylinder process compared to constant-exponent assumptions.

$$W_{exp} = \frac{P_5 V_5 - P_4 V_4}{1 - \eta_{exp}} \quad (29)$$

This process represents the main work-producing stage of the engine.

During the exhaust process, the burned gases are discharged from the cylinder to the exhaust manifold.

At state 6, the gas temperature is taken as the exhaust temperature, (T_{exh}) Its value is obtained from the entropy balance under the assumption that the residual gases inside the cylinder undergo an isentropic process. The exhaust mass flow rate is obtained from the overall mass balance of the engine:

$$\dot{m}_{exh} = \dot{m}_{air} + \dot{m}_{fuel} \quad (30)$$

Assuming steady-state operation and negligible mass accumulation inside the control volume.

3.2.7 Brake Power Calculation

The net work per cycle produced by the engine is obtained from the difference between expansion and compression work:

$$W_{net} = W_{exp} + W_{comp} \quad (31)$$

The engine power output is determined from the number of thermodynamic cycles occurring per unit time. For a four-stroke engine, each cylinder completes one cycle every two crankshaft revolutions. Therefore, the brake power is calculated as

$$\dot{W}_{net} = W_{net} \frac{N_{cyl}}{2 \cdot period} \quad (32)$$

Where:

- N_{cyl} is the number of cylinders;

- N is the engine rotational speed in revolutions per second;

Brake power is subsequently obtained by applying the mechanical efficiency to account for friction and auxiliary losses.

$$\dot{W}_{brake} = \eta_{mec} \cdot \dot{W}_{net} \quad (33)$$

In this work, the friction mean effective pressure is estimated using an empirical correlation that includes contributions from baseline friction, cylinder pressure effects, and piston speed. This approach allows the mechanical efficiency to vary with engine operating conditions, providing a more realistic representation compared to constant-efficiency assumptions.

The mechanical efficiency, in terms of mean pressures can be estimated by

$$\eta_{mec} = 1 - \bar{P}_f / \bar{P}_i \quad (34)$$

In equation 34, mean piston friction pressure due to piston rings and lubrication is given by

$$\bar{P}_f = \bar{P}_{f,0} + k_s(2\bar{P}_c + \bar{P}_i)k_w w/D \quad (35)$$

For liquid fuels, $\bar{P}_{f,0}$, is taken as 70 kPa, and k_s and k_w are, respectively, 0.02 and 0.7 kPa-s (Martins, 2013). The mean compression and indicated pressures are given, respectively, by

$$\bar{P}_c = r p_0 (T_1/T_0) (\chi^\xi - 1)/\xi \quad (36)$$

$$\bar{P}_i = \eta_{ind} r \rho_0 \phi LHV / OF_{stoi}, \quad (37)$$

where ρ_0 is the density of the fresh charge (air–fuel mixture) at the intake conditions, evaluated from the intake pressure and temperature using the ideal gas equation and

$$\xi = 0.277 + 0.06[1 - \phi + (1 - 0.1\chi^{0.5})(1 - \phi)^{2.5}] \quad (38)$$

The convenience factor is the exponent of the compression ratio in the Otto cycle thermal efficiency for different equivalence ratio (Martin et al., 2015).

3.2.8 Exhaust Gas and Heat Availability

Based on the thermodynamic state at the end of the exhaust process, the exhaust-gas temperature is calculated as follows:

$$T_{exh} = T_6 \quad (39)$$

The thermal power available in the exhaust gases is calculated using the sensible enthalpy difference relative to the ambient reference temperature:

$$\dot{Q}_{exh} = \dot{m}_{exh} c_{p,exh} (T_{exh} - T_{ref}) \quad (40)$$

Where $c_{p,exh}$ is the specific heat capacity of the combustion product mixture and T_{ref} is the reference temperature of 363 K (90 °C), which represents a practical lower limit for effective heat recovery in thermal systems (MORAN; SHAPIRO; BAILEY, 2018; SAIDUR; AHAMED; MASJUKI, 2010). $T_{ref} = 363$ K was adopted as the practical lower temperature limit for heat recovery, which represents a practical lower limit for efficient heat recovery rather than the environmental dead state, whereas $T_0 = 298$ K represents the environmental dead-state condition used in exergy calculations. This heat stream represents the energy source used for charging the phase change material thermal storage system and for driving the RC subsystem.

3.3 Engine Efficiency

The chemical energy supplied by the fuel is determined from the fuel mass flow rate and the lower heating value (LHV). The LHV value of hydrous ethanol, used in the engine modeling, is 26.36 MJ/kg.(ANP, 2025).

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot LHV \quad (41)$$

The brake thermal efficiency of the engine is defined as:

$$\eta_{bth} = \frac{\dot{W}_{brake}}{\dot{Q}_{fuel}} \quad (42)$$

This parameter quantifies the fraction of fuel chemical energy converted into useful mechanical power.

3.4 Exergy Analysis

Exergy is defined as the maximum theoretical useful work that can be obtained from a system as it comes into equilibrium with a reference environment. Unlike energy, which is conserved, exergy is destroyed due to irreversibility associated with real processes, such as heat transfer across finite temperature differences, friction, and mixing.

Flow exergy can be separated into physical and chemical components. Within the scope of this study, physical exergy is used to assess the thermal potential of the exhaust gases for waste heat recovery, whereas chemical exergy is considered in the calculation of fuel exergy.

In internal combustion engines, a significant fraction of the fuel chemical energy is rejected through the exhaust gases. However, only part of this energy is thermodynamically available for conversion into useful work. The exergy analysis provides a more realistic measure of the recoverable energy potential compared to a simple energy balance.

Exergy analyses of internal combustion engines reported by Heywood (2018), Rakopoulos and Giakoumis (2006), and Caton (2018) have shown that the ratio of exhaust-gas exergy to exhaust-gas thermal energy generally lies between 40% and 60%, depending primarily on exhaust temperature, engine load, and combustion characteristics. This indicates that, although a considerable amount of thermal energy remains in the exhaust gases, only part of this energy can be converted into useful work because of the limitations imposed by the Second Law of Thermodynamics.

Therefore, exergy analysis is an essential tool for evaluating the real potential of WHR systems, such as RCs, particularly when coupled with TES systems.

In ICEs, a significant portion of the fuel's chemical exergy is degraded during combustion and rejected through exhaust gases and cooling systems. Therefore, exergy analysis enables the identification of true efficiency limits and highlights opportunities for improving energy utilization, particularly in WHR applications (DINCER; ROSEN, 2013).

The exergy analysis of the exhaust gases was performed to evaluate the thermodynamic quality of the energy available for recovery. The physical exergy of the exhaust gas stream was calculated using:

$$\dot{E}_x = \dot{m}[(h - h_0) - T_0(s - s_0)] \quad (43)$$

Where h and s are the specific enthalpy and entropy of the flow; h_0 and s_0 are the properties evaluated at environmental conditions; T_0 is the ambient temperature; and $\dot{m}$ is the mass flow rate. The thermodynamic properties of the exhaust gases were derived using mixture

relations based on the composition of the combustion products. The reference environment was established at $T_0 = 298$ K and $P_0 = 1$ atm.

Exergy efficiency is calculated using the equation below and represents the ratio between the useful mechanical work delivered by the engine (brake power) and the chemical exergy of the fuel supplied to the system. Unlike conventional thermal efficiency, which is based solely on energy conservation, exergy efficiency accounts for the quality of energy and the irreversibilities associated with real processes, providing a more realistic measure of system performance (DINCER; ROSEN, 2013; MORAN; SHAPIRO; BAILEY, 2018).

$$\eta_{ex} = \frac{\dot{W}_{brake}}{\dot{E}x_{fuel}} \quad (44)$$

Where $\dot{W}_{brake}$ is the brake power output of the engine and $\dot{E}x_{fuel}$ is the rate of chemical exergy supplied by the fuel. The chemical exergy of the fuel can be estimated as a function of its lower heating value (LHV), adjusted by a correction factor that accounts for the fuel composition (SZARGUT; MORRIS; STEWARD, 1988).

In ICEs, the exergy efficiency is typically lower than the corresponding thermal efficiency due to the significant exergy destruction occurring during combustion, heat transfer, and fluid flow processes. These irreversibilities degrade the ability of the system to convert the available work potential into useful output. Consequently, exergy efficiency serves as a key indicator for identifying performance limitations and guiding the improvement of energy conversion systems, particularly in applications involving WHR and advanced thermodynamic cycles (DINCER; ROSEN, 2013; HEYWOOD, 1988).

In the context of the present study, the exergy efficiency is used to evaluate the effectiveness of the ethanol-fueled range extender and to quantify the potential benefits of integrating the CBCC, especially in terms of improved utilization of the fuel's thermodynamic potential.

Based on the sensible enthalpy difference between the exhaust gases and the reference state, the exhaust-energy rate was calculated according to Equation 40. To further assess the thermodynamic quality of this energy stream, the ratio between exergy and energy was introduced as a performance indicator:

$$\dot{E}x_{fuel} = \dot{m}_{fuel} \cdot \beta \cdot LHV \quad (45)$$

$$\psi = \frac{\dot{E}_{x_{exh}}}{\dot{Q}_{exh}} \quad (46)$$

Where the physical exergy rate of the exhaust gases is evaluated according to Szargut, Morris, and Steward (1988), and the corresponding thermal energy rate is obtained from the exhaust sensible enthalpy difference. This dimensionless parameter, often referred to as the exergy-to-energy ratio, represents the fraction of the exhaust thermal energy that is theoretically convertible into useful work under ideal reversible conditions.

For high-temperature exhaust gases, this ratio tends to be relatively high, indicating a greater potential for energy recovery. However, as the exhaust temperature approaches the ambient level, the exergy content decreases significantly, even if the energy content remains substantial. It should be emphasized that the exergy-to-energy ratio represents a theoretical upper limit for useful work recovery and does not correspond to the actual recoverable power of the system (DINCER; ROSEN, 2013).

In the context of the present study, the parameter ψ is used to evaluate the suitability of the exhaust gases as a heat source for the PCM-based thermal storage system and the subsequent RC. It provides a more realistic measure of the recoverable energy potential and supports the assessment of the effectiveness of the proposed CBCC architecture in utilizing waste heat.

The Exergy efficiency of the cycle is defined as:

$$\eta_{ex} = \frac{\dot{W}_{RC,net}}{\dot{E}x_{source}} \quad (47)$$

Exergy analysis allows evaluating the thermodynamic quality of the recovered heat and identifying irreversibilities within the system (DINCER, ROSEN, 2013).

3.5 Thermal Storage Model

Part of the thermal energy available in the engine exhaust gases was recovered and stored in a phase change material (PCM)-based thermal energy storage system. The PCM acts as a thermal buffer by absorbing heat during engine operation and subsequently releasing the stored energy to the bottoming Rankine cycle. This intermediate storage stage enables temporal decoupling between exhaust-heat generation and power conversion.

In the proposed configuration, the exhaust-gas stream downstream of the engine exhaust turbine is used as the heat source during the PCM charging process. The storage model assumes that the thermal energy effectively transferred from the exhaust gases to the storage medium is

predominantly stored as latent heat during the phase transition. Sensible heating of the PCM, containment structure, heat exchanger, insulation, and auxiliary components was not explicitly included in the storage-capacity calculation.

For the present model, a binary mixture containing 32 mol% lithium chloride (LiCl) and 68 mol% lithium hydroxide (LiOH) was adopted as the PCM. Selection of this composition was supported by the experimental characterization reported by Hassan et al. (2023). The authors investigated several LiCl–LiOH compositions and identified the 32 mol% LiCl–68 mol% LiOH mixture as a suitable candidate for medium-temperature latent heat thermal energy storage because of its phase-change temperature range (269°C – 292°C) and relatively high heat of fusion (379 kJ/kg).

Because the adopted composition is a near-eutectic, phase change occurs over a temperature interval rather than at a single temperature. For the simplified thermodynamic model, the phase transition was represented by an effective PCM temperature of 280 °C, which lies within the experimentally reported melting interval.

Useful latent thermal energy stored in the PCM was determined from the active mass of storage material and the adopted heat of fusion. Conversely, the required PCM mass was calculated from the useful thermal energy accumulated during the charging period. The amount of stored energy was obtained from the effective thermal power transferred from the exhaust gases to the PCM and the prescribed charging duration.

After accounting for the adopted charging effectiveness, effective charging power corresponds to the fraction of the temperature-constrained recoverable exhaust heat that is transferred to the storage material. Therefore, the latent-energy balance determines the active PCM mass required to store the recovered energy, but it does not determine the rate at which heat can be transferred between the exhaust gases and the PCM.

Thermal feasibility of the charging process was assessed separately using the temperature-constrained heat-transfer formulation presented in Section 3.6. This distinction is important because a material may exhibit high latent heat and adequate storage capacity while still presenting insufficient charging or discharging power because of low thermal conductivity, limited heat-transfer area, or inadequate exchanger geometry.

According to the reported thermal-cycling tests, the LiCl–LiOH mixture exhibited only limited variation in melting temperature and was therefore considered thermally stable within the operating range relevant to the present model. Nevertheless, the material is hygroscopic, and practical implementation would require appropriate sealing, moisture control, corrosion-resistant containment, and compatibility assessment between the salt mixture and the heat-

exchanger materials. These aspects were not explicitly modeled and should be investigated experimentally in future studies.

It should also be noted that the calculated PCM mass represents only the active storage material. The total mass and volume of a practical thermal storage unit would additionally include the heat exchanger, containment vessel, insulation, piping, structural supports, and auxiliary components.

3.6 Heat Transfer Modeling for the PCM System

Heat transfer between the exhaust gases and the thermal storage medium was represented using a lumped-parameter formulation. Rather than explicitly resolving heat-exchanger geometry, local temperature distributions, or the transient phase-change front, the model estimates the thermal power available for PCM charging from the exhaust-gas temperature level and an assumed heat-transfer effectiveness.

Accordingly, three distinct thermal quantities are considered in the present formulation: the total exhaust thermal power relative to the reference state, the thermally recoverable power established by the temperature constraint between the heat source and the storage medium, and the effective charging power obtained after applying the heat-transfer effectiveness.

After, the minimum allowable exhaust-gas outlet temperature was defined as:

$$T_{exh,out,min} = T_{PCM,eff} + \Delta T_{min,charge}, \quad (48)$$

where $T_{PCM,eff}$ is the effective phase-transition temperature adopted for the storage material and $\Delta T_{min,charge}$ is the minimum temperature approach in the exhaust-to-storage heat exchanger.

The thermally recoverable exhaust power was then estimated as:

$$Q_{exh,recorveable} = \dot{m}_{exh} \cdot c_{p_{exh,avg}} \cdot (T_{exh,in} - T_{exh,out,min}) \quad (49)$$

Where $c_{p_{exh,avg}}$ represents the average specific heat capacity of the exhaust-gas mixture over the considered temperature interval.

However, not all of this recoverable heat is effectively transferred to the PCM. To represent real-world limitations of the heat exchanger, we introduce an effectiveness factor. The actual heat stored in the PCM during charging is then:

$$Q_{charge} = \varepsilon_{Hx,charge} \cdot Q_{exh,recorveable} \quad (50)$$

The parameter $\varepsilon_{Hx,charge}$ represents how efficiently the heat exchanger transfers heat from the exhaust gases to the PCM. In this study, a value of 0.80 was assumed, meaning that 80% of the recoverable heat is effectively stored. This nominal overall charging effectiveness of 0.80 adopted in the baseline model. This parameter represents the fraction of the temperature-constrained recoverable exhaust heat that is effectively transferred to and stored in the PCM. It accounts, in a lumped manner, for unmodeled heat-transfer resistances, incomplete PCM utilization, and thermal losses. The adopted value lies within the range reported for enhanced PCM thermal storage systems, although the literature does not provide a universal effectiveness value applicable to all gas-to-PCM configurations. Consequently, the value of 0.80 was treated as a design assumption and evaluated through sensitivity analysis rather than as an experimentally validated property of the proposed heat exchanger.

The present formulation distinguishes three different thermal quantities: the total exhaust thermal power relative to the reference state, the thermally recoverable power determined by the source–storage temperature constraint, and the effective charging power after application of the heat-transfer effectiveness. The model does not explicitly resolve the heat-transfer area, overall heat-transfer coefficient, local PCM temperature distribution, or moving phase front. Therefore, the calculated charging power represents a nominal thermodynamic estimate that requires subsequent heat-exchanger design and experimental validation.

3.7 Rankine Cycle Model

The power output of the Rankine cycle was evaluated at three different levels. First, the thermodynamic net power of the cycle was calculated as the difference between the expander power and the pump consumption. This quantity represents the ideal mechanical power available from the cycle and is defined as $\dot{W}_{RC,net}$

Unlike conventional Rankine cycles employing surface condensers, the proposed configuration utilizes a direct-contact feedwater reservoir. The wet steam discharged from the screw expander enters the reservoir and condenses through direct interaction with the liquid inventory contained within the vessel. The resulting condensate is subsequently pressurized by the feedwater pump and recirculated to the PCM heater. This arrangement simplifies the system architecture by integrating condensation, condensate collection, and feedwater storage into a single component.

Subsequently, mechanical losses associated with bearings, seals, transmission components, and the screw expander mechanical efficiency were considered, resulting in the shaft mechanical power ($\dot{W}_{mech}$).

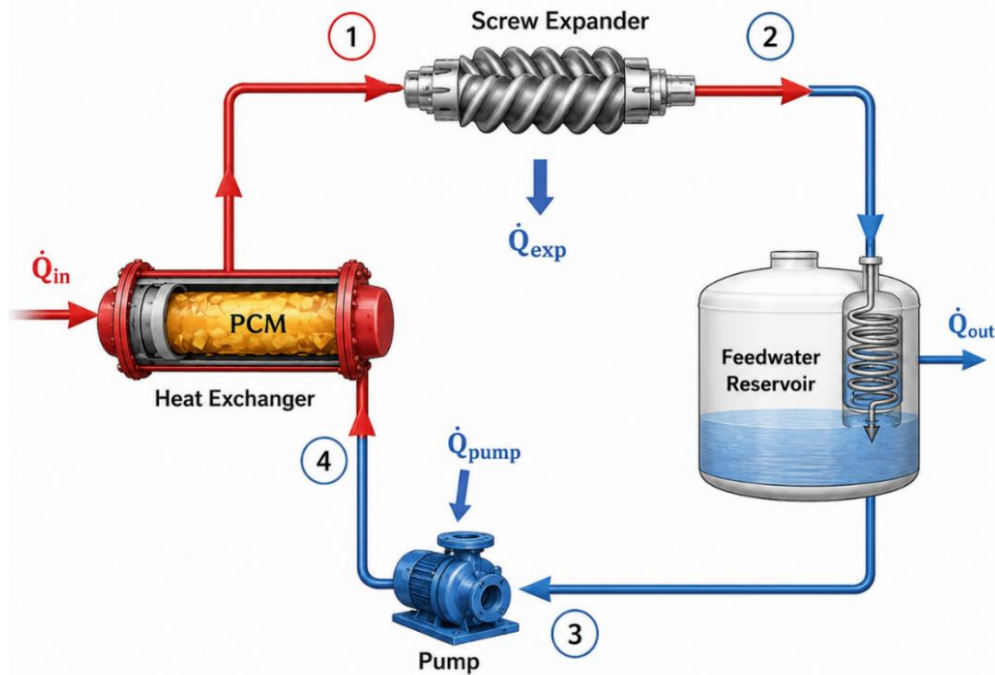
Finally, the electrical power delivered to the vehicle electrical system was obtained by applying the generator efficiency to the shaft power, resulting in the net electrical output ($\dot{W}_{elect}$).

This distinction allows the separation between thermodynamic cycle performance, mechanical conversion losses, and electrical conversion losses, providing a more realistic assessment of the practical power generation capability of the proposed CBCC architecture.

The bottoming cycle employed in the proposed architecture is a steam Rankine cycle integrated with a PCM-based thermal energy storage system. The primary objective of this subsystem is to recover a portion of the thermal energy available in the exhaust gases of the internal combustion engine and convert it into useful electrical power through a screw expander-generator assembly.

Figure 7 presents the schematic diagram of the proposed PCM-assisted Rankine cycle. The system receives thermal energy from the PCM thermal storage unit during the discharge process and converts part of this energy into useful mechanical and electrical power through the expansion of steam in a screw expander.

Figure 7 - PCM-assisted Rankine cycle with screw expander and direct-contact feedwater reservoir.



Source: Author's own elaboration.

The Rankine cycle was modeled under steady-state conditions, with variations in kinetic and potential energy neglected. The pump and screw expander were represented using isentropic efficiencies to account for the principal irreversibilities associated with the compression and expansion processes. The condensation process was assumed to occur in a direct-contact feedwater reservoir operating at the expander outlet pressure. This component simultaneously performs the functions of steam condensation, condensate collection, and feedwater storage upstream of the pump.

Unlike conventional waste heat recovery systems directly coupled to the exhaust gas stream, the proposed architecture operates in a temporally decoupled configuration. Thermal energy from the engine exhaust gases is initially accumulated in the PCM-based thermal storage system and subsequently released to the Rankine cycle during the discharge period. Consequently, the Rankine subsystem operates intermittently rather than continuously, allowing the thermal energy collected over a relatively long charging period to be delivered at a higher thermal power level over a shorter discharge period.

Water was selected as the Rankine-cycle working fluid because of its favorable thermodynamic properties, high latent heat of vaporization, low feed-pump power requirement, low cost, environmental compatibility, and suitability for medium- and high-temperature thermal energy storage applications. The selected PCM consists of a binary mixture containing 32 mol% LiCl and 68 mol% LiOH. This composition undergoes phase transition over an experimentally reported temperature interval of approximately 269–292 °C, with a latent heat of fusion of approximately 379 kJ/kg.

The phase-transition temperature range of the selected PCM is compatible with the thermal requirements of the Rankine subsystem, providing the temperature potential required for steam generation during the storage discharge process. The final expander inlet conditions, including the degree of steam superheating, are not prescribed at this stage but are determined through the optimization procedure described in the subsequent sections.

Because the selected mixture exhibits a melting interval rather than a single melting temperature, the thermal storage model represents its phase-transition behavior using an effective phase-change temperature or an equivalent enthalpy-based approximation. This simplification allows the latent storage capacity to be incorporated into the thermodynamic model while acknowledging that the actual charging and discharging processes occur progressively between the solidus and liquidus temperatures.

The integrated TES–Rankine subsystem was designed considering a fixed nominal PCM discharge time of 300 s. The thermal power supplied to the Rankine cycle was therefore

determined from the useful thermal energy stored in the PCM and the prescribed discharge duration. Accordingly, neither the Rankine-cycle net power output nor the steam mass flow rate was imposed as a design target. Both quantities were calculated by the thermodynamic model according to the thermal power available during the discharge process.

The principal design assumptions and operating parameters adopted for the integrated TES–Rankine subsystem are summarized in Table 1.

Table 1 - Rankine Cycle - Design Assumptions

Parameter	Symbol	Adopted value or treatment
Expander inlet pressure	P_1	Optimization variable
Expander outlet pressure	P_2	Optimization variable
Degree of steam superheating	$\Delta T_{superheat}$	Optimization variable
Expander isentropic efficiency	η_{exp}	0.75
Pump isentropic efficiency	η_{pump}	0.65
Expander mechanical efficiency	$\eta_{mech,exp}$	0.95
Generator efficiency	η_{gen}	0.95
Effective PCM transition temperature	$T_{PCM,eff}$	280 °C
Minimum charging temperature approach	$\Delta T_{min,charge}$	20 K
Charging heat-transfer effectiveness	$\epsilon_{HX,charge}$	0.80
PCM charging time	t_{charge}	1800 s
Nominal PCM discharge time	$t_{discharge}$	300 s
Rankine-cycle working fluid	-	Water/steam
Expander inlet condition	-	Superheated steam
Expander outlet condition	-	Wet steam, calculated by the model

The expander inlet pressure, expander outlet pressure, and degree of superheating were treated as optimization variables. Therefore, no fixed expander inlet state was prescribed. Instead, the optimization procedure determined the combination of pressure and superheating

that maximized the thermodynamic performance of the Rankine subsystem while respecting the adopted pressure and temperature bounds.

The steam entering the screw expander was assumed to be in the superheated region. Its temperature was calculated as the saturation temperature corresponding to the optimized inlet pressure plus the optimized superheating increment ($\Delta T_{superheat}$), in order to supply superheated steam to the expander at a temperature slightly above saturation. During expansion, the working fluid was allowed to enter the two-phase region. Therefore, the vapor quality at the expander outlet was determined as a model result rather than imposed as a fixed design condition.

The thermal power released during the PCM discharge was obtained by dividing the total stored thermal energy by the chosen discharge time:

$$\dot{Q}_{discharge} = \frac{E_{PCM}}{t_{discharge}} \quad (51)$$

A shorter discharge period increases the instantaneous thermal power delivered to the Rankine cycle without increasing the total stored energy.

It is important to note that the physically achievable discharge power also depends on the rate of heat transfer between the PCM and the working fluid. This rate is influenced by factors such as the thermal conductivity of the PCM, the geometry of the heat exchanger, the available heat-transfer area, the phase-change behavior, and the minimum temperature difference between the PCM and the steam. Since these aspects were not modeled using a detailed heat-exchanger approach (e.g., based on overall heat transfer coefficient and heat exchanger area), the calculated discharge power should be interpreted as a nominal thermodynamic estimate that requires further validation through heat-exchanger design and experimental analysis.

To ensure a realistic representation of the proposed PCM-assisted Rankine cycle, pressure losses were considered in both the evaporator and condenser. The thermodynamic model also incorporates realistic pump and screw expander efficiencies, together with pressure drops associated with heat exchangers and connecting piping.

Unlike conventional Rankine cycle analyses based on fixed operating conditions, the present study employed a genetic algorithm to optimize the principal operating parameters of the PCM–Rankine subsystem. The optimization procedure was applied to determine the most suitable combination of expander inlet pressure, degree of superheating, and outlet vapor

quality capable of maximizing the thermodynamic performance of the cycle while maintaining feasible operating conditions for the screw expander.

The optimization variables and the respective search ranges adopted during the genetic algorithm optimization in this study are summarized in Table 2.

Table 2 - Rankine Cycle – Optimization variables

Parameter	Optimization Range
Expander inlet pressure (P_1)	8-12 bar
Expander outlet (P_2)	1.1 - 2.0 bar
Degree of superheating (ΔT_{sh})	0-20 °C

The steam entering the expander was assumed to be in the superheated region. Therefore, the inlet thermodynamic state was defined by the optimized pressure and the optimized degree of superheating above the saturation temperature. During expansion, the working fluid may enter the two-phase region, and the vapor quality at the expander outlet was calculated as part of the optimization procedure rather than being imposed as a predefined condition.

The use of superheated steam at the expander inlet improves thermodynamic performance and reduces excessive moisture formation during expansion. However, the optimization process intentionally allows the outlet state to reach the wet-steam region because screw expanders are capable of operating with moderate liquid fractions without severe efficiency penalties (SMITH *et al.* 2011).

The condenser pressure range was selected based on saturation temperatures close to 100 °C, as recommended by Smith et al. (2011) for steam screw expander systems. Operating near atmospheric pressure provides a practical compromise between maximizing expansion work and limiting excessive volumetric flow rates at the expander outlet.

The optimized operating conditions were subsequently used in the thermodynamic and exergy analyses presented in the following sections.

The power output of the expander is calculated as:

$$\dot{W}_{exp} = \dot{m}_{st}(h_1 - h_2) \quad (52)$$

The real expansion process is modeled using an isentropic efficiency:

$$\eta_{exp} = \frac{h_1 - h_2}{h_1 - h_{2,s}} = 0.75 \quad (53)$$

Where $h_{2,s}$ is the enthalpy at the end of an isentropic expansion. An expander efficiency of $\eta_{exp} = 0.75$ was adopted, consistent with values reported for screw expanders operating with wet steam (SMITH et al. 2011).

The pump pressurizes the liquid water withdrawn from the direct-contact feedwater reservoir and delivers it to the PCM heater. Since the reservoir operates at the expander outlet pressure, the pump inlet state was assumed to correspond to saturated or slightly subcooled liquid conditions. The pump work is given by:

$$\dot{W}_{pump} = \dot{m}_{st}(h_4 - h_3) \quad (54)$$

The pump efficiency is defined as:

$$\eta_{pump} = \frac{h_{4s} - h_3}{h_4 - h_3} = 0.65 \quad (55)$$

A pump efficiency of $\eta_{pump} = 0.65$ was assumed, based on typical values for small-scale Rankine systems (ÇENGEL; BOLES, 2019).

In the refined TES–Rankine formulation, the steam mass flow rate is determined from the thermal power available during the TES discharge period rather than from a prescribed Rankine-cycle power output. Therefore, the net power generated by the bottoming cycle becomes a result of the available discharge heat rate and the thermodynamic states of the working fluid.

The thermal power supplied to the Rankine-cycle evaporator during TES discharge is given by:

$$\dot{Q}_{in,RC} = \dot{Q}_{discharge} \quad (56)$$

The specific heat input required by the Rankine cycle is calculated as:

$$q_{in,RC} = h_1 - h_4 \quad (57)$$

Where h_1 is the specific enthalpy at the screw-expander inlet and h_4 is the specific enthalpy at the pump outlet. This enthalpy difference includes the sensible heating, evaporation,

and superheating processes occurring before expansion. The steam mass flow rate is then obtained from:

$$\dot{m}_{steam} = \frac{\dot{Q}_{in,RC}}{h_1 - h_4} \quad (58)$$

The power produced by the screw expander is calculated as:

$$\dot{W}_{exp} = \dot{m}_{steam}(h_1 - h_2) \quad (59)$$

Where h_2 is the specific enthalpy at the expander outlet after accounting for the adopted expander isentropic efficiency. The pump power consumption is calculated as:

$$\dot{W}_{pump} = \dot{m}_{steam}(h_4 - h_3) \quad (60)$$

Where h_3 and h_4 are the specific enthalpies at the pump inlet and outlet, respectively. The thermodynamic net power output of the Rankine cycle is therefore calculated as:

$$\dot{W}_{RC,net} = \dot{m}_{steam}[(h_1 - h_2) - (h_4 - h_3)] \quad (61)$$

The specific net work of the Rankine cycle is defined as:

$$w_{RC,net} = (h_1 - h_2) - (h_4 - h_3) \quad (62)$$

Therefore:

$$\dot{W}_{RC,net} = \dot{m}_{steam} \cdot w_{RC,net} \quad (63)$$

The thermodynamic net power does not include the mechanical losses of the screw expander or the electrical conversion losses of the generator. The useful mechanical shaft power is calculated as:

$$\dot{W}_{shaft,RC} = \eta_{mech,exp} \cdot \dot{W}_{RC,net} \quad (64)$$

The electrical power generated by the Rankine subsystem is then calculated as:

$$\dot{W}_{electric,RC} = \eta_{gen} \cdot \dot{W}_{shaft,RC} \quad (65)$$

Combining both efficiencies:

$$\dot{W}_{electric,RC} = \eta_{mech,exp} \cdot \eta_{gen} \cdot \dot{W}_{RC,net} \quad (66)$$

Accordingly, the effective electrical power output of the Rankine subsystem is determined by the thermal power released by the TES, the resulting steam mass flow rate, the specific net work of the cycle, and the mechanical and electrical conversion efficiencies. The nominal power target adopted during the preliminary design stage is therefore not imposed in the refined model.

The thermal input required by the Rankine cycle is determined from the steam mass flow rate and the enthalpy rise between the pump outlet and the screw-expander inlet:

$$\dot{Q}_{in,RC} = \dot{m}_{steam}(h_1 - h_4) \quad (67)$$

For the TES-assisted configuration, this thermal requirement must be supplied by the storage system during the discharge period. The nominal discharge power is calculated by dividing the thermal energy stored in the PCM by the selected discharge time:

$$\dot{Q}_{discharge} = \frac{E_{PCM}}{t_{discharge}} \quad (68)$$

The thermal energy accumulated during the charging period is calculated as:

$$E_{PCM} = \dot{Q}_{charge} \cdot t_{charge} \quad (69)$$

This relation can also be written as:

$$\dot{Q}_{discharge} = \dot{Q}_{charge} \frac{t_{charge}}{t_{discharge}} \quad (70)$$

This equation represents the temporal concentration of the thermal energy stored in the TES. When the discharge time is shorter than the charging time, the instantaneous thermal power supplied to the Rankine cycle becomes greater than the thermal charging rate. However, the total amount of stored energy does not increase. For the simplified TES–Rankine coupling adopted in the thermodynamic model:

$$\dot{Q}_{in,RC} = \dot{Q}_{discharge} \quad (71)$$

Nevertheless, this equality represents a nominal energy-based operating condition. In an actual system, the thermal power supplied to the Rankine cycle would also be constrained by the heat-transfer capability of the PCM-to-steam heat exchanger:

$$\dot{Q}_{in,RC} \leq U \cdot A_{discharge} \cdot \Delta T_{lm,discharge} \quad (72)$$

Where U is the overall heat-transfer coefficient, $A_{discharge}$ is the effective heat-transfer area, and $\Delta T_{lm,discharge}$ is the logarithmic mean temperature difference during discharge. The expander inlet temperature must also remain below the effective PCM temperature by a minimum temperature difference:

$$T_1 \leq T_{PCM,eff} - \Delta T_{min,discharge} \quad (73)$$

Where T_1 is the steam temperature at the expander inlet and $\Delta T_{min,discharge}$ is the minimum temperature difference required to transfer heat from the PCM to the water–steam stream.

Since the heat-exchanger geometry, overall heat-transfer coefficient, local PCM temperature distribution, and solidification front were not explicitly modeled, the calculated discharge power represents a nominal thermodynamic estimate. Therefore, the practical feasibility of the predicted thermal and electrical power levels should be confirmed by transient numerical simulations and future experimental investigations.

The RC efficiency is calculated by the ratio of net work delivered to required input heat. Therefore, the following equation gives us this important parameter, which shows the quality of the modeled cycle.

$$\eta_{RC} = \frac{\dot{W}_{RC,net}}{\dot{Q}_{in,RC}} \quad (74)$$

3.8 Integration with Exhaust Heat Recovery

The integration between the engine exhaust gases, the TES subsystem, and the Rankine cycle was evaluated through two different heat-recovery configurations: a directly coupled Rankine cycle and a TES-assisted Rankine cycle.

The total thermal power carried by the exhaust gases relative to the environmental reference condition is denoted by $\dot{Q}_{exh}$.

However, the total exhaust thermal power does not correspond entirely to heat that can be transferred to the PCM. The recoverable exhaust power is limited by the minimum exhaust-gas outlet temperature required to maintain a finite temperature difference between the exhaust gases and the PCM. The minimum allowable exhaust-gas outlet temperature is defined as:

$$T_{exh,out,min} = T_{PCM,eff} + \Delta T_{min,charge} \quad (75)$$

The thermally recoverable exhaust power is calculated as:

$$\dot{Q}_{exh,recoverable} = \dot{m}_{exh} \cdot c_{p_{exh,avg}} [T_{exh,in} - T_{exh,out,min}] \quad (76)$$

The effective thermal power transferred from the exhaust gases to the storage subsystem is then obtained by applying the charging heat-transfer effectiveness, according to equation 50.

3.8.1 Directly coupled Rankine-cycle

In the directly coupled configuration, the recoverable exhaust heat is transferred directly to the Rankine cycle while the engine is operating. Therefore:

$$\dot{Q}_{in,RC,direct} = \dot{Q}_{charge} \quad (77)$$

The corresponding steam mass flow rate is calculated as:

$$\dot{m}_{steam,direct} = \frac{\dot{Q}_{in,RC,direct}}{h_1 - h_4} \quad (78)$$

The thermodynamic net power of the directly coupled Rankine cycle is:

$$\dot{W}_{RC,net,direct} = \dot{m}_{steam,direct} \cdot W_{RC,net} \quad (79)$$

The corresponding electrical power is:

$$\dot{W}_{electric,RC,direct} = \eta_{mech,exp} \cdot \eta_{gen} \cdot \dot{W}_{RC,net,direct} \quad (80)$$

The directly coupled configuration operates while the engine is running, but its power output is constrained by the instantaneous recoverable exhaust heat rate. Consequently, this configuration generally produces a comparatively low continuous power output.

3.8.2 TES-assisted Rankine-cycle configuration

In the TES-assisted configuration, the recoverable exhaust heat is accumulated during the charging period:

$$E_{PCM} = \dot{Q}_{charge} \cdot t_{charge} \quad (81)$$

The stored energy is subsequently released over the selected discharge period:

$$\dot{Q}_{in,RC,TES} = \frac{E_{PCM}}{t_{discharge}} \quad (82)$$

This relation can also be written:

$$\dot{Q}_{in,RC,TES} = \dot{Q}_{charge} \cdot \frac{t_{charge}}{t_{discharge}} \quad (83)$$

The corresponding steam mass flow rate is:

$$\dot{m}_{steam,TES} = \frac{\dot{Q}_{in,RC,TES}}{h_1 - h_4} \quad (84)$$

The thermodynamic net power of the TES-assisted Rankine cycle is:

$$\dot{W}_{RC,net,TES} = \dot{m}_{steam,TES} \cdot W_{RC,net} \quad (85)$$

The corresponding electrical power is:

$$\dot{W}_{electric,RC,TES} = \eta_{mech,exp} \cdot \eta_{gen} \cdot \dot{W}_{RC,net,TES} \quad (86)$$

The nominal thermal-power amplification factor produced by temporal decoupling is defined as:

$$F_{temporal} = \frac{\dot{Q}_{in,RC,TES}}{\dot{Q}_{in,RC,direct}} \quad (87)$$

3.9 Global System Efficiency

The global efficiency of the integrated system must be evaluated using a consistent temporal basis. Since the internal combustion engine operates during the TES charging period and the Rankine subsystem subsequently operates during the discharge period, the instantaneous Rankine-cycle power cannot be added directly to the engine power without considering the duration of each operating stage.

The brake thermal efficiency of the engine is calculated as:

$$\eta_{engine} = \frac{\dot{W}_{engine}}{\dot{Q}_{fuel}} \quad (88)$$

where $\dot{W}_{engine}$ is the engine brake power and $\dot{Q}_{fuel}$ is the fuel energy input rate. The fuel energy supplied during the charging period is:

$$E_{fuel} = \dot{Q}_{fuel} \cdot t_{charge} \quad (89)$$

The useful mechanical energy produced by the engine during the same period is:

$$E_{engine} = \dot{W}_{engine} \cdot t_{charge} \quad (90)$$

The electrical energy generated by the TES-assisted Rankine cycle during the discharge period is:

$$E_{RC,electric} = \dot{W}_{electric,RC,TES} \cdot t_{discharge} \quad (91)$$

The global energy efficiency of the complete charge–discharge cycle is therefore calculated as:

$$\eta_{global,TES} = \frac{(E_{engine} + E_{RC,electric})}{E_{fuel}} \quad (92)$$

Substituting the corresponding energy terms:

$$\eta_{global,TES} = \frac{(\dot{W}_{engine} \cdot t_{charge} + \dot{W}_{electric,RC,TES} \cdot t_{discharge})}{\dot{Q}_{fuel}} \quad (93)$$

For the directly coupled Rankine cycle, the global efficiency is calculated as:

$$\eta_{global,direct} = \frac{(\dot{W}_{engine} + \dot{W}_{electric,RC,direct})}{\dot{Q}_{fuel}} \quad (94)$$

If the direct and TES-assisted configurations recover the same quantity of exhaust energy and no additional storage losses are considered, their cycle-averaged global efficiencies are theoretically equivalent. The principal difference lies in the power-delivery profile. The directly coupled configuration produces a lower continuous power output, whereas the TES-assisted configuration produces a higher intermittent power output.

The absolute increase in global efficiency relative to the engine-only configuration is calculated as:

$$\Delta\eta_{global,rel} = \left[\frac{(\eta_{global} - \eta_{engine})}{\eta_{engine}} \right] \cdot 100 \quad (95)$$

The distinction between absolute improvement and relative improvement must be maintained. For example, an increase from 39.1% to 40.5% corresponds to an absolute increase of approximately 1.4 percentage points, but to a relative improvement of approximately 3.6%.

The global efficiency reported for the TES-assisted system therefore represents the complete charge–discharge energy balance and not the apparent instantaneous efficiency observed during the high-power TES discharge period.

4. RESULTS AND DISCUSSION

This section presents and discusses the main results obtained from the thermodynamic modeling of the proposed CBCC applied to an ethanol-based range extender. The analysis is structured to directly address the research questions defined in the Introduction, focusing on three key aspects: (i) the thermodynamic and exergy-based performance of the system (RQ1),

(ii) the implications of temporal decoupling enabled by TES (RQ2), and (iii) the influence of PCM-based storage design on energy utilization and system behavior (RQ3).

The analysis begins with the evaluation of the ICE operating point selected as the thermal source for the proposed system. In the final configuration, the ICE was treated as a fixed boundary condition, while the optimization procedure was applied to the PCM–Rankine subsystem. The engine results provide the baseline operating parameters used in the thermodynamic modeling of the integrated CBCC system.

Following this, the energy and exergy distributions of the optimized engine are analyzed in order to quantify the magnitude and quality of the exhaust gas energy available for recovery. These results establish the theoretical limits and practical constraints associated with waste heat utilization and provide the necessary inputs for the design of the PCM-based thermal storage system.

Subsequently, the performance of the Rankine bottoming cycle supplied by the PCM-based thermal storage is evaluated, highlighting its contribution to additional power generation and its impact on overall system efficiency. Particular attention is given to the limitations of directly coupled configurations and the performance improvements enabled by TES.

Finally, the role of the PCM-based TES system is examined in terms of its capacity to enable temporal decoupling between heat generation and power conversion. A parametric analysis is conducted to evaluate the influence of key design parameters, such as thermal effectiveness and charging time, on energy utilization and system behavior. The results are interpreted considering the assumptions and limitations of the adopted model, ensuring consistency with the hypotheses formulated for the CBCC architecture.

4.1 Internal Combustion Engine Performance

The thermodynamic model of the ethanol-fueled range extender was developed based on the Toyota 2NR-FBE engine, a commercially available 1.5-L spark-ignition engine widely used in the Brazilian market. The engine was operated under steady-state conditions with a compression ratio of 14 and an intake pressure of 150 kPa.

Unlike the preliminary analyses performed during the development stage of this work, the internal combustion engine was not optimized in the final configuration of the proposed system. Instead, the engine was treated as a fixed thermal source supplying exhaust energy to the PCM-based thermal storage system. This approach is consistent with the primary objective

of the present study, which focuses on the optimization and performance evaluation of the PCM–Rankine subsystem integrated into the Carnot Battery Combined Cycle (CBCC).

The EES model combines combustion stoichiometry, mass and energy balances, polytropic compression and expansion, and a calculated mechanical efficiency. The complete engine code is provided in Appendix 1. Table 2 summarizes the operating parameters used throughout the remainder of the analysis. These values were subsequently used as boundary conditions for the PCM charging process and for the thermodynamic analysis of the Rankine cycle subsystem.

Table 3 - Summary of engine performance parameters used as boundary conditions for the PCM–Rankine subsystem.

Parameter	Value
Engine Model	Toyota 2NR - FBE
Displacement	1.5 L
Compression Ratio	14
Intake Pressure (P_{intake})	150 kPa
Fuel mass flow rate ($\dot{m}_{fuel}$)	0.0066 kg/s
Fuel exergy rate ($\dot{E}x_{fuel}$)	184.4 kW
Fuel energy rate ($\dot{Q}_{fuel}$)	177.3 kW
Net Power Rate ($\dot{W}_{net}$)	69.38 kW
Mechanical efficiency (η_{mec})	0.84
Exhaust mass flow rate ($\dot{m}_{out}$)	0.0805 kg/s
Exhaust energy rate ($\dot{Q}_{exh}$)	47.92 kW
Exhaust exergy rate ($\dot{E}x_{exh}$)	27.56 kW

The engine produced 69.38 kW of brake power from a fuel energy input of 177.3 kW, corresponding to a brake thermal efficiency of 39.13%. The exhaust stream carried 47.92 kW of sensible thermal power relative to the adopted reference state and 27.56 kW of physical exergy. These values establish the thermal and exergetic limits for the PCM-Rankine subsystem.

The 47.92 kW exhaust value is not equivalent to the heat that can be transferred to the PCM. Recoverability also depends on the exhaust temperature, the PCM phase-transition temperature, and the minimum temperature difference required across the charging heat exchanger. This distinction is quantified in Sections 4.2 and 4.3.

The exhaust thermal power represents the total thermal resource carried by the exhaust gases relative to the reference state. However, only part of this quantity can be transferred to the PCM because the charging process is limited by the exhaust-gas temperature, the PCM phase-transition temperature, and the minimum temperature difference required in the heat exchanger. The thermally recoverable fraction is evaluated in Section 4.3.

4.2 Energy and Exergy Availability of Exhaust Gases

Exhaust heat is one of the main energy-loss pathways in spark-ignition engines and a relevant source for waste heat recovery. At the selected operating point, the exhaust thermal power was 47.92 kW, equivalent to 27.03% of the fuel energy input. This fraction lies within the range commonly reported for spark-ignition engines and depends on engine load, speed, mixture composition, and compression ratio (HEYWOOD, 1988; STONE, 1992; IRAN; CHEM; NOUROZI, 2021).

Table 4 - Energy distribution

Energy flow	Value (kW)	Fraction (%)
Fuel Energy	177.3	100
Brake Power	69.38	39.13
Exhaust energy	47.92	27.03
Cooling + losses	60.0	33.84

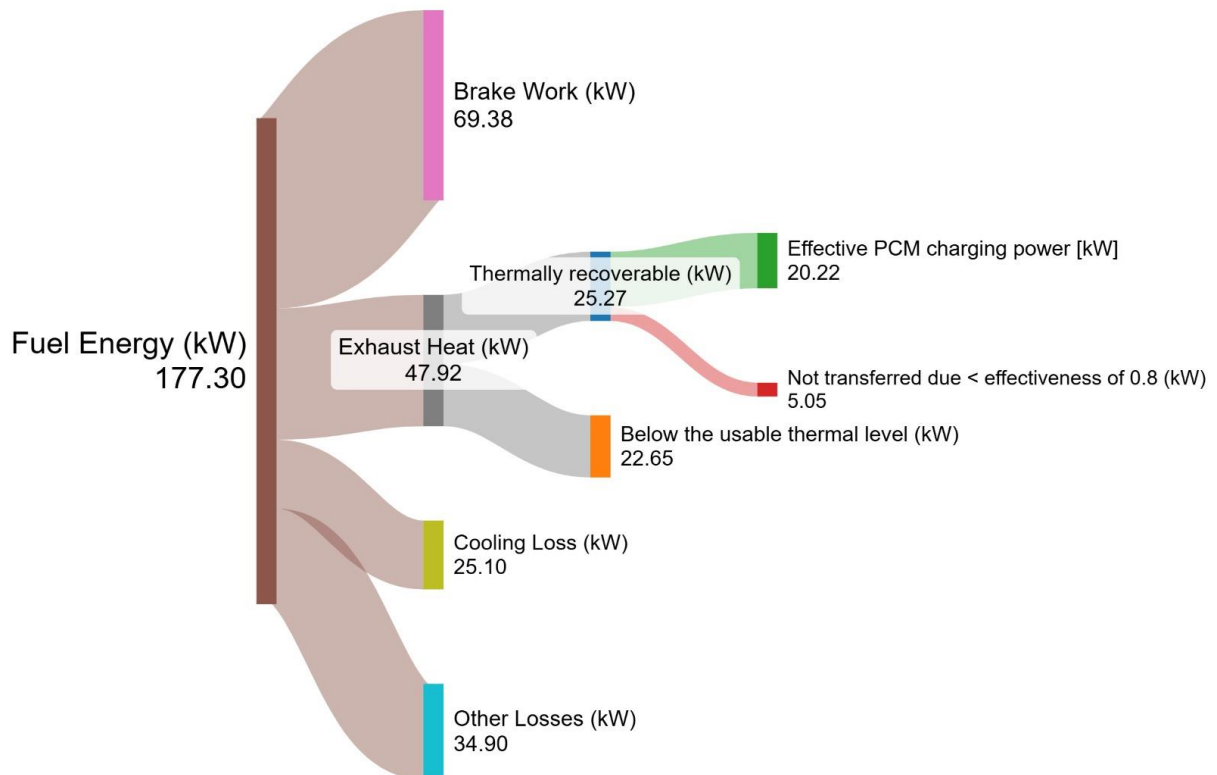
The exhaust thermal power was evaluated between the exhaust-gas inlet temperature of approximately 573 °C and an outlet reference temperature of 90 °C, adopted as the minimum practical exhaust-gas temperature for heat recovery to maintain an adequate temperature driving force and avoid operational limitations commonly considered in waste heat recovery systems (Shu et al., 2013). This value describes the total sensible heat above the reference state, but it cannot be used directly as the charging input because heat transfer to a PCM at approximately 280 °C requires a finite temperature difference.

With a minimum charging approach of 20°C, the exhaust outlet temperature was constrained to 300 °C. Cooling the exhaust only from 573 °C to this limit yields 25.27 kW of thermally recoverable power. Applying the charging heat-transfer effectiveness of 0.80 reduces the effective PCM input to 20.22 kW. Figure 8 summarizes the engine energy distribution and shows how the exhaust stream is divided into thermally recoverable heat, heat below the usable temperature level, effective PCM charging, and heat-transfer losses.

The exhaust physical exergy was 27.56 kW, giving an exergy-to-energy ratio of 57.5%. The obtained result is consistent with values reported in the literature, where the exergy-to-energy ratio of internal combustion engine exhaust gases typically ranges between 45% and 60%, depending on engine operating conditions, fuel type, and exhaust temperature (POZZATO; RIZZO; ONORI, 2021; SANDU; MAZILU, 2019).

This ratio indicates the theoretical work potential of the exhaust stream rather than the electrical power that can be recovered in practice. Finite-temperature heat transfer, incomplete exhaust cooling, pressure losses, thermal storage, expansion, and electromechanical conversion reduce the useful output below this upper limit (DINCER; ROSEN, 2013)

Figure 8 - Energy distribution in the engine modeling.



Source: Author's own elaboration using EES simulation.

4.3 Heat Recovery Performance and PCM Effectiveness

The refined heat-recovery model separates total exhaust thermal power from the fraction that is compatible with the PCM temperature level. For the selected PCM transition temperature of 280 °C and a 20°C minimum charging approach, the exhaust outlet temperature cannot fall below 300 °C. The resulting thermally recoverable power is 25.27 kW (as detailed in Figure 7), substantially lower than the total exhaust thermal power of 47.92 kW.

A nominal charging heat-transfer effectiveness of 0.80 (Zhang et al. (2020)) was applied to this recoverable fraction, resulting in 20.22 kW transferred to the PCM. Of the total exhaust thermal power, 22.65 kW remains below the selected usable temperature level, while approximately 5.05 kW of the recoverable fraction is not transferred because of the assumed heat-exchanger effectiveness.

The effectiveness value is a system-level modeling assumption rather than an experimentally validated characteristic of the proposed exchanger. It represents convection, conduction, contact resistance, and phase-change limitations in a single parameter. Its influence is examined in Section 4.10, while detailed determination of the exchanger area, overall heat-transfer coefficient, and transient phase front is left for future work.

4.4 PCM Thermal Storage Sizing

The PCM storage unit accumulates the effective exhaust heat during engine operation and supplies the Rankine cycle during a subsequent discharge period. At the baseline charging power of 20.22 kW and a charging duration of 1800 s, the stored thermal energy is 36.39 MJ. Assuming storage predominantly through latent heat and a fusion enthalpy of 379 kJ/kg, the required PCM mass is 96.02 kg. Table 5 summarizes the main parameters adopted for the PCM thermal storage sizing.

Table 5 - PCM modeling results

Parameter	Value
Total exhaust thermal power	47.92 kW
Thermally recoverable exhaust power	25.27 kW
Effective PCM charging power	20.22 kW
Latent heat of fusion	379 kJ/kg
Charging time	1800 s

Parameter	Value
Stored thermal energy	36,518 kJ
Required PCM mass	96.02 kg

The calculated mass is a thermodynamic estimate for the active storage material. A practical vehicular system would also include the heat exchanger, containment, insulation, structural supports, and allowances for volume change, corrosion, and incomplete phase transformation. Sensible energy in the PCM and containment was not included in the baseline sizing.

The storage unit does not create additional exhaust energy. Its function is to shift the recovered heat in time, allowing a low continuous charging rate to be released at a higher rate during Rankine-cycle operation. The feasibility of this power concentration depends on the discharge-side heat-transfer capability.

4.5 Genetic Optimization of the PCM–Rankine Subsystem

A genetic algorithm implemented in EES was used to identify the Rankine-cycle operating point that provides the best performance within the selected equipment and thermal constraints. The objective function combines electrical output, electrical conversion efficiency, and expander outlet vapor quality, with penalties for insufficient outlet quality or violation of the minimum temperature difference between the PCM and the steam stream.

The charging and discharge durations were fixed at 1800 s and 300 s, respectively. The optimization variables were the expander inlet pressure, expander outlet pressure, and degree of superheating, as listed in Table 6.

Table 6 - Optimization variables

Parameter	Lower Bound	Upper Bound
Expander inlet pressure (P_1)	800 kPa	1200 kPa
Expander outlet pressure (P_2)	110 kPa	200 kPa
Degree of superheating (ΔT_{sh})	0 °C	20 °C

The upper limit adopted for the expander inlet pressure (1200 kPa) was established based on commercially available screw machinery. According to technical specifications provided by DENAIR screw compressor systems, operating pressures around 12 bar represent

a practical limit for standard industrial screw machines. Superheating was limited to 20 °C to preserve an adequate temperature difference between the 280 °C PCM and the steam stream, reduce the sensible heat demand, and remain consistent with the simplified discharge heat-transfer model. The optimal operating conditions obtained from three independent optimization runs are summarized in Table 7.

Table 7 - Optimal operation parameters

Parameter	Optimized Value (1)	Optimized Value (2)	Optimized Value (3)
Expander inlet pressure (P_1)	1200 kPa	1200 kPa	1200 kPa
Expander outlet pressure (P_2)	110.1 kPa	110.1 kPa	110.6 kPa
Degree of superheating (ΔT_{sh})	19.6 °C	20.0 °C	19.1 °C
Objective function	0.7734	0.7736	0.7725

The solution converged to the upper bounds for inlet pressure and superheating and to the lower bound for outlet pressure. Within the prescribed search space, these conditions maximize the available expansion work while maintaining a positive discharge temperature approach of approximately 72 K. The optimum is therefore the best solution within the imposed practical limits, not an unconstrained thermodynamic maximum.

The outlet vapor quality of 0.9306 corresponds to a liquid mass fraction of about 6.9%. This wet-steam condition is consistent with the positive-displacement screw-expander concept adopted in the model, although machine-specific leakage, built-in volume ratio, rotor clearances, and liquid tolerance would require experimental verification (SMITH et al. 2011). It is important to note that the optimal solution consistently converged to the upper pressure limit of 1200 kPa. This behavior suggests that additional performance improvements could potentially be achieved at higher operating pressures. However, the pressure range was deliberately restricted to ensure compatibility with commercially available screw machinery. Consequently, the obtained optimum should be interpreted as the best solution within the practical operating limits imposed by currently available industrial equipment.

The optimized operating conditions obtained in this section are subsequently employed in the detailed thermodynamic, exergy, and global efficiency analyses presented in the following sections.

4.6 Performance of the Optimized Rankine Cycle

The Rankine cycle constitutes the power-conversion stage of the proposed Carnot Battery Combined Cycle, converting the thermal energy stored in the PCM into mechanical and electrical power. Unlike a conventional waste heat recovery configuration directly coupled to the engine exhaust, the TES-assisted Rankine cycle operates during a separate discharge period and is therefore not directly constrained by the instantaneous exhaust heat rate.

The stored energy of 36.39 MJ is released over 300 s, providing a nominal thermal input of 121.30 kW to the Rankine cycle. Because the mass flow rate is determined from this available heat and the optimized cycle states, the resulting power is heat-limited rather than imposed as a target.

The steam mass flow rate is determined from the thermal power supplied during discharge and from the specific energy required to heat, evaporate, and condition the working fluid. Based on the optimized thermodynamic states obtained from the EES model, the expander power, pump consumption, net cycle power, mechanical shaft power, electrical output, Rankine-cycle efficiency, and expander outlet vapor quality are evaluated.

The optimized operating condition should be interpreted as a heat-limited result rather than a prescribed power target. In other words, the net power of the Rankine cycle is obtained from the thermal energy effectively stored and subsequently released by the PCM, together with the optimized specific performance of the cycle.

Table 8 presents a compilation of the main thermodynamic and performance results obtained after the optimization procedure.

Table 8 - Summary of optimized Rankine cycle performance

Parameter	Value
Expander inlet enthalpy (h_1)	2836 kJ/kg
Expander outlet enthalpy (h_2)	2523 kJ/kg
Pump inlet enthalpy (h_3)	428.9 kJ/kg
Pump outlet enthalpy (h_4)	430.9 kJ/kg
RC mass flow rate ($\dot{m}_{st}$)	0.0504 kg/s

Parameter	Value
Expander outlet quality (x_2)	0.930
Expander power ($\dot{W}_{exp}$)	15.78 kW
Pump power ($\dot{W}_{pump}$)	0.09 kW
Thermodynamic net RC power ($\dot{W}_{net,RC}$)	15.69 kW
Thermal power required in the evaporator ($\dot{Q}_{in,R}$)	121.30 kW
Screw expander efficiency (η_{exp})	0.75
Pump efficiency (η_{pump})	0.65
Screw expander mechanical efficiency ($\eta_{mec,exp}$)	0.95
Electrical generator efficiency ($\eta_{electric}$)	0.95
Mechanical shaft power ($\dot{W}_{real,exp}$)	14.9 kW
Electrical generator output ($\dot{W}_{electric}$)	14.1 kW
RC efficiency (η_{RC})	12.93 %

4.7 Exergy Efficiency and Energy Quality

The exergy analysis provides a measure of the thermodynamic quality of the exhaust heat and establishes the theoretical upper limit for useful work production. The engine model predicted a total exhaust thermal power of 47.92 kW and a physical exhaust exergy rate of 27.56 kW. The ratio between these quantities is approximately 57.5%, indicating that slightly more than half of the exhaust thermal energy retains theoretical useful-work potential under ideal reversible conditions.

However, the total exhaust thermal power should not be interpreted as the heat entirely available for PCM charging. After accounting for the effective PCM transition temperature and the minimum temperature difference required during charging, the thermally recoverable exhaust power was limited to approximately 25.27 kW. Applying the charging heat-transfer effectiveness of 0.80 reduced the effective charging power to approximately 20.22 kW.

Therefore, the thermal energy available in the exhaust gases can be interpreted in three successive stages. First, the exhaust gases carry a total thermal power of 47.92 kW relative to the adopted reference temperature. Second, only 25.27 kW are thermodynamically recoverable, corresponding to the fraction of exhaust-gas thermal energy available above the effective PCM phase-change temperature, while satisfying the minimum temperature approach imposed in the charging heat exchanger. Finally, after accounting for the assumed heat-transfer effectiveness, approximately 20.22 kW are effectively transferred to the PCM and stored as latent heat during the charging process.

The PCM does not increase the amount of exhaust exergy. Its contribution is to redistribute the recovered thermal energy over time, enabling the Rankine cycle to operate intermittently at a higher instantaneous power level.

The exergy efficiency of the PCM–Rankine subsystem must be evaluated using a common time basis. It is not appropriate to compare the instantaneous electrical power generated during the 300 s discharge period directly with the exhaust exergy rate available during the 1800 s charging period.

Over the complete charging period, the exhaust stream provides approximately 49.61 MJ of physical exergy. The electrical energy generated by the Rankine subsystem must be calculated from the revised electrical output multiplied by the 300 s discharge duration. The cycle-based exergy efficiency is then obtained by comparing this electrical energy with the exhaust exergy supplied during charging.

Over the complete charge–discharge sequence, the exhaust stream supplied approximately 49.61 MJ of physical exergy, while the Rankine subsystem generated approximately 4.25 MJ of electrical energy. Accordingly, the cycle-based exergy efficiency of the PCM–Rankine subsystem was approximately 8.56%. This value represents the fraction of the exhaust physical exergy converted into electrical energy over a consistent temporal basis.

A more detailed analysis could evaluate the exergy effectively transferred to the PCM and the exergy destruction during charging and discharge. However, this would require a more detailed representation of the heat exchangers and of the temperature evolution of the storage material. Within the simplified model adopted in this study, the energy-based comparison over the complete charge–discharge cycle provides a more consistent system-level indicator.

4.8 Global System Efficiency

Global system efficiency was evaluated over the complete charge–discharge sequence because the internal combustion engine and the Rankine subsystem operate during different time intervals. While the engine operates throughout the 1800 s thermal-storage charging period, the Rankine cycle operates subsequently during the 300 s PCM discharge period. A consistent comparison must therefore be based on the total useful energy produced by both subsystems relative to the fuel energy supplied during engine operation.

At the selected operating condition, the ethanol-fueled engine produces 69.38 kW of brake power from a fuel-energy input rate of 177.30 kW, resulting in a brake thermal efficiency of 39.13%. During the charging period, the engine simultaneously generates useful mechanical energy and supplies exhaust heat to the PCM storage system. Once charging is completed, part of the stored thermal energy is converted into mechanical power by the Rankine subsystem during the subsequent discharge period.

For consistency, the global efficiency was calculated on a mechanical-energy basis. Accordingly, the brake energy produced by the engine during the 1800 s charging period was added to the mechanical shaft energy delivered by the Rankine cycle during the 300 s discharge period. This combined useful energy was then divided by the total fuel energy supplied to the engine during the charging interval.

Using the calculated engine and Rankine-cycle outputs, the integrated CBCC architecture achieved a global mechanical efficiency of approximately 40.53%. Compared with the engine-only brake thermal efficiency of 39.13%, this result represents an absolute increase of approximately 1.40 percentage points and a relative improvement of approximately 3.58%.

An equivalent interpretation may be obtained by distributing the intermittent Rankine-cycle shaft output over the complete 1800 s charging interval. Since the Rankine subsystem operates for 300 s, its cycle-averaged mechanical contribution corresponds to the discharge shaft power multiplied by the discharge-to-charge time ratio. This treatment prevents the relatively high instantaneous Rankine output from being interpreted as a continuous contribution throughout the entire engine operating period.

Distinguishing instantaneous power from cycle-averaged power is essential when assessing the integrated system. Although the PCM-assisted Rankine cycle delivers approximately 14.91 kW of shaft power during discharge, this output is available for only one-sixth of the engine operating duration. Consequently, the improvement in global efficiency

remains modest even though the instantaneous bottoming-cycle power is substantially higher than that of the directly coupled configuration.

When additional thermal-storage and discharge losses are neglected, the directly coupled and TES-assisted configurations recover approximately the same amount of exhaust energy over a complete operating sequence. Their cycle-averaged efficiencies are therefore expected to remain similar when the same thermodynamic states and conversion efficiencies are adopted. The principal distinction lies in the temporal delivery of the recovered energy: the directly coupled Rankine cycle provides a lower continuous output while the engine is operating, whereas the TES-assisted configuration provides a higher intermittent output after the charging stage.

4.9 Temporal decoupling (Carnot Battery)

Temporal decoupling between exhaust-heat recovery and power generation represents the main functional contribution of the PCM-based Carnot Battery. Under the steady-state engine condition considered in this study, the exhaust stream provides an approximately constant thermal source. Rather than compensating for short-term exhaust fluctuations, the storage subsystem accumulates the thermally recoverable exhaust energy over an extended period and subsequently releases it over a shorter interval.

In the directly coupled configuration, the Rankine cycle receives approximately 20.22 kW of thermal power while the engine is operating. Under the optimized thermodynamic conditions, this thermal input produces a steam mass flow rate of approximately 0.00819 kg/s. As a result, the directly coupled cycle generates approximately 2.615 kW of thermodynamic net power, 2.484 kW of mechanical shaft power, and 2.36 kW of electrical output after generator losses.

With thermal storage, the same effective charging power is supplied to the PCM over a period of 1800 s, resulting in approximately 36.39 MJ of stored thermal energy. Releasing this energy over a fixed discharge period of 300 s increases the nominal thermal input to the Rankine cycle to approximately 121.30 kW.

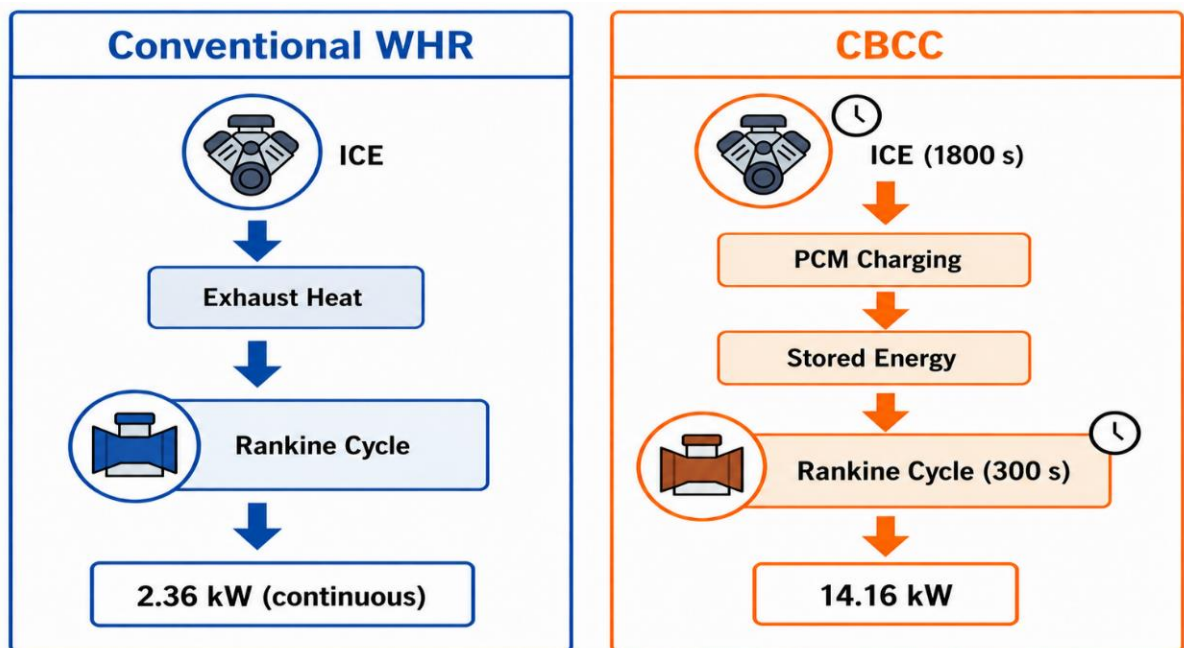
At this higher discharge power, the optimized Rankine cycle operates with a steam mass flow rate of approximately 0.05043 kg/s and produces 15.69 kW of thermodynamic net power. After accounting for the mechanical efficiency of the screw-expander assembly, the available shaft power reaches approximately 14.91 kW. Application of the generator efficiency results in an electrical output of approximately 14.16 kW.

Comparison of the two configurations indicates a thermal- and power-concentration factor of approximately six. Relative to direct coupling, the TES-assisted system supplies nearly six times more thermal power to the Rankine cycle and consequently produces approximately six times more thermodynamic, mechanical, and electrical power. This increased output is available, however, for only one-sixth of the charging duration.

Despite the substantial difference in instantaneous power, the total thermal energy supplied to the Rankine cycle remains approximately unchanged when additional storage and discharge losses are neglected. Direct coupling provides approximately 20.22 kW over 1800 s, whereas the TES-assisted configuration provides approximately 121.30 kW over 300 s. Both cases therefore correspond to approximately 36.39 MJ of recovered thermal energy.

Because both configurations were evaluated using the same optimized thermodynamic states, the Rankine-cycle efficiency remains approximately 12.93%. After mechanical and generator losses, the effective electrical conversion efficiency is approximately 11.67%. Thus, the higher instantaneous electrical output of the TES-assisted configuration results primarily from the increased thermal input during discharge rather than from an improvement in the intrinsic efficiency of the Rankine cycle.

Figure 9 - Comparison between conventional direct waste heat recovery and PCM-assisted temporal decoupling in the proposed Carnot Battery Combined Cycle architecture.



Source: Author's own elaboration.

Beyond power concentration, temporal decoupling provides additional operational flexibility by allowing the engine and the bottoming cycle to operate sequentially. Such an arrangement reduces the direct dependence of the Rankine subsystem on real-time engine operation and may facilitate operation closer to a selected design point. Nevertheless, the practical realization of this benefit depends on the ability of the thermal storage unit and heat exchanger to deliver the calculated discharge power within the prescribed 300 s interval.

4.10 Influence of PCM Design Parameters

The storage response is governed by the PCM transition temperature, latent heat, heat-transfer effectiveness, charging duration, and discharge duration. These parameters affect different aspects of the system and should not be evaluated in isolation.

A lower transition temperature increases the exhaust temperature interval available for heat recovery, but it also lowers the temperature level supplied to the Rankine cycle and may restrict steam superheating. Conversely, a higher transition temperature improves heat quality while reducing the recoverable fraction. The selected value of 280 °C represents a compromise between these effects.

Table 9 shows the effect of charging effectiveness for a fixed 1800 s charging period and 300 s discharge period. The reported PCM mass is the mass required to store the full latent-energy amount in each scenario; if the installed mass were fixed, lower effectiveness would instead produce a lower state of charge.

Table 9 - Influence of charging effectiveness on PCM storage performance

Charging effectiveness	Charging power	Stored energy	Required PCM mass	Thermal input during 300s discharge
0.4	10.11 kW	18.19 MJ	48.01 kg	60.65 kW
0.5	12.64 kW	22.74 MJ	60.01 kg	75.81 kW
0.6	15.16 kW	27.29 MJ	72.01 kg	90.97 kW
0.7	17.69 kW	31.84 MJ	84.01 kg	106.13 kW
0.8	20.22 kW	36.39 MJ	96.02 kg	121.30 kW

The results are nearly linear because the model applies a constant effectiveness to a fixed thermally recoverable heat rate. Doubling effectiveness from 0.40 to 0.80 doubles the charging power, stored energy, latent-storage requirement, and nominal discharge power.

Table 10 evaluates charging duration at the baseline effectiveness of 0.80. The PCM mass again represents the capacity required to absorb all available heat. For a fixed 96.02 kg

storage unit, charging beyond 1800 s would require sensible storage, partial heat rejection, or a larger PCM inventory once complete melting is reached.

Table 10 - Influence of charging time on PCM storage performance

Charging effectiveness	Stored energy	Required PCM mass	Thermal input during 300s discharge
900 s	18.19 MJ	48.01 kg	60.65 kW
1200 s	24.26 MJ	64.01 kg	80.86 kW
1800 s	36.39 MJ	96.02 kg	121.30 kW
2400 s	48.52 MJ	128.02 kg	161.73 kW
3000 s	60.65 MJ	160.02 kg	202.16 kW
3600 s	72.78 MJ	192.03 kg	242.59 kW

Discharge duration changes power but not stored energy. Shorter discharge periods increase Rankine-cycle input and output, but also require greater heat-transfer area and higher internal PCM heat flux. Latent heat mainly determines the required active-material mass; it does not, by itself, guarantee a high charging or discharge rate.

4.11 Practical Implications for Heavy-Duty Electric Trucks

The proposed CBCC architecture was also evaluated from the perspective of its potential application as an onboard range extender for heavy-duty battery-electric trucks. Current commercial electric trucks commonly employ battery capacities between approximately 400 and 600 kWh. Therefore, a nominal battery capacity of 500 kWh was adopted as a representative case for the present practical assessment.

For example, the Volvo FH Electric is available with installed battery capacities between 360 and 540 kWh, while the Mercedes-Benz eActros 600 employs a total installed capacity of 621 kWh. Therefore, the selected capacity represents a realistic order of magnitude for long-haul and regional heavy-duty electric-truck applications (VOLVO TRUCKS, 2022; DAIMLER TRUCK, 2024).

During each CBCC operating cycle, the internal combustion engine produces 69.38 kW of brake power for 1800 s, while the PCM is charged by the recoverable exhaust heat. After the charging phase, the Rankine subsystem operates for 300 s and delivers approximately 14.91 kW of mechanical shaft power or 14.16 kW of electrical power.

On a mechanical-energy basis, the ICE produces approximately 34.69 kWh during the 1800 s operating period, whereas the Rankine subsystem contributes approximately 1.24 kWh during its 300 s discharge period. Therefore, each complete 1800 s operating sequence provides

approximately 35.93 kWh, corresponding to an average mechanical power of approximately 61.6 kW.

Under these idealized conditions, approximately 8.1 h would be required to provide an amount of energy equivalent to the full nominal capacity of a 500 kWh battery pack. However, this value does not include the electrical conversion losses of the ICE generator or the charging losses of the battery.

Assuming a generator efficiency of 95% for the ICE and using the calculated Rankine-cycle electrical output of 14.16 kW, the average electrical power supplied by the CBCC is approximately 58.5 kW. This results in a theoretical charging time of approximately 8.5 h. When an additional battery-charging efficiency of approximately 95% is considered, the estimated time increases to approximately 9.0 h.

These values represent the time required to supply 500 kWh under stationary conditions or in the absence of traction demand. During vehicle operation, the battery state of charge will depend on the balance between the electrical power produced by the CBCC and the instantaneous power consumed by the traction system and auxiliary loads. Consequently, the proposed system should be interpreted primarily as a range extender capable of reducing the battery discharge rate rather than as a high-power fast-charging system.

Vehicle mass is another relevant consideration. Battery mass represents one of the principal practical constraints associated with heavy-duty vehicle electrification. NREL vehicle-technology scenarios consider pack-level battery masses ranging from approximately 2.50 to 4.70 kg/kWh, depending on the assumed technological development level. For a nominal 500 kWh battery, this range corresponds to approximately 1.25–2.35 t. An intermediate assumption of 4.03 kg/kWh results in a battery-pack mass of approximately 2.02 t. These values include the pack-level contribution rather than only the electrochemical cells and are therefore more appropriate for a preliminary vehicle-level mass assessment (BROOKER et al., 2020).

A conventional heavy-duty diesel tractor has a representative mass of approximately 7.4 t. Removing the diesel engine, transmission, and driveline and replacing them with an electric driveline results in an estimated battery-electric tractor mass of approximately 5.85 t before the battery pack is installed.

For a 500 kWh battery, the pack mass may vary from approximately 2.0 to 3.6 t, depending on the adopted pack-level energy density. An intermediate value of approximately 2.5 t results in a battery-electric tractor mass of approximately 8.35 t, or about 0.95 t more than the representative diesel configuration.

The mass of the proposed CBCC range extender was estimated from its principal components, including the 1.5-L ethanol engine, generator, exhaust heat exchanger, PCM, containment and insulation, Rankine-cycle equipment, condenser, pumps, piping, cooling system, controls, and fuel storage. Since the mechanical layout has not yet been developed in detail, a preliminary range of approximately 0.7–1.2 t was obtained, with a central estimate close to 0.9 t.

Using this intermediate estimate, the complete battery-electric truck equipped with a 500 kWh battery and the proposed CBCC would have a tractor mass of approximately 9.25 t. This corresponds to an increase of approximately 1.85 t relative to the representative diesel tractor.

The additional mass must be considered together with the operational benefits of the range extender. The CBCC may reduce dependence on external charging infrastructure, extend vehicle operation in remote routes, and enable the use of a smaller battery in applications where the range extender is available for prolonged operation. In practice, the most advantageous configuration may therefore involve reducing the battery capacity rather than adding the CBCC to a full-size 500 kWh battery pack.

For example, replacing part of the battery capacity with the range extender could partially offset the mass of the CBCC. At a representative pack-level specific energy of 0.20 kWh/kg, each 100 kWh reduction in battery capacity would reduce vehicle mass by approximately 500 kg. Consequently, reducing the battery from 500 to 300 kWh could save approximately 1.0 t, which would be comparable to the estimated mass of the complete CBCC system.

This result suggests that the practical value of the proposed architecture lies not only in recharging a large onboard battery, but also in enabling battery downsizing while maintaining operational range and flexibility. Nevertheless, a definitive assessment requires detailed mechanical design, component selection, vehicle packaging analysis, fuel-storage sizing, and evaluation of the resulting payload capacity.

5 CONCLUSIONS

This dissertation presented a thermodynamic and exergy-based assessment of a Carnot Battery Combined Cycle (CBCC) architecture for an ethanol-fueled range extender intended for heavy-duty electrified transport. The proposed configuration integrates an Otto-cycle internal combustion engine, a phase change material (PCM)-based thermal energy storage

system, and a wet-steam Rankine cycle equipped with a screw expander. In the final model, the engine was treated as a fixed thermal source, while the operating conditions of the PCM–Rankine subsystem were optimized within practical equipment and thermal constraints.

The engine model predicted a brake power output of 69.38 kW from a fuel-energy input of 177.30 kW, corresponding to a brake thermal efficiency of 39.13%. The exhaust gases carried 47.92 kW of thermal power and 27.56 kW of physical exergy, resulting in an exergy-to-energy ratio of approximately 57.5%. These results confirm that the exhaust stream contains a relevant amount of recoverable energy, while the exergy value establishes the theoretical limit for useful-work production. After accounting for the PCM transition temperature and the minimum charging temperature approach, the thermally recoverable exhaust power was limited to approximately 25.27 kW. With a charging effectiveness of 0.80, the effective thermal power transferred to the PCM was approximately 20.22 kW.

For the baseline storage design, the PCM charging and discharge times were fixed at 1800 s and 300 s, respectively. Under these conditions, approximately 36.39 MJ of thermal energy were stored. Considering a latent heat of fusion of 379 kJ/kg, the required active PCM mass was estimated at 96.02 kg. This value represents only the phase-change material and does not include the additional mass and volume associated with the heat exchanger, containment, insulation, structural supports, piping, and auxiliary equipment.

The genetic optimization was applied to the expander inlet pressure, expander outlet pressure, and degree of superheating. Within the prescribed search space, the optimum converged to an inlet pressure of approximately 1200 kPa, an outlet pressure of approximately 110.1 kPa, and a superheating degree close to 20 °C. The expander outlet vapor quality was approximately 0.9306, which is compatible with the wet-steam operating capability assumed for the screw expander. Because the solution reached the upper bounds for inlet pressure and superheating and the lower bound for outlet pressure, the result should be interpreted as the optimum within the adopted practical constraints rather than as an unconstrained thermodynamic maximum.

During the 300 s discharge period, the stored energy supplied a nominal thermal input of 121.30 kW to the Rankine cycle. The resulting steam mass flow rate was approximately 0.0504 kg/s. The optimized cycle produced 15.69 kW of thermodynamic net power, approximately 14.9 kW of mechanical shaft power, and approximately 14.16 kW of electrical output after generator losses. The Rankine-cycle thermal efficiency was approximately 12.93%, while the effective electrical conversion efficiency was approximately 11.67%. These outputs

were determined from the thermal energy available during discharge and were not imposed as nominal power targets.

Regarding RQ1, the CBCC increased the useful energy obtained from the fuel by converting part of the exhaust heat into electrical energy. When the complete charge–discharge sequence is evaluated on a consistent time basis, the global efficiency increases from 39.13% for the engine alone to approximately 40.53% for the integrated system. This corresponds to an absolute improvement of approximately 1.4 % and a relative improvement of approximately 3.6%. The improvement results from waste-heat conversion rather than from any increase in the exergy originally contained in the exhaust gases.

Regarding RQ2, the principal contribution of the PCM-based Carnot Battery is temporal decoupling and power concentration. In the directly coupled configuration, the Rankine cycle receives approximately 20.22 kW of thermal power and produces approximately 2.615 kW of thermodynamic net power and 2.36 kW of electrical power while the engine is operating. In the TES-assisted configuration, the same amount of recovered thermal energy is accumulated over 1800 s and released over 300 s, increasing the instantaneous thermal and electrical power levels by a factor of approximately six. However, the TES-assisted subsystem operates for only one-sixth of the charging duration. Therefore, neglecting additional storage and discharge losses, the total recovered energy remains essentially unchanged; the principal benefit is the ability to deliver it at a higher, controllable, and dispatchable power level.

Regarding RQ3, the results confirmed that the operational behavior of the CBCC is strongly influenced by PCM and heat-transfer design parameters. Charging effectiveness and charging duration determine the amount of energy stored and the required PCM capacity, whereas discharge duration determines the instantaneous thermal power supplied to the Rankine cycle. The PCM transition temperature establishes a compromise between the amount of exhaust heat that can be recovered and the temperature level available for steam generation and superheating. Latent heat mainly influences the required active-material mass, but it does not, by itself, guarantee adequate charging or discharge power. Consequently, storage-material selection and heat-exchanger design must be evaluated together.

Within the assumptions of the developed model, the two hypotheses proposed in this dissertation were supported. The results support H1 by showing that thermal storage improves the practical utilization and dispatchability of exhaust heat and increases the total useful output of the range-extender system, while the theoretical exhaust exergy remains determined by the exhaust thermodynamic state. The results also support H2 by demonstrating the sensitivity of

the CBCC to storage capacity, charging effectiveness, charging duration, discharge duration, and heat-transfer limitations.

From a vehicle-level perspective, the proposed CBCC should be interpreted primarily as a range-extending and battery-discharge-rate-reduction system rather than as an onboard fast-charging device. The estimated charging time for delivering an amount of electrical energy equivalent to a 500 kWh battery capacity is several hours under idealized stationary conditions. Moreover, the additional mass of the CBCC must be considered in the vehicle design. Its practical advantage may therefore be greater when the range extender enables a reduction in installed battery capacity, partially offsetting the mass of the engine, thermal storage, Rankine-cycle equipment, and associated auxiliaries.

The main limitations of this study arise from the steady-state thermodynamic formulation and the simplified treatment of thermal storage. The model does not explicitly resolve transient PCM melting and solidification, the spatial temperature distribution within the storage unit, the overall heat-transfer coefficient, the required heat-transfer area, exhaust-side pressure losses, thermal losses during storage, or the detailed off-design performance of the screw expander. Vehicle dynamics, supervisory control, packaging, durability, corrosion, safety, cost, and experimental validation were also outside the scope of the present work.

Future research should therefore develop a transient PCM and heat-exchanger model based on detailed geometry and UA or effectiveness–NTU relationships, including pressure drop, thermal losses, incomplete phase change, and cycling degradation. Experimental studies should validate the charging and discharge rates and determine whether the nominal 121.30 kW thermal transfer during the 300 s discharge period is physically achievable. Further work should also evaluate expander off-design behavior, integrated control strategies, vehicle duty cycles, battery downsizing, component packaging, mass, cost, and lifecycle environmental performance. Overall, the proposed CBCC does not increase the theoretical energy or exergy available in the engine exhaust; instead, it provides a means of storing, concentrating, and dispatching part of that resource, thereby improving operational flexibility and modestly increasing the cycle-averaged efficiency of ethanol-based range extenders.

APPENDICES

APPENDIX 1

```
1:
2: Subprogram air_reaction (x_hc; y_hc; z_hc; beta_st; phi: OF_act; Y_O2; YP_CO2; YP_H2O; YP_O2; YP_N2; OFV_stoi;
   a_st; R_gp)
3: $UnitSystem SI mass J K Pa
4: "CxHyOz + a_st*O2 + beta_st*N2 ----> b_st*CO2 + d_st*H2O + e_st*N2 + f_st*O2"
5:
6: b_st = x_hc
7: 2*d_st = y_hc
8: 2*a_st + z_hc = 2*b_st + d_st
9: 2*e_st = a_st*beta_st
10:
11: m_oxy = a_st*1 [kmol]*molarmass(O2)
12: m_N2 = a_st*beta_st*1 [kmol]*molarmass(N2)
13: Y_O2 = m_oxy/(m_oxy+m_N2)
14:
15: MW_oxi = 1/(Y_O2/(molarmass(O2)) + (1-Y_O2)/(molarmass(N2)))
16: R_xyz = R#/molarmass(CH4)
17:
18: R_oxi = R#/MW_oxi
19: P = 101325 [Pa]
20: T = 298,15 [K]
21: P = rho_oxi*R_oxi*T
22: P = rho_xyz*R_xyz*T
23:
24:
25: m_CHO_st = x_hc*1 [kmol]*molarmass(C) + y_hc*1 [kmol]*molarmass(H) + z_hc*1 [kmol]*molarmass(O)
26: m_oxy_st = m_oxy + m_N2
27: OFV_stoi = m_oxy_st/rho_oxi/(m_CHO_st/rho_xyz)
28:
29: OF_st = m_oxy_st/m_CHO_st
30: phi = OF_st/OF_act
31: OF_act = m_oxy_act/m_CHO_st
32: m_oxy_act = a_act*(1 [kmol]*molarmass(O2) + 1 [kmol]*beta_st*molarmass(N2))
33:
34: m_prod = m_oxy_act + m_CHO_st
35:
36: YP_CO2 = b_act*1 [kmol]*molarmass(CO2)/m_prod
37: YP_H2O = d_act*1 [kmol]*molarmass(H2O)/m_prod
38: YP_N2 = e_act*1 [kmol]*molarmass(N2)/m_prod
39: YP_O2 = 1 - (YP_CO2 + YP_H2O + YP_N2)
40: b_act = b_st
41: d_act = d_st
42: e_act = a_act*beta_st
43:
44:
45: MW_prod = 1/(YP_O2/molarmass(O2) + YP_N2/molarmass(N2) + YP_CO2/molarmass(CO2) + YP_H2O/molarmass(H2O))
46:
47: R_gp = R#/MW_prod
48:
49: End
50:
51: Subprogram Rankine_PCM_CBCC (P_RC1; P_RC2; DeltaT_superheat; m_dot_exh; T_exh_in; Q_dot_exh_total;
   Ex_dot_exh; W_dot_engine; Q_dot_fuel; Q_dot_charge; E_PCM_stored; m_PCM; Q_dot_discharge; m_dot_RC; x_RC2;
   W_dot_net_RC; W_dot_electric_RC; W_dot_electric_RC_direct; eta_RC_electric; eta_global_TES; eta_global_direct;
   eta_exergy_cycle; DeltaT_available_discharge; F_obj)
52: $UnitSystem SI C kPa kJ mass
53:
54: {=====}
55: { RANKINE CYCLE ONLY - WET STEAM / SCREW EXPANDER }
56: { Unified PCM-CBCC subprogram linked to ethanol ICE }
57: {=====}
```

```

58:
59: {-----}
60: { OPTIMIZATION VARIABLES }
61: {-----}
62:
63: "Set bounds in Variable Info: 800 to 1200 kPa"
64:
65: "Set bounds in Variable Info: 110 to 200 kPa"
66:
67: "Set bounds in Variable Info: 0 to 20 C"
68:
69: {-----}
70: { FIXED DESIGN PARAMETERS }
71: {-----}
72:
73: t_charge = 1800 [s]
74: t_discharge = 300 [s]
75:
76: T_PCM_eff = 280 [C]
77:
78: DeltaT_min_charge = 20 [K]
79: DeltaT_min_discharge = 20 [K]
80:
81: epsilon_HX_charge = 0,80
82:
83: h_fusion_PCM = 379 [kJ/kg]
84:
85: eta_pump = 0,65
86: eta_exp = 0,75
87: eta_mech_exp = 0,95
88: eta_gen = 0,95
89:
90: DeltaP_evap = 100 [kPa]
91:
92: P_RC3 = P_RC2
93: P_RC4 = P_RC1 + DeltaP_evap
94:
95: x_RC3 = 0
96:
97: {-----}
98: { ENGINE EXHAUST INPUTS }
99: {-----}
100:
101: cp_exh_avg = 1,15 [kJ/kg-K]
102:
103:
104:
105: {-----}
106: { TEMPERATURE-CONSTRAINED }
107: { EXHAUST HEAT RECOVERY }
108: {-----}
109:
110: T_exh_out_min = T_PCM_eff + DeltaT_min_charge
111:
112: Q_dot_exh_recoverable = m_dot_exh*cp_exh_avg*(T_exh_in - T_exh_out_min)
113:
114: Q_dot_charge = epsilon_HX_charge*Q_dot_exh_recoverable
115:
116: E_PCM_stored = Q_dot_charge*t_charge
117:

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```

118: m_PCM = E_PCM_stored/h_fusion_PCM
119:
120: {-----}
121: { PCM DISCHARGE }
122: {-----}
123:
124: Q_dot_discharge = E_PCM_stored/t_discharge
125:
126: Q_dot_in = Q_dot_discharge
127:
128: {-----}
129: { STATE 3 - PUMP INLET }
130: {-----}
131:
132: h_RC3 = enthalpy(Steam; P=P_RC3; x=x_RC3)
133: s_RC3 = entropy(Steam; P=P_RC3; x=x_RC3)
134: v_RC3 = volume(Steam; P=P_RC3; x=x_RC3)
135: T_RC3 = temperature(Steam; P=P_RC3; x=x_RC3)
136:
137: {-----}
138: { STATE 4 - PUMP OUTLET }
139: {-----}
140:
141: h_RC4s = h_RC3 + v_RC3*(P_RC4 - P_RC3)
142:
143: h_RC4 = h_RC3 + (h_RC4s - h_RC3)/eta_pump
144:
145: s_RC4 = entropy(Steam; P=P_RC4; h=h_RC4)
146: T_RC4 = temperature(Steam; P=P_RC4; h=h_RC4)
147:
148: {-----}
149: { STATE 1 - EXPANDER INLET }
150: {-----}
151:
152: T_sat_RC1 = temperature(Steam; P=P_RC1; x=1)
153:
154: T_RC1 = T_sat_RC1 + DeltaT_superheat
155:
156: h_RC1 = enthalpy(Steam; P=P_RC1; T=T_RC1)
157: s_RC1 = entropy(Steam; P=P_RC1; T=T_RC1)
158:
159: {-----}
160: { STATE 2 - EXPANDER OUTLET }
161: {-----}
162:
163: h_RC2s = enthalpy(Steam; P=P_RC2; s=s_RC1)
164:
165: h_RC2 = h_RC1 - eta_exp*(h_RC1 - h_RC2s)
166:
167: s_RC2 = entropy(Steam; P=P_RC2; h=h_RC2)
168: T_RC2 = temperature(Steam; P=P_RC2; h=h_RC2)
169: x_RC2 = quality(Steam; P=P_RC2; h=h_RC2)
170:
171: {-----}
172: { SPECIFIC CYCLE VALUES }
173: {-----}
174:
175: w_exp = h_RC1 - h_RC2
176:
177: w_pump = h_RC4 - h_RC3

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178:
179:  $w_{net\_RC} = w_{exp} - w_{pump}$ 
180:
181:  $q_{in\_RC} = h_{RC1} - h_{RC4}$ 
182:
183:  $q_{out\_RC} = h_{RC2} - h_{RC3}$ 
184:
185:  $\eta_{RC} = w_{net\_RC} / q_{in\_RC}$ 
186:
187: {-----}
188: { STEAM MASS FLOW RATE }
189: {-----}
190:
191:  $m_{dot\_RC} = Q_{dot\_in} / q_{in\_RC}$ 
192:
193: {-----}
194: { RANKINE POWER OUTPUTS }
195: {-----}
196:
197:  $W_{dot\_exp} = m_{dot\_RC} * w_{exp}$ 
198:
199:  $W_{dot\_pump} = m_{dot\_RC} * w_{pump}$ 
200:
201:  $W_{dot\_net\_RC} = W_{dot\_exp} - W_{dot\_pump}$ 
202:
203:  $W_{dot\_shaft\_RC} = \eta_{mech\_exp} * W_{dot\_net\_RC}$ 
204:
205:  $W_{dot\_electric\_RC} = \eta_{gen} * W_{dot\_shaft\_RC}$ 
206:
207:  $\eta_{RC\_electric} = W_{dot\_electric\_RC} / Q_{dot\_in}$ 
208:
209:  $Q_{dot\_out} = m_{dot\_RC} * q_{out\_RC}$ 
210:
211:  $m_{water} = m_{dot\_RC} * t_{discharge}$ 
212:
213: {-----}
214: { THERMAL COMPATIBILITY }
215: {-----}
216:
217:  $\Delta T_{available\_discharge} = T_{PCM\_eff} - T_{RC1}$ 
218:
219: {Required condition:
220:  $\Delta T_{available\_discharge} \geq \Delta T_{min\_discharge}$ }
221:
222:  $T_{RC1\_max} = T_{PCM\_eff} - \Delta T_{min\_discharge}$ 
223:
224: {-----}
225: { DIRECT RANKINE MODE }
226: {-----}
227:
228:  $Q_{dot\_in\_direct} = Q_{dot\_charge}$ 
229:
230:  $m_{dot\_RC\_direct} = Q_{dot\_in\_direct} / q_{in\_RC}$ 
231:
232:  $W_{dot\_exp\_direct} = m_{dot\_RC\_direct} * w_{exp}$ 
233:
234:  $W_{dot\_pump\_direct} = m_{dot\_RC\_direct} * w_{pump}$ 
235:
236:  $W_{dot\_net\_RC\_direct} = W_{dot\_exp\_direct} - W_{dot\_pump\_direct}$ 
237:

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238: W_dot_shaft_RC_direct = eta_mech_exp*W_dot_net_RC_direct
239:
240: W_dot_electric_RC_direct = eta_gen*W_dot_shaft_RC_direct
241:
242: eta_RC_direct = W_dot_net_RC_direct/Q_dot_in_direct
243:
244: eta_RC_direct_electric = W_dot_electric_RC_direct/Q_dot_in_direct
245:
246: {-----}
247: { GLOBAL SYSTEM EFFICIENCY }
248: {-----}
249:
250:
251:
252: eta_engine = W_dot_engine/Q_dot_fuel
253:
254: {TES Rankine average electrical power over charging period}
255:
256: W_dot_electric_RC_avg = W_dot_electric_RC*t_discharge/t_charge
257:
258: eta_global_TES = (W_dot_engine + W_dot_electric_RC_avg)/Q_dot_fuel
259:
260: eta_global_direct = (W_dot_engine + W_dot_electric_RC_direct)/Q_dot_fuel
261:
262: Delta_eta_global_abs = eta_global_TES - eta_engine
263:
264: Delta_eta_global_rel = Delta_eta_global_abs/eta_engine
265:
266: {-----}
267: { ENERGY-BASED EXERGY }
268: {-----}
269:
270: E_exh_exergy = Ex_dot_exh*t_charge
271:
272: E_RC_electric = W_dot_electric_RC*t_discharge
273:
274: eta_exergy_cycle = E_RC_electric/E_exh_exergy
275:
276: {-----}
277: { TEMPORAL DECOUPLING }
278: {-----}
279:
280: F_temporal = Q_dot_discharge/Q_dot_charge
281:
282: Energy_check_charge = Q_dot_charge*t_charge
283:
284: Energy_check_discharge = Q_dot_discharge*t_discharge
285:
286: Energy_balance_error = Energy_check_charge - Energy_check_discharge
287:
288: {-----}
289: { OUTPUTS }
290: {-----}
291:
292: eta_RC_percent = eta_RC*100
293:
294: eta_RC_electric_percent = eta_RC_electric*100
295:
296: eta_engine_percent = eta_engine*100
297:

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298: eta_global_TES_percent = eta_global_TES*100
299:
300: eta_global_direct_percent = eta_global_direct*100
301:
302: Delta_eta_global_abs_percent = Delta_eta_global_abs*100
303:
304: Delta_eta_global_rel_percent = Delta_eta_global_rel*100
305:
306: eta_exergy_cycle_percent = eta_exergy_cycle*100
307:
308: {-----}
309: { GENETIC OPTIMIZATION }
310: {-----}
311:
312: W_ref = 20 [kW]
313: eta_ref = 0,15
314: x_ref = 1
315:
316: Penalty_x = IF(x_RC2; 0,85; 100*(0,85 - x_RC2); 0; 0)
317:
318: Penalty_T = IF(DeltaT_available_discharge; DeltaT_min_discharge; 100*(DeltaT_min_discharge -
DeltaT_available_discharge)/DeltaT_min_discharge; 0; 0)
319:
320: F_obj = 0,50*(W_dot_electric_RC/W_ref) + 0,30*(eta_RC_electric/eta_ref) + 0,20*(x_RC2/x_ref) - Penalty_x - Penalty_T
321:
322: End
323:
324: $UnitSystem SI mass J K Pa
325:
326: "
_____
"

327:
328: beta_st = 3,76
329:
330:
331: m_in = m_f + m_oxi                                     "mass of fuel"
332: OF_act = m_oxi/m_f
333: phi = OF_st/OF_act
334:
335: SFC = m_dot_fuel*convert(kg/s ; g/hr)/W_dot_net
336:
337:
338: Q_in = m_f*HC
339: eta=W_net/Q_in                                         "efficiency"
340: Q_dot_in = m_dot_fuel *HC
341: m_dot_fuel = m_f*N_cyl/(2*period)
342: m_dot_oxi = m_oxi*N_cyl/(2*period)
343: m_dot_out = m_dot_fuel + m_dot_oxi
344:
345: m_dot_fuel = m_dot_g + m_dot_d
346: Y_d = m_dot_d/m_dot_fuel
347: Y_g = m_dot_g/m_dot_fuel
348: Q_dot_LHV = (m_dot_fuel*HC)*convert(W; kW)
349:
350: Call air_reaction (x_hc; y_hc; z_hc; beta_st; phi; OF_act; Y_O2; YP_CO2; YP_H2O; YP_O2; YP_N2; OFV_stoi; a_st;
R_gp)
351:
352: "_____MOTOR DATA_____"
353: stroke$ = '4$'
354: period=1/RPS

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355: bore= 72,5 [mm]*convert(mm;m)
356: stroke= 90,6 [mm]*convert(mm;m)
357: CR= 14 [-]
358: N_cyl=4 [-]
359: RPS =4500 [1/min]*convert(1/min;1/s)
360: T_amb=converttemp(C;K;25 [C])
361: P_amb = 101 [kPa]*convert(kPa;Pa)
362: P_intake= 150 [kPa]*convert(kPa;Pa)
363: T_intake=converttemp(C;K;90 [C])
364: phi = 0,8
365: Y_d = 0,15
366: x_hc = 2
367: y_hc = 6
368: z_hc = 1
369: HC = lowerheatingvalue(C2H5OH)
370:
371:
372:
373:
374: " MOTOR DATA "
375: PTC_exit = 160 [kPa]*convert(kPa;Pa)
376: r_tc = P_intake*T_amb/(P_amb*T_intake)
377:
378: Vol_dis_cyl=pi*bore^2*stroke/4
379:
380: Vol_cl=Vol_dis_cyl/(CR-1)
381: Vol_BDC=Vol_dis_cyl+Vol_cl
382: Vol_TDC=Vol_cl
383: eta_gen = 0,95
384:
385:
386: "
"

387: " Martin Internal Combustion Engine Model "
388:
389:
390: T_0 = converttemp(C ; K; 25 [C])
391: P_0 = 1[atm]*convert(atm; Pa)
392: rho_0 = density(Air;T=T_0;P=P_0)
393: w = 2*stroke*RPS
394:
395:
396: xi=0,277 + 0,06*(1-phi + (1-0,1*CR^0,5)*(1-phi)^2,5)
397:
398: eta_ind + epsilon_P + epsilon_E + epsilon_R = 1
399: e1 = -0,2
400: e2 = 0,8
401: e3 = -0,4
402:
403: cf_1 = 1,8
404: cf_2 = 1,2
405:
406: r0m0 = 0,14*cf_1*(density(Air;T=T_amb;P=P_amb)/viscosity(Air;T=T_amb))^(-0,2)
407: epsilon_P = r0m0*(1[s/m^2]*r_tc*w*bore)^e1*(CR^e2 + 3*CR^e3)
408:
409: epsilon_P = Q_exp/Q_in
410:
411: epsilon_E = 1/(CR^xi)-epsilon_P*(2*cf_2/(CR^xi + 1))
412:
413: epsilon_R = 0,01

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414: eta_e = eta_mec*eta_ind
415: eta_mec = 1 - p_bar_f/p_bar_ind
416:
417: p_bar_ind = eta_ind*r_tc*rho_0*phi*HC/OF_st
418:
419: p_bar_f = p_bar_f0 + k_s*(2*p_bar_c + p_bar_ind) + k_w*w/bore
420:
421: p_bar_c = P_intake*(CR^xi-1)/xi
422:
423: k_s = 0,02 [-]
424: k_w = 0,7 [kPa-s]*convert(kPa-s ; Pa-s)
425: p_bar_f0 = 70 [kPa]*convert(kPa; Pa)
426:
427: "
      "
428:
429: "Klein and Nellis Model "
430: MW_prod = YP_O2*molarmass(O2) + YP_N2*molarmass(N2) + YP_CO2*molarmass(CO2) + YP_H2O*molarmass(H2O)
431: R_prod = R#/MW_prod
432:
433:
434: hf_CO2 = enthalpy_formation(CO2)
435: hf_H2O = enthalpy_formation(H2O)
436:
437: " State 1 is fixed by the temperature and pressure. "
438: (T[1]=700 [K]) "guess for initial temperature"
439: T[1]=T[6] "initial gas temperature must match final gas temperature"
440: P[1]=P_intake "pressure"
441:
442: Vol[1]=Vol_TDC "volume"
443: m[1]=Vol[1]/v[1] "mass"
444: u[1] = intenergy(Air;T=T[1])
445: s[1] = entropy(Air;T=T[1];P=P[1])
446: v[1] = volume(Air;T=T[1];P=P[1])
447: rho[1] = 1/v[1]
448: h[1] = enthalpy(Air;T=T[1])
449: TC[1] = converttemp (K; C ; T[1])
450:
451: "State 2"
452: Vol[2]=Vol_BDC "volume"
453: P[2]=P_intake "pressure"
454: h_in= enthalpy(Air;T=T_intake)
455: m_in=m[2]-m[1] "mass balance"
456: m_in*h_in = P_intake*(Vol[2] - Vol[1]) + m[2]*u[2] - m[1]*u[1] "energy balance"
457: v[2] = Vol[2]/m[2]
458: rho[2] = 1/v[2]
459: u[2] = intenergy(Air;T=T[2])
460: s[2] =entropy(Air;P=P[2];v=v[2])
461: T[2] = temperature(Air;P=P[2];v=v[2])
462: h[2] = enthalpy(Air;T=T[2])
463: TC[2] = converttemp (K; C ; T[2])
464:
465: "State 3"
466: n_comp= 0,95*cp(Air; T=T_amb)/cv(Air; T=T_amb) "n_comp = 1.35"
467: m[3]=m[2] "mass balance"
468: Vol[3]=Vol_TDC "volume"
469: v[3]=Vol[3]/m[3]
470: rho[3] = 1/v[3] "specific volume"

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471: P[3]*Vol[3]^(n_comp)=P[2]*Vol[2]^(n_comp)           "polytropic relationship"
472: T[3]= temperature(Air,P=P[3];v=v[3])               "temperatur
e"
473: s[3]=entropy(Air,P=P[3];v=v[3])                   "entropy"
474: u[3]= intenergy(Air,T=T[3])                       "internal
energy"
475: h[3] = enthalpy(Air,T=T[3])                       "enthalpy"
476: TC[3] = converttemp (K; C ; T[3])
477:
478: W_comp=-(P[3]*Vol[3]-P[2]*Vol[2])/(1-n_comp)       "work"
479: W_comp=Q_comp+m[3]*u[3]-m[2]*u[2]                 "energy balance"
480:
481: "State 4"
482:
483: m[4]=m[3]                                           "mass balance"
484: Vol[4]=Vol_TDC                                    "volume"
485: v[4]=Vol[4]/m[4]                                   "specific volume"
486: rho[4] = 1/v[4]
487: Q_in=m[4]*u[4]-m[3]*u[3]                          "energy balance"
488: u4_CO2 = YP_CO2*(intenergy(CO2;T=T[4]) - intenergy(CO2;T=T_amb))
489: u4_H2O = YP_H2O*(intenergy(H2O;T=T[4]) - intenergy(H2O;T=T_amb))
490: u[4] = YP_O2*intenergy(O2;T=T[4]) + YP_N2*intenergy(N2;T=T[4]) + u4_CO2 + u4_H2O
491: MW_4 = 1/(YP_O2/molarmass(O2) + YP_N2/molarmass(N2) + YP_CO2/molarmass(CO2) + YP_H2O/molarmass(H2O))
492: R_4 = R#/MW_4
493: P[4]*v[4] = R_4*T[4]                              "pressure"
494: s[4]= YP_O2*entropy(O2;T=T[4];P=P[4]) + YP_N2*entropy(N2;T=T[4];P=P[4]) + YP_CO2*entropy(CO2;T=T[4];P=P[4]) +
YP_H2O*entropy(H2O;T=T[4];P=P[4])                    "entropy"
495: h[4] = YP_O2*enthalpy(O2;T=T[4]) + YP_N2*enthalpy(N2;T=T[4]) + YP_CO2*(enthalpy(CO2;T=T[4]) - hf_CO2) + YP_H2O
*(enthalpy(H2O;T=T[4]) - hf_H2O)                    "enthalpy"
496: TC[4] = converttemp (K; C ; T[4])
497:
498: "State 5"
499: m[5]=m[4]                                           "mass balance"
500: Vol[5]=Vol_BDC                                    "volume"
501: v[5]=Vol[5]/m[5]                                   "specific volume"
502: rho[5] = 1/v[5]
503: P[5]*Vol[5]^(n_exp)=P[4]*Vol[4]^(n_exp)
504: R_5 = R_4
505: P[5]*v[5] = R_5*T[5]                              "temperature"
506:
507: s[5]= YP_O2*entropy(O2;T=T[5];P=P[5]) + YP_N2*entropy(N2;T=T[5];P=P[5]) + YP_CO2*entropy(CO2;T=T[5];P=P[5]) +
YP_H2O*entropy(H2O;T=T[5];P=P[5])
508:
509: u5_CO2 = YP_CO2*(intenergy(CO2;T=T[5]) - intenergy(CO2;T=T_amb))
510: u5_H2O = YP_H2O*(intenergy(H2O;T=T[5]) - intenergy(H2O;T=T_amb))
511: u[5]= YP_O2*intenergy(O2;T=T[5]) + YP_N2*intenergy(N2;T=T[5]) + u5_CO2 + u5_H2O
512: "internal energy"
513: h[5] = YP_O2*enthalpy(O2;T=T[5]) + YP_N2*enthalpy(N2;T=T[5]) + YP_CO2*(enthalpy(CO2;T=T[5]) - hf_CO2) + YP_H2O
*(enthalpy(H2O;T=T[5]) - hf_H2O)                    "enthalpy"
514: TC[5] = converttemp (K; C ; T[5])
515:
516: {n_exp = 1,25}
517: W_exp= (P[5]*Vol[5]-P[4]*Vol[4])/(1-n_exp)         "work"
518: 0=W_exp+Q_exp+m[5]*u[5]-m[4]*u[4]                 "energy balance"
519:
520: "State 6"
521: Vol[6]=Vol_BDC                                    "volume"
522: P[6]= P_intake                                    "pressure"
523: s[6]=s[5]                                           "entropy balance on gas remaining in cylinder"
524: s[6] = YP_O2*entropy(O2;T=T[6];P=P[6]) + YP_N2*entropy(N2;T=T[6];P=P[6]) + YP_CO2*entropy(CO2;T=T[6];P=P[6]) +

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YP_H2O*entropy(H2O;T=T[6];P=P[6]) "temperature"
525: TC[6] = converttemp (K; C ; T[6])
526:
527: u6_CO2 = YP_CO2*(intenergy(CO2;T=T[6]) - intenergy(CO2;T=T_amb))
528: u6_H2O = YP_H2O*(intenergy(H2O;T=T[6]) - intenergy(H2O;T=T_amb))
529: u[6]= YP_O2*intenergy(O2;T=T[6]) + YP_N2*intenergy(N2;T=T[6]) + u6_CO2 + u6_H2O
530:
531: h[6] = YP_O2*enthalpy(O2;T=T[6]) + YP_N2*enthalpy(N2;T=T[6]) + YP_CO2*(enthalpy(CO2;T=T[6]) - hf_CO2) + YP_H2O
*(enthalpy(H2O;T=T[6]) - hf_H2O) "enthalpy"
532: Cp_6 = YP_O2*cp(O2;T=T[6]) + YP_N2*cp(N2;T=T[6]) + YP_CO2*cp(CO2;T=T[6]) + YP_H2O*cp(H2O;T=T[6])
533:
534: R_6 = R_4
535: P[6]*v[6] = R_6*T[6]
536: v[6]=Vol[6]/m[6]
537: rho[6] = 1/v[6]
538: 0=m_out+m[6]-m[5] "mass balance"
539:
540: "Turbo charger SIMPLE analysis"
541: "compressor"
542: Q_c + W_c = h[c_s] - h[c_e]
543: Q_c = 0
544: h[c_e] = enthalpy(Air;T=T_amb)
545: s_se = entropy(Air;T=T_amb;P=P_amb)
546: h[c_si] = enthalpy(Air;s=s_se;P=P_intake)
547: eta_c = 0,65
548: eta_c = (h[c_si] - h[c_e])/(h[c_s] - h[c_e])
549:
550: p_bmep = p_bar_ind - p_bar_f
551: bmep_bar = p_bmep*convert(Pa;bar)
552:
553: "Turbine"
554: Q_t - W_t = h[t_s] - h[t_e]
555: Q_t = 0
556: W_t = W_c/eta_t
557: eta_t = 0,65
558: h[t_e] = enthalpy(Air;T=T[6])
559: T_7=temperature(Air;h=h[t_s])
560: TC_7 = converttemp(K; C; T_7) "exhaust gas temperature "C, goes to PCMH"
561: {Q_exh = (h[t_e] - enthalpy(Air;T= 363 [K]))*m_dot_out*convert(W; kW)}
562: Q_exh = (h[t_e] - enthalpy(Air;T=363 [K]))*m_dot_out*convert(W; kW)
563:
564:
565: deltaH_m = 1163 [J/g]*convert(J/g; J/kg)
566: T_exh=TC_7
567:
568: eta_e = W_dot_net/Q_dot_LHV
569:
570: Q_dot_exp = Q_exp*N_cyl/(2*period)*convert(W; kW)
571: Q_dot_comp = Q_comp*N_cyl/(2*period)*convert(W; kW)
572:
573: W_dot_net = W_net*N_cyl/(2*period)*convert(W; kW)
574:
575: W_dot_brake = eta_mec*W_dot_net
576: eta_brake = W_dot_brake/Q_dot_LHV
577:
578:
579:
580: Vol_dis=Vol_dis_cyl*N_cyl "total displacement of the engine"
581: Vol_dis_liter=Vol_dis*convert(m^3;liter) "in cubic inches"
582:

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583: "Cooling system (assumido ou modelado)"
584: m_dot_cool = 0,4 [kg/s]
585: cp_cool = 4180 [J/kg-K]
586: T_cool_in = converttemp(C; K; 80 [C]) "Coolant inlet temperature"
587: T_cool_out = converttemp(C; K; 95 [C]) "Coolant outlet temperature"
588:
589: Qdot_cool = m_dot_cool * cp_cool * (T_cool_out - T_cool_in)
590: Qdot_cool_kW = Qdot_cool * convert(W;kW)
591:
592: Qdot_fuel_kW = Q_dot_LHV * convert(W;kW)
593:
594: Qdot_unacc = Qdot_fuel_kW - (W_dot_brake + Q_exh + Qdot_cool_kW)
595:
596: "=====
597: "EXERGY ANALYSIS"
598: "=====
599:
600: "Reference environment"
601: T0_ex = T_amb
602: P0_ex = P_amb
603: P_exh = 101325 [kPa] "Exhaust gas pressure at WHR/PCM inlet"
604:
605: "-----"
606: "1) Physical exergy of exhaust gases"
607: "-----"
608: "Specific enthalpy of products at state 6, referred to T0"
609: h6_ph = YP_O2*(enthalpy(O2;T=T[6]) - enthalpy(O2;T=T0_ex)) + YP_N2*(enthalpy(N2;T=T[6]) - enthalpy(N2;T=T0_ex)) + YP_CO2*(enthalpy(CO2;T=T[6]) - enthalpy(CO2;T=T0_ex)) + YP_H2O*(enthalpy(H2O;T=T[6]) - enthalpy(H2O;T=T0_ex))
610:
611: "Mixture entropy at state 6"
612: s6_mix = YP_O2*entropy(O2;T=T[6];P=P_exh) + YP_N2*entropy(N2;T=T[6];P=P_exh) + YP_CO2*entropy(CO2;T=T[6];P=P_exh) + YP_H2O*entropy(H2O;T=T[6];P=P_exh)
613:
614: "Mixture entropy at dead state"
615: s0_mix = YP_O2*entropy(O2;T=T0_ex;P=P0_ex) + YP_N2*entropy(N2;T=T0_ex;P=P0_ex) + YP_CO2*entropy(CO2;T=T0_ex;P=P0_ex) + YP_H2O*entropy(H2O;T=T0_ex;P=P0_ex)
616:
617: "Specific physical exergy of exhaust gases"
618: exh_ph = h6_ph - T0_ex*(s6_mix - s0_mix)
619:
620: ex_ph_exh = m_dot_out*((h6_ph)-T0_ex*(s6_mix-s0_mix))
621:
622:
623: "Exergy rate of exhaust gases"
624: Ex_dot_exh = m_dot_out * exh_ph
625: Ex_dot_exh_kW = Ex_dot_exh*convert(W;kW)
626:
627:
628:
629: "-----"
630: "2) Useful work exergy"
631: "-----"
632: Ex_dot_work = W_dot_brake
633:
634: "-----"
635: "3) Fuel exergy (approximation)"
636: "-----"
637: "For ethanol, a common approximation is chemical exergy " beta_ex * LHV"
638: beta_ex = 1,04

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639: Ex_dot_fuel = beta_ex * Q_dot_LHV
640:
641: "-----"
642: "4) Exergy quality factor of exhaust gases"
643: "-----"
644: psi_exh = Ex_dot_exh_kW / Q_exh
645:
646: "-----"
647: "5) Exergy efficiency of the engine"
648: "-----"
649: eta_ex_engine = Ex_dot_work / Ex_dot_fuel
650: eta_ex_engine_percent = eta_ex_engine*100
651:
652: "-----"
653: "6) Exergy destruction / unutilized exergy"
654: "-----"
655: Ex_dot_dest = Ex_dot_fuel - Ex_dot_work - Ex_dot_exh_kW
656: Ex_dot_dest_percent = Ex_dot_dest/Ex_dot_fuel*100
657:
658: "Optional fractions for Sankey"
659: f_ex_fuel = 1
660: f_ex_work = Ex_dot_work/Ex_dot_fuel
661: f_ex_exh = Ex_dot_exh_kW/Ex_dot_fuel
662: f_ex_dest = Ex_dot_dest/Ex_dot_fuel
663:
664: Check_exergy = f_ex_work + f_ex_exh + f_ex_dest
665:
666: Duplicate i=1;6
667:   Vol_cc[i]=Vol[i]*convert(m^3;cm^3)
668:   P_bar[i]=P[i]*convert(Pa;bar)
669:
670:
671: End
672:
673: "=====
674: " UNIFIED ETHANOL ICE + PCM-RANKINE COUPLING"
675: " Rankine inputs are supplied directly by the ethanol engine model."
676: "=====
677:
678: P_RC1 = 1200 [kPa]
679: "Set bounds in Variable Info: 800 to 1200 kPa"
680:
681: P_RC2 = 110,1 [kPa]
682: "Set bounds in Variable Info: 110 to 200 kPa"
683:
684: DeltaT_superheat = 20 [C]
685: "Set bounds in Variable Info: 0 to 20 C"
686:
687: $VarInfo P_RC1 Guess=1200 [kPa] Lower=800 [kPa] Upper=1200 [kPa]
688: $VarInfo P_RC2 Guess=110,1 [kPa] Lower=110 [kPa] Upper=200 [kPa]
689: $VarInfo DeltaT_superheat Guess=20 [C] Lower=0 [C] Upper=20 [C]
690:
691: "Input mapping to Rankine_PCM_CBCC:"
692: "m_dot_exh <- m_dot_out; T_exh_in <- TC_7; Q_dot_exh_total <- Q_exh"
693: "Ex_dot_exh <- Ex_dot_exh_kW; W_dot_engine <- W_dot_brake; Q_dot_fuel <- Q_dot_LHV"
694:
695: Call Rankine_PCM_CBCC (P_RC1; P_RC2; DeltaT_superheat; m_dot_out; TC_7; Q_exh; Ex_dot_exh_kW; W_dot_brake;
  Q_dot_LHV; Q_dot_charge; E_PCM_stored; m_PCM; Q_dot_discharge; m_dot_RC; x_RC2; W_dot_net_RC;
  W_dot_electric_RC; W_dot_electric_RC_direct; eta_RC_electric; eta_global_TES; eta_global_direct; eta_exergy_cycle;
  DeltaT_available_discharge; F_obj)

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82: { MASS FLOW RATE      }
83: {-----}
84:
85: m_dot_RC = W_dot_net_target/w_net
86:
87: W_dot_exp = m_dot_RC*w_exp
88: W_dot_pump = m_dot_RC*w_pump
89: W_dot_net = W_dot_exp - W_dot_pump
90:
91: Q_dot_in = m_dot_RC*q_in
92: Q_dot_out = m_dot_RC*q_out
93:
94: {-----}
95: { PCM / CBCC OPERATION }
96: {-----}
97:
98: Q_PCM_stored = 40500 [kJ]      "Example: 45 kW during 900 s"
99: t_discharge = Q_PCM_stored/Q_dot_in
100:
101: {-----}
102: { DIRECT RANKINE MODE   }
103: {-----}
104:
105: Q_dot_exh_direct = 22,5 [kW]
106:
107: W_dot_RC_direct = eta_RC*Q_dot_exh_direct
108:
109: {-----}
110: { GLOBAL SYSTEM EXAMPLE }
111: {-----}
112:
113: W_dot_engine = 65,87 [kW]
114: Q_dot_fuel = 183,4 [kW]
115:
116: eta_engine = W_dot_engine/Q_dot_fuel
117: eta_global_direct = (W_dot_engine + W_dot_RC_direct)/Q_dot_fuel
118: eta_global_target = (W_dot_engine + W_dot_net_target)/Q_dot_fuel
119:
120: {-----}
121: { USEFUL OUTPUTS      }
122: {-----}
123:
124: T_RC1_C = T_RC1
125: T_RC2_C = T_RC2
126: T_RC3_C = T_RC3
127: T_RC4_C = T_RC4
128:
129: eta_RC_percent = eta_RC*100
130: eta_engine_percent = eta_engine*100
131: eta_global_direct_percent = eta_global_direct*100
132: eta_global_target_percent = eta_global_target*100

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