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Vol. 32, No. 3. \ 2024 ISSN: 2616 - 9916

"Thermal Energy Recovery from Automotive Engine Exhaust Using a Shell-and-Tube Heat Exchanger: A Sustainable Approach to Waste Heat Utilization"

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Received:	**15/5/2025**	**Accepted:**	**16/6/2025**	**Published:**	**29/6/2025**

Abstract

Experimental investigations were conducted to quantify the recoverable thermal energy from the exhaust of a 60 kW internal combustion engine (ICE). The evaluation of the shell-and-tube heat exchanger's performance was conducted using water as the working fluid for thermal energy transfer. Utilizing the acquired data, a computational simulation was conducted to optimize both the thermal performance and geometric configuration of the heat exchanger. Two separate heat exchangers were incorporated into the system: one dedicated to the generation of saturated steam and the other designed for the production of superheated vapor. These units were configured in both parallel and series arrangements. In the series configuration, exhaust gases first traversed the superheater followed by the saturation unit. Conversely, in the parallel setup, the exhaust stream was split, with each fraction passing through either the superheated or saturated exchanger simultaneously. In both configurations, the water stream was first introduced to the saturation unit, then routed through the superheater. Significantly, the analysis revealed that under operating pressures below 30 bar, the parallel configuration yielded superior power output, whereas at pressures exceeding 30 bar, the series configuration demonstrated greater performance. The turbine output power was also estimated by incorporating realistic isentropic efficiency assumptions. The findings reveal that harnessing the available waste heat from the diesel engine can yield a minimum additional power output of 18%, underscoring the system’s capability to improve overall energy efficiency in a sustainable and economically viable way.

Keywords :Diesel Engine. Thermal Energy. Sustainability. Heat Recovery. Waste Heat.

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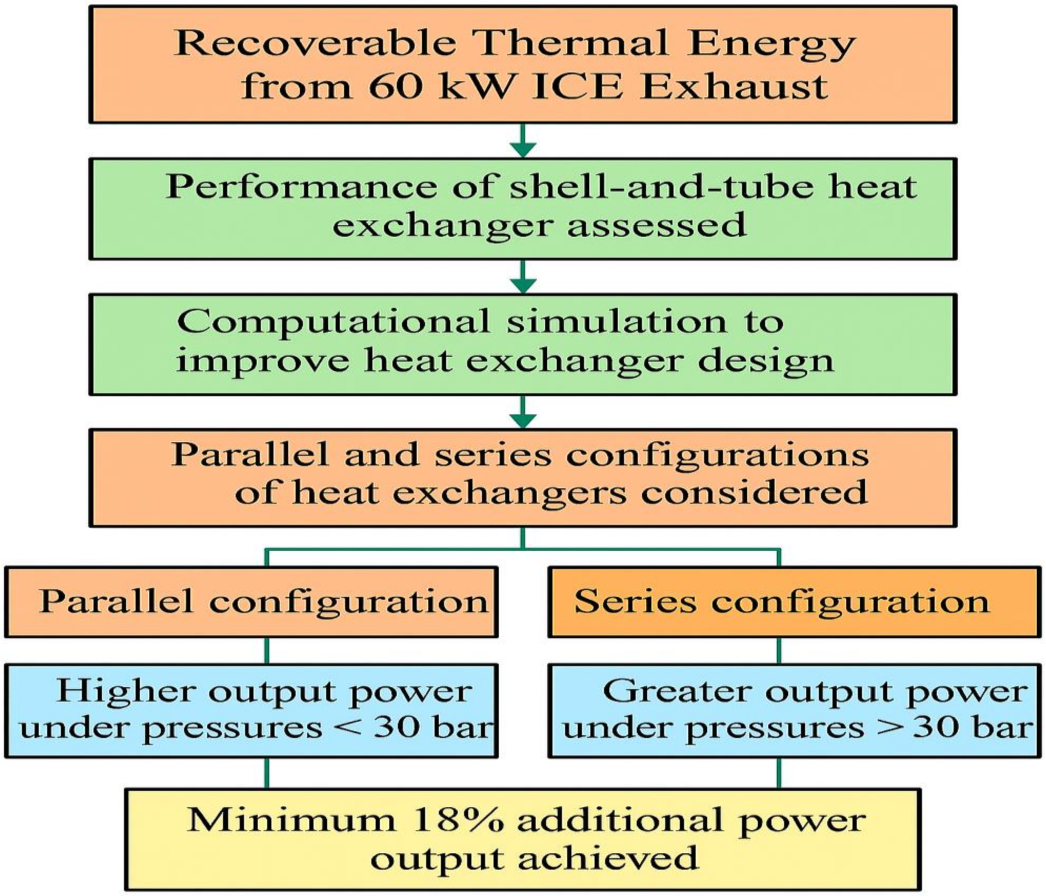
Graphical Abstract



1. Introduction

Dieselengines constitute a major class of internal combustion engines (ICEs), widely utilized across various industrial and transportation sectors, widely acknowledged for their high thermal efficiency and broad range of applications. These engines are commonly utilized as prime movers across diverse sectors, including agriculture, construction, marine transport, and power generation. Compact, air-cooled diesel engines with capacities up to 35 kW are typically deployed in irrigation systems, small-scale agricultural machinery, and construction equipment. In contrast, larger agricultural operations employ tractors with power outputs reaching up to 150 kW. For applications within the 35–150 kW range, both air- and water-cooled engines are available; however, water-cooled configurations are generally preferred for higher power levels due to their superior thermal management capabilities, unless specific operational requirements necessitate air-cooled systems [1].

In heavy-duty applications, such as earth-moving machinery, diesel engines frequently achieve outputs of 520 kW or higher, extending to 740 kW in extreme scenarios. Marine propulsion systems generally require engines with a minimum output of 150 kW, while high-speed diesel engines exceeding 220 kW are typically employed in on-road vehicles, such as trucks. Furthermore, diesel engines are extensively used in standby and distributed power generation systems, functioning as auxiliary energy sources for small- to mid-capacity electrical grids. Despite their efficiency, diesel engines typically convert only approximately 35% of the input fuel energy into mechanical work, with the remaining energy primarily dissipated as heat through exhaust gases and coolant fluids. Although recent advancements have led to marginal improvements in efficiency, a substantial portion of the energy remains unrecovered and is released into the environment [2].

In water-cooled systems, about 35% of the input energy is lost through the coolant, with an additional 30–40% discharged via exhaust gases [3]. The potential to recover this waste heat is significantly influenced by engine load conditions. According to Johnson, a standard 3.0-liter diesel engine with a rated output of 115 kW can dissipate between 20 kW and 40 kW of waste heat, contingent upon its operating regime. For typical driving cycles, the average recoverable thermal power from exhaust gases is estimated to be approximately 23 kW, highlighting a considerable opportunity for energy recovery and reuse within the framework of sustainable thermal management. Given that nearly two-thirds of the input fuel energy is lost as thermal waste—primarily via engine coolant and exhaust emissions—recovering exhaust heat from diesel engines offers a valuable pathway to enhance overall system efficiency and fuel economy. Exhaust gases, in particular, represent a high-grade thermal energy source that can be exploited using advanced thermodynamic cycles to generate supplementary mechanical or electrical power [4].

These heat recovery techniques are actively being developed by both engine manufacturers and research institutions, reflecting a growing interest in sustainable energy conversion technologies. Among the various strategies under exploration, the integration of a Bottoming Organic Rankine Cycle (ORC) has emerged as a particularly promising solution, this is particularly true for heavy-duty diesel engines used in commercial transportation [5].

The application of the Rankine Cycle in vehicular systems traces back to the early 1970s, when a U.S. Department of Energy (DOE)-funded initiative—spearheaded by Mack Trucks in collaboration with Thermo Electron Corporation successfully demonstrated its technical viability. In that initiative, an ORC unit was retrofitted to a Mack diesel engine, with laboratory tests showing a reduction in brake-specific fuel consumption (bsfc) by 10–12%, a finding further validated by highway on-road trials [6].

Over subsequent years, similar efforts by other automotive manufacturers and research organizations have contributed to the continual optimization of Rankine-based recovery systems. More recently, the competitiveness of waste heat recovery via Rankine Cycle Systems (RCS) has improved significantly, driven by advancements in core components such as heat exchangers, condensers, and expanders, in tandem with the increasing cost of fossil fuels [7].

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Consequently, RCS integration is now considered both a technically and economically viable solution not only for heavy-duty diesel platforms [7] but also for light-duty applications, including passenger vehicles [9], thereby reinforcing its role in the decarbonization and energy optimization of the transportation sector. Apart from turbo-compounding systems, the majority of current technologies aimed at partial recovery of exhaust energy rely on compact heat exchangers to extract thermal energy from engine exhaust gases [8].

For high-temperature applications, these heat exchangers must provide sufficient heat transfer surface area to manage the thermal load while conforming to practical limitations related to size, weight, and integration within the engine system. Additionally, the associated pressure drop must remain within acceptable limits to prevent excessive parasitic losses that could diminish net engine output and reduce overall thermal efficiency[10].

This study primarily aims to investigate the feasibility of exhaust heat recovery from a 60 kW diesel-powered automotive engine through both experimental and numerical approaches. The objective is to enhance thermal efficiency by incorporating a shell-and-tube heat exchanger into the engine's exhaust system. The research focuses on evaluating the thermal performance of the exchanger across varying engine speeds and load conditions and optimizing its design for maximum energy recovery. To achieve this, a series of controlled experiments were conducted in which the heat exchanger, using water as the working fluid, was integrated into the exhaust flow. The system's thermal characteristics and dynamic operational performance were systematically analyzed. Based on the experimental results, a detailed numerical model was developed to refine the heat exchanger's design [11].

A significant contribution of this work lies in the development of an optimized heat exchanger configuration capable of generating superheated steam suitable for application in a bottoming thermodynamic cycle. The turbine output power was subsequently estimated by applying an isentropic efficiency model, thereby quantifying the system's energy recovery potential and demonstrating the feasibility of integrating such a solution into conventional diesel engine platforms [12].

2. Experimental setup

The experimental setup utilized a four-cylinder Isuzu 4JB1-T diesel engine, which was integrated with a precision water-cooled dynamometer to ensure accurate control and measurement of the applied mechanical load [13]. The engine's key specifications are outlined in (Table. 1), while a schematic overview of the experimental arrangement is presented in (Fig. 1). During testing, the engine operated under various load conditions at a constant rotational speed, allowing for systematic measurement of exhaust gas temperatures to quantify the recoverable thermal energy. The exhaust stream was subsequently routed through a shell-and-tube heat exchanger to assess its thermal performance. The resulting experimental data served as foundational input for computational simulations aimed at optimizing the exchanger's thermal and geometric characteristics [14].

Table 1 Engine Specifications of the Isuzu 4JB1-T Diesel Engine

Parameter	Specification
Engine model	4JB1-T
Manufacturer	Isuzu
Engine type	Four-cylinder, turbocharged diesel engine with water cooling
Cylinder diameter	93 mm
Piston travel length	102 mm
Compression ratio	17.5:1
Peak torque output	215 N·m @ 2200 rpm

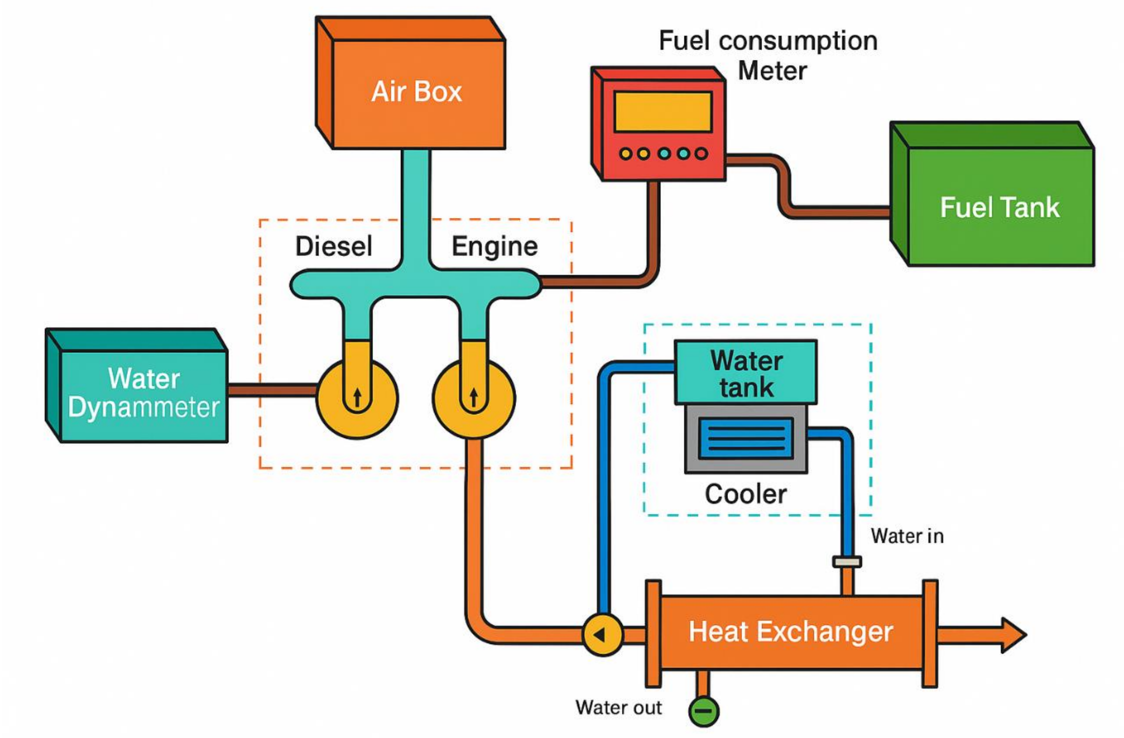

Figure 1: Illustrative schematic of the experimental configuration.

3. Configuration and Design Parameters of the Heat Exchanger

The experimental data gathered during the trials served as the foundation for optimizing the shell-and-tube heat exchanger design using advanced computational simulation methods. Key design parameters—including shell diameter, tube count, total length, and internal pressure drop—were thoroughly evaluated to determine their impact on thermal performance [15]. The parametric analysis facilitated the identification of an optimal configuration, resulting in a

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refined heat exchanger model tailored for enhanced energy recovery efficiency. The detailed specifications of the finalized shell-and-tube heat exchanger design are presented in (Table. 2) [16].

Table 2 Engine specification

Parameter	Specification
Heat Exchanger Type	**A counterflow shell-and-tube exchanger was used, with hot fluid in the tubes and cold fluid in the shell.**
Shell Inner Diameter	**70 mm**
Number of Tubes	**18**
Tube Inner Diameter	**10 mm**
Effective Heat Exchanger Length	**2.0 meters**

Two separate heat exchangers were employed: the first unit was designed to generate saturated steam from subcooled water, while the second unit was responsible for converting the saturated steam into superheated vapor [17]. These two units were configured in two different thermal arrangements, namely parallel and series configurations, as illustrated in (Fig. 2). In the series configuration, the exhaust gas sequentially passes through the superheater and then the steam generator. Conversely, in the parallel configuration, the exhaust stream is bifurcated and distributed across both units simultaneously [18].

3.1 Computational Fluid Dynamics (CFD) Model

The optimized shell-and-tube heat exchanger configuration was evaluated for thermal performance using the Flow Simulation CFD module within SolidWorks 2009. The simulation mesh comprised 109,992 cells, affording high spatial resolution. In the secondary heat exchanger (Fig. 2), the cold stream was represented by water maintained at 301 K, whereas in the primary exchanger (Fig. 2), the cold side was modeled as saturated steam at its operating pressure [19]. The hot stream was treated as air with a mass flow rate of 0.10215 kg/s and an inlet temperature of 938 K, at an absolute pressure of 101.325 kPa. Cold-side inlet pressure and mass flow rate were varied parametrically. All simulations assumed steady, incompressible flow. Turbulence was captured using the standard k–ε RANS model, a two-equation approach widely adopted for industrial fluid-flow and heat-transfer analyses [20].

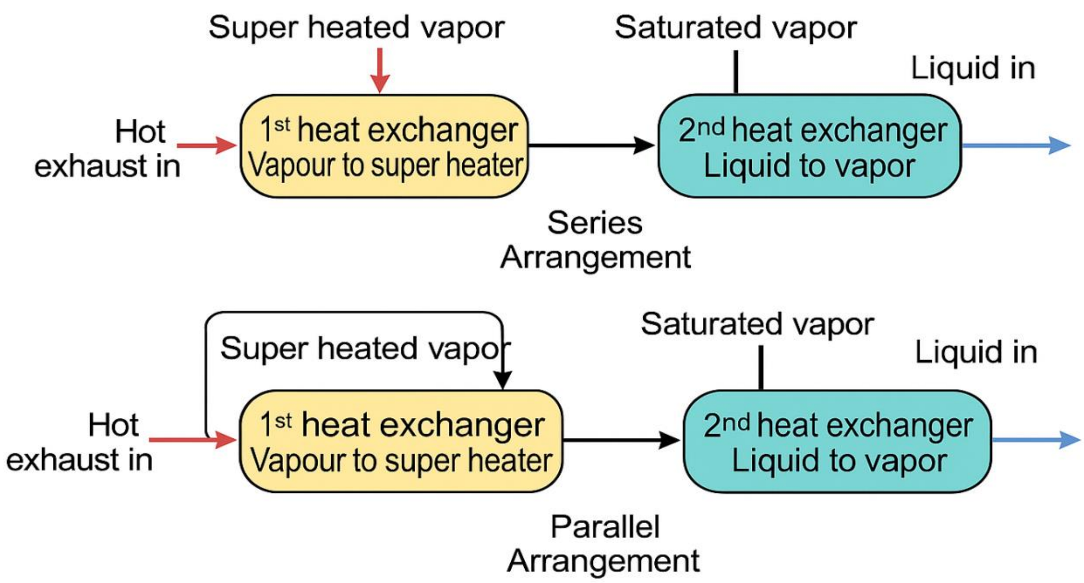

Figure 2: Heat Exchanger Arrangement

4. Results

Effective heat exchanger design for engine exhaust heat recovery necessitates quantifying the thermal energy available in the exhaust stream. Accordingly, baseline experiments were performed to assess exhaust gas temperatures under varying engine speeds and power outputs. As shown in (Fig. 3), the results reveal an approximately linear relationship between engine powers and exhaust temperature across the three tested rotational speeds [21]. Higher speeds and loads correspond to increased exhaust temperatures, indicating enhanced potential for thermal energy recovery under high-power operating conditions [22].

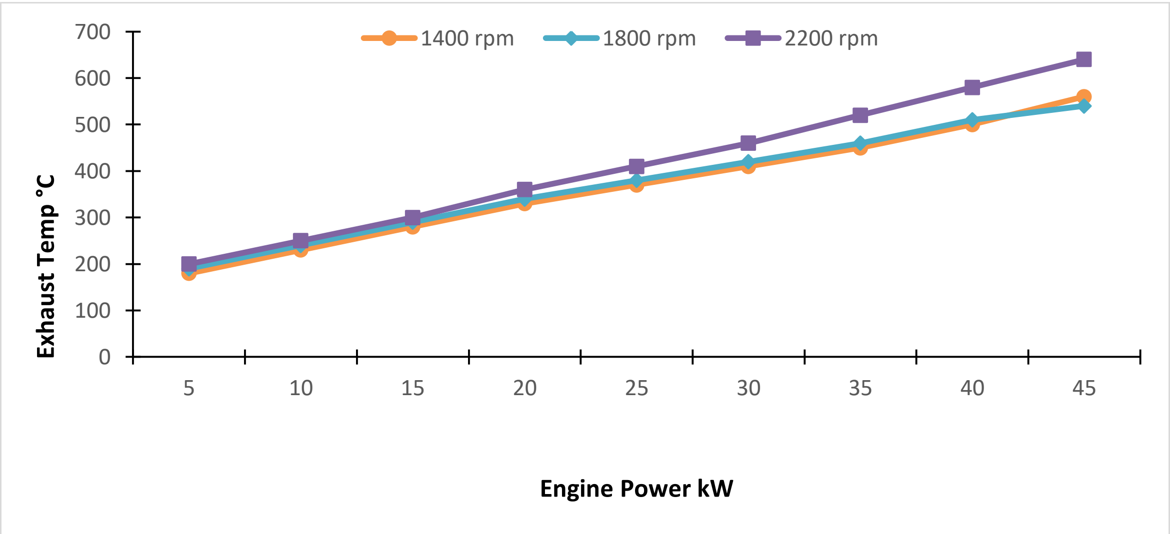

Figure 3: Variation of Exhaust Gas Temperature with Engine Power

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(Fig. 4) analysis of the relationship between engine power and the amount of recoverable thermal energy from exhaust gases reveals a generally increasing trend, although the relationship is not perfectly linear. This underlines the importance of waste energy recovery at higher engine loads. The results strongly validate the motivation behind this study, confirming that a considerable portion of waste heat is available for conversion into useful energy. For example, at the lowest tested engine speed of 1400 rpm, the engine produced approximately 35 kW of power, yielding a recoverable energy of about 15.5 kW [23]. At 1800 rpm, with the engine generating around 40 kW, the recoverable energy was roughly 17 kW. Meanwhile, at 2200 rpm, the engine produced approximately 45 kW, with a recoverable energy of about 18 kW. These values suggest that depending on engine speed and load, up to 50% of the engine's output power could potentially be recovered from the exhaust gases. These estimates assume the heat exchanger is capable of lowering the exhaust gas temperature to 50°C under all operating conditions [24]. Building on the experimental results, the shell and tube heat exchanger design was further optimized using computer simulations. As illustrated in (Fig. 4), the heat exchanger's effectiveness tends to decrease with increasing shell diameter due to a drop in turbulence, which limits heat transfer. In contrast, variations in working pressure showed minimal effect on performance [25].

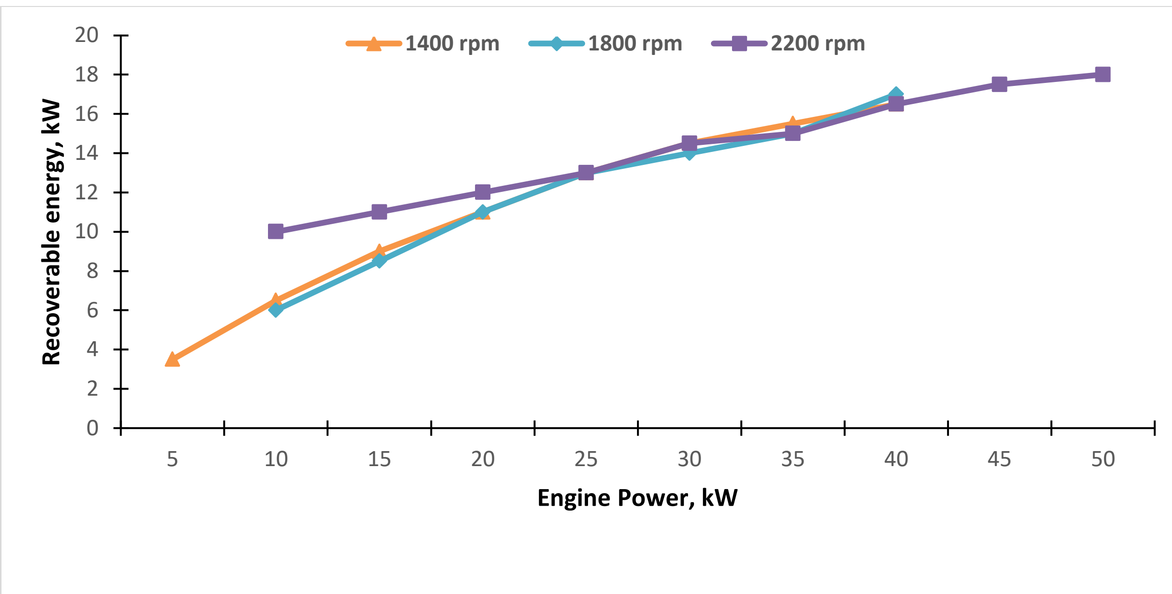

Figure 4: Empirical Evaluation of Load-Dependent Recoverable Energy in Internal Combustion Engines

(Fig. 5) presents the relationship between the shell diameter of a shell-and-tube heat exchanger and its corresponding thermal effectiveness under varying working pressures (10, 20, 30, and 40 bar). The data indicate a clear negative correlation between shell diameter and effectiveness, particularly beyond a diameter of 44 mm [26]. Initially, from 38 mm to approximately 44 mm, the effectiveness remains relatively stable, ranging between 0.66 and 0.67, with only marginal gains. This suggests an optimal range for shell diameter where turbulence and surface contact are maximized, leading to efficient heat exchange. However, beyond this threshold (i.e., above 44 mm), the effectiveness shows a consistent and significant decline, dropping to around 0.56 at 58 mm shell diameter [27]. This reduction is attributed to the diminished turbulence and increased cross-sectional flow area, which weakens convective heat transfer due to a lower Reynolds number in the shell side flow. Notably, the impact of working pressure is minimal across the tested range. The curves for 10, 20, 30, and 40 bar pressures exhibit nearly identical trends, indicating that shell diameter is the dominant parameter influencing effectiveness within the considered pressure range. This analysis underscores the critical role of shell diameter optimization in heat exchanger design. Maintaining a smaller shell diameter promotes higher turbulence levels, thereby enhancing heat transfer efficiency. Conversely, increasing the diameter beyond the optimal point leads to diminishing returns and compromises the exchanger's performance, regardless of pressure conditions [28].

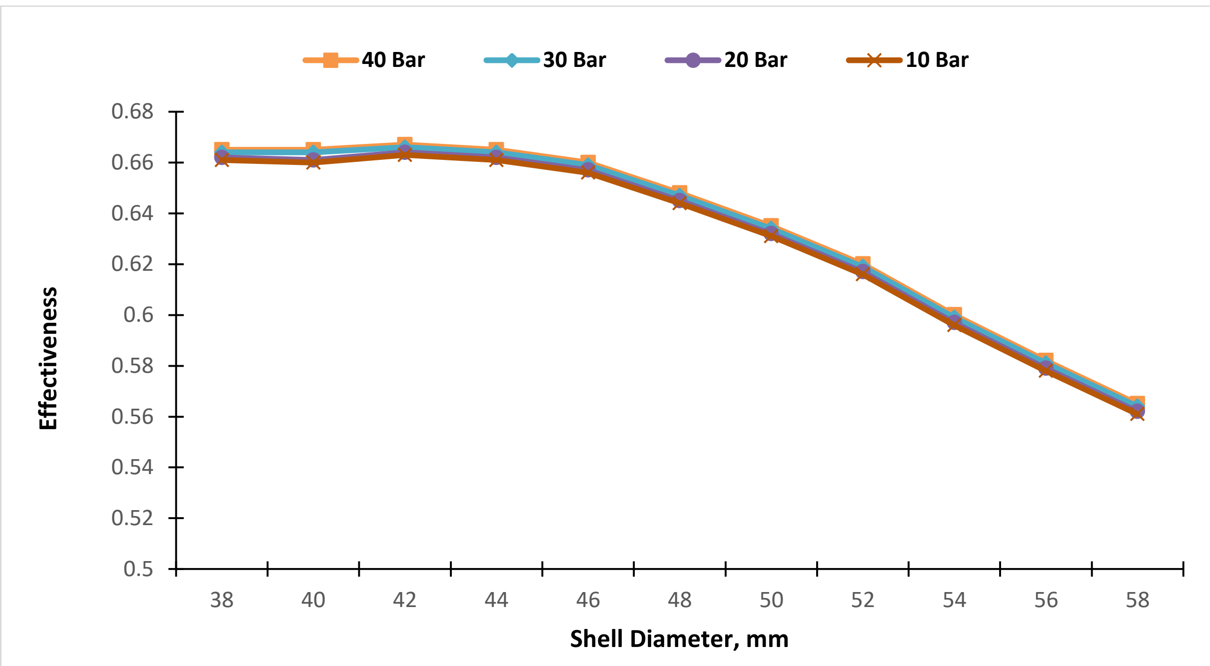

Figure 5: Effectiveness of Heat Exchanger as a Function of Shell Diameter Based on CFD Analysis

(Fig. 6) shows the relationship between heat exchanger length and thermal effectiveness across various working pressures (10, 20, 30, and 40 bar). As the length increases from 1.0 m to 2.5 m, a clear trend of increasing effectiveness is observed, with values rising from approximately 0.74 to 0.90. However, this improvement begins to plateau after around 1.8 meters, indicating diminishing returns with further length extension [29]. The convergence of performance across all pressure levels suggests that length plays a more dominant role in influencing effectiveness than pressure within this range. (Fig. 7) evaluates the additional output power that can be recovered from the exhaust stream of a diesel engine using a dual-stage shell-and-tube heat exchanger configuration, comparing series and parallel arrangements under

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varying working pressures. The data indicate that parallel configuration consistently outperforms series at lower to moderate pressures, achieving a maximum output of approximately 12.6 kW at 30 bar, compared to 11 kW for the series setup at the same pressure — a ~15% performance gain. However, at higher pressures (beyond ~35 bar), the series configuration exhibits superior performance, suggesting that its design better accommodates high-pressure operational regimes. The optimal power recovery for both configurations occurs near 30–34.5 bar, after which performance slightly declines, likely due to increased thermal losses or system inefficiencies at higher pressures [29]. These findings underscore the importance of system arrangement selection based on anticipated pressure conditions, with parallel setups being ideal for subcritical applications and series configurations better suited for higher-pressure environments. Overall, the proposed system demonstrates the capacity to reclaim up to 20.7% of the diesel engine's exhaust thermal energy as usable output power, assuming an isentropic turbine efficiency of 85%. This marks a substantial improvement from the previously estimated 18%, reinforcing the effectiveness and feasibility of integrating waste heat recovery (WHR) systems into mid-scale diesel engine operations for enhanced energy efficiency and sustainability [30].

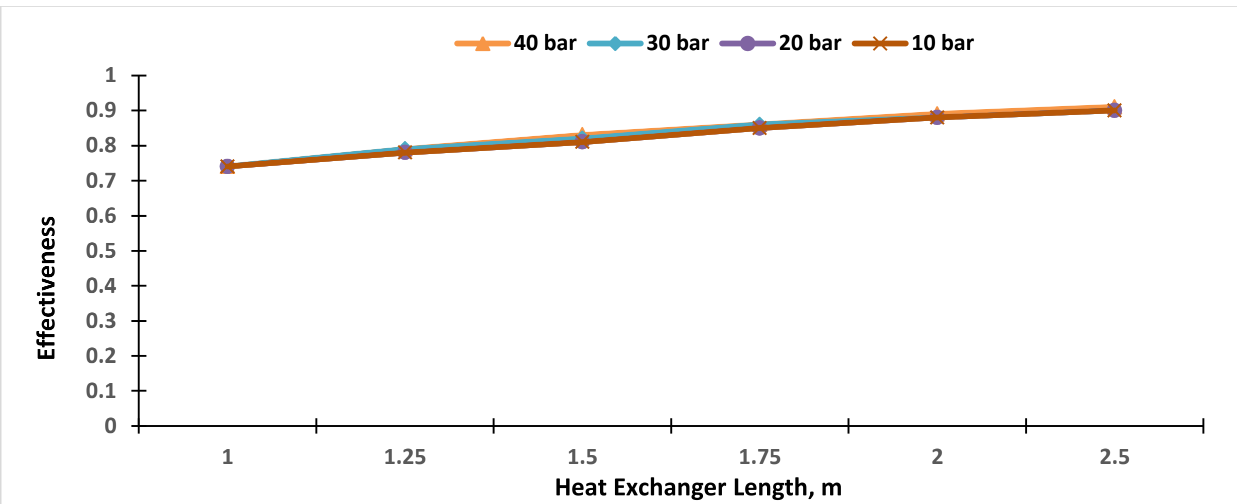

Figure 6 Effectiveness of Heat Exchanger as a Function of Length Based on CFD Simulation

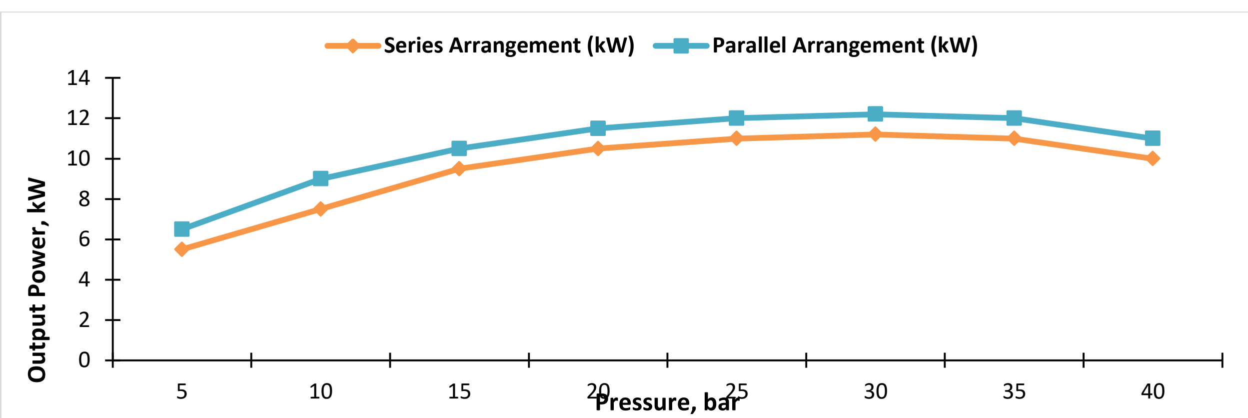

Figure 7 Variation of Net Power Output with Working Pressure Based on CFD Simulation

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5. Statistical Assessment of Power Output Across Engine Speeds Using ANOVA

A comprehensive statistical analysis was performed to evaluate the influence of engine speed (1400 rpm, 1800 rpm, and 2200 rpm) on the power output of the thermal recovery system under varying load conditions. The analysis was conducted using one-way ANOVA, and the results are summarized in (Table. 3). The degrees of freedom (df) for between-group variation is 2, corresponding to the three different engine speeds, while within-group variation accounts for 24 degrees of freedom [31]. The calculated F-value (F calc.) was found to be 0.309, which is substantially lower than the critical F-value (F crit.) of 3.403 at a 95% confidence level. Additionally, the P-value associated with the F-test was 0.737, indicating no statistically significant difference in recovered power output across the tested engine speeds [32].

These findings suggest that within the operational ranges considered, engine speed does not have a statistically meaningful effect on the performance of the proposed dual-stage shell-and-tube heat exchanger system [33]. The stability of the recovered power across different speeds implies a robust and consistent performance of the thermal recovery setup, thereby enhancing its practical applicability for real-world diesel engine systems. This statistical validation further reinforces the system’s reliability and underscores its suitability for integration into various mid-scale engine applications without performance sensitivity to minor speed variations [34].

Table 3: ANOVA Results for Power Output at Different RPMs

Source of Variation	df	SS (Sum of Squares)	MS (Mean Square)	F calc.	P-value	F crit.
Between Groups	2	10,896.30	5,448.15	0.309	0.737	3.403
Within Groups	24	422,577.78	17,607.41			
Total	26	433,474.07				

The results in (Table. 3) clearly indicate the lack of statistically significant differences in recovered power output across engine speeds, highlighting the versatility and operational consistency of the thermal recovery configuration in diesel applications [34].

6 Conclusions

The experimental and simulation results confirm the feasibility and effectiveness of recovering waste heat from diesel engine exhaust using a dual-stage shell-and-tube heat exchanger. This system significantly improves overall engine efficiency and supports reductions in fuel consumption, greenhouse gas emissions, and harmful pollutants, enhancing environmental sustainability. The study demonstrated that up to approximately 20.7% of the engine’s exhaust thermal energy can be recovered as usable power, assuming an isentropic turbine efficiency of 85%. Performance is highly dependent on system configuration and operating conditions. The parallel heat exchanger arrangement yielded higher power recovery at subcritical pressures (below ~34.5 bar), showing around a 15% increase at 30 bar. In contrast, at higher pressures, the series configuration became more effective, indicating that optimal arrangement selection should be pressure-specific. Geometric parameters also influenced thermal performance. Increasing the heat exchanger length improved effectiveness, reaching values close to 0.90 at 2.5 meters. However, gains diminished beyond approximately 1.8 meters, suggesting limited benefit from further length extension. While working pressure had minimal impact on effectiveness, larger shell diameters led to reduced performance due to lower turbulence and heat transfer rates. These findings underscore the potential of optimized thermal recovery systems to enhance diesel engine efficiency and sustainability. Proper configuration of heat exchanger design and operating conditions can significantly improve waste heat utilization, offering a viable solution for energy and emissions optimization in diesel-powered systems (Table.4).

Table 4 Summary of Key Results from the Waste Heat Recovery Study

Parameter	Result
Maximum Recoverable Power	~20.7% of engine exhaust energy (at 85% turbine efficiency)
Optimal Working Pressure	~34.5 bar
Best Configuration (≤34.5 bar)	Parallel arrangement (~15% higher output at 30 bar)
Best Configuration (>34.5 bar)	Series arrangement
Heat Exchanger Length Effectiveness Plateau	~1.8 meters
Maximum Heat Exchanger Effectiveness	~0.90 at 2.5 meters length
Shell Diameter Impact	Effectiveness decreases with diameter increase due to reduced turbulence.
Pressure Impact on Effectiveness	Minimal; effectiveness nearly constant across 10–40 bar

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ARTICLE
JOURNAL`S UNIVERSITY OF BABYLON FOR ENGINEERING SCIENCES (JUBES)
مـــجلــة جـــــامعة بـــــــابـل للعلــــــوم الهندسية
Vol. 32, No. 3. \ 2024
ISSN: 2616 - 9916

"استعادة الطاقة الحرارية من عادم محرك السيارة باستخدام مبادل حراري من نوع الغلاف والأنابيب: نهج مستدام لاستغلال الحرارة المهدرة"

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الخلاصة

تم إجراء تجارب مخبرية لقياس كمية الطاقة الحرارية القابلة للاسترداد من عادم محرك احتراق داخلي بقدرة 60 كيلوواط. تم تقييم أداء المبادل الحراري من نوع الغلاف والأنابيب باستخدام الماء كسائل عامل لنقل الطاقة الحرارية. وبالاستفادة من البيانات المُجمعة، أُجريَت محاكاة حاسوبية لتحسين الأداء الحراري والتكوين الهندسي للمبادل الحراري. تضمن النظام مبادلين حراريين منفصلين: أحدهما مخصص لتوليد البخار المشبع، والآخر لإنتاج البخار المحمص. تم ترتيب هذه الوحدات في تكوينين: متوازٍ ومتسلسل. في التكوين المتسلسل، يمر غاز العادم أولاً عبر وحدة التحميص، ثم إلى وحدة التشبع. بينما في التكوين المتوازي، ينقسم تيار العادم، ليمر كل جزء من التيار عبر أحد المبدلين في الوقت ذاته. في كلا التكوينين، يُدخل تيار الماء أولاً إلى وحدة التشبع، ثم يوجه إلى وحدة التحميص. وأظهرت التحليلات أنه عند ضغوط تشغيل أقل من 30 بار، يوفر التكوين المتوازي ناتج طاقة أعلى، بينما عند الضغوط التي تتجاوز 30 بار، يحقق التكوين المتسلسل أداءً أفضل. كما تم تقدير ناتج طاقة التوربين باستخدام افتراضات واقعية للكفاءة الأيزنتروبية. تكشف النتائج أن استغلال الحرارة المهدرة من محرك الديزل يمكن أن يوفر ناتج طاقة إضافي لا يقل عن 18%، مما يبرز قدرة النظام على تحسين كفاءة الطاقة الكلية بطريقة مستدامة وقابلة للتطبيق اقتصادياً.

الكلمات الدالة :محرك ديزل، الطاقة الحرارية، الاستدامة، استرداد الحرارة، الحرارة المهدرة.