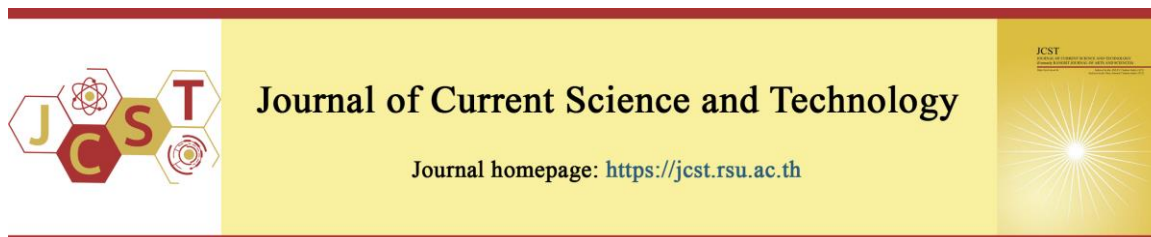


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Waste Heat Recovery from Biomass Charcoal Cookstoves via Elliptical-Tube Helical Coils: Application in a Medium Sized Thai Restaurant

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Abstract

This study proposes a compact and high-performance waste-heat recovery system designed for small food-service establishments using biomass cookstoves (BCS). The system incorporates elliptical-tube helical coil heat exchangers (HHEs) to intensify secondary flows and enhance convective heat transfer compared with that of conventional circular-tube designs. Four geometric configurations (H01–H04), differing in coil diameters, pitch lengths, and tube ovalities, were evaluated through an integrated framework consisting of: (i) CFD-based thermal assessment under wall-temperature conditions of 160°C, 180°C, and 200°C; (ii) techno-economic analysis, including annual energy cost savings, net present value (NPV), and payback period; and (iii) environmental impact assessment focusing on CO₂ emission reduction. Heat-transfer oil was used as the working fluid at a constant inlet temperature of 30°C. The CFD results confirm the strong influence of coil geometry on heat-transfer enhancement. H01 consistently demonstrates the highest thermal performance, outperforming H02, H03, and H04 by 2.78%, 4.10%, and 12.83%, respectively. Deploying H01 as both the exhaust-gas (HE_a) and hot-water (HE_b) recovery units under optimal conditions (200°C and $\dot{m} = 2.5$ L/min) yields 17,144.34 kWh/year of recoverable thermal energy, equivalent to 71,422.27 THB in electricity savings. Techno-economic indicators further reveal strong feasibility, with an NPV of 263,887.97 THB and a payback period of only 1.36 years. CO₂ emission reduction reaches 3,948.17 kg/year, highlighting the system's environmental significance. Overall, the elliptical-tube HHE configuration (H01) offers superior energy, economic, and environmental benefits for BCS-based food enterprises, aligning effectively with renewable-energy goals and contributing to more sustainable thermal practices in the food-service sector.

Keywords: biomass cookstove, waste heat recovery, CFD, economic evaluation, sustainable restaurant operation

1. Introduction

The food and beverage sector is a major economic driver in Thailand, contributing approximately 28.1% of the national Gross domestic product (GDP). Small- and medium-sized food-service enterprises (SMEs) are particularly important in terms of employment, local economic development, and social prosperity. However, these businesses increasingly struggle with rising thermal-

energy costs due to their high temperature and continuous cooking requirements. In response, biomass cookstoves (BCS) are widely adopted because charcoal offers low fuel cost and a high calorific value (~30 MJ/kg) (Lohri et al., 2016). Yet, current BCS heating systems achieve only 23.7–27.6% thermal efficiency (Deng et al., 2023), indicating substantial untapped waste-heat potential.

Recent advancements in BCS performance have largely focused on improving combustion processes through computational fluid dynamics (CFD), such as chamber-material optimization (Ghiwe et al., 2024), advanced airflow control (Hayyat et al., 2025), and emission reduction (Kole & Ancha, 2025). Although these developments have improved heating efficiency, a significant fraction of the generated thermal energy continues to be lost through high-temperature exhaust gases without being recovered.

Waste-heat recovery (WHR) systems have been actively investigated across various sectors—including domestic wastewater (Ali et al., 2024), industrial low-grade heat (Ravindran et al., 2024), kitchen wastewater (Singh et al., 2023), and distributed heat-to-power conversion (Guerron et al., 2024). These studies confirm the potential of WHR to reduce energy costs and carbon emissions. However, most existing WHR solutions remain targeted toward medium- to large-scale industrial applications, where installation space, capital investment, and steady-state heat availability are less restrictive. In contrast, small-scale food service SMEs require compact, cost-efficient, and operationally simple WHR systems, a need that remains largely underexplored in literature.

Helical-coil heat exchangers (HHEs) represent a promising compact WHR technology due to enhanced secondary-flow structures, high structural stability, and adaptable geometric form. Prior studies (Chater et al., 2023; Ghazanfari et al., 2024; Zheng et al., 2024) have demonstrated strong heat-transfer performance in helical-coil systems used in high-efficiency thermal applications. Yet, these studies have primarily focused on circular-tube configurations, which may not fully exploit opportunities for convective enhancement. In particular, the influence of elliptical-tube cross-sections, combined with varied coil curvature and pitch, on WHR performance under real BCS operating conditions has not been systematically investigated.

Therefore, the novelty of this work lies in the development and assessment of an elliptical-tube HHE system tailored for small food-service SMEs using BCS waste-heat recovery. This study provides:

1. Design and CFD-based evaluation of four elliptical-tube HHE geometries for BCS exhaust-heat recovery

2. Integrated energy, economic, and environmental performance assessment to quantify adoption feasibility

3. Identification of an optimal HHE design that yields the highest practical impact for SME food-service operations

This research ultimately aims to deliver a scalable WHR solution that sustainably reduces thermal-energy costs and CO₂ emissions in Thailand's food-service industry.

2. Objectives

This study aims to design, analyze, and assess a compact waste-heat recovery system for small food-service establishments using biomass cookstoves (BCS). The specific objectives are as follows:

1. To evaluate and compare the thermal performance of four elliptical-tube helical coil heat exchanger (HHE) geometries (H01–H04) using CFD simulations under practical BCS operating conditions, and to identify the optimal configuration.

2. To investigate the influence of key operating parameters, including coil-surface temperature and working-fluid flow rate, on the heat-transfer performance, thermal-hydraulic behavior, and annual recoverable thermal energy of the selected HHE when applied to both the exhaust-gas recovery unit (HE_a) and the hot-water generation unit (HE_b).

3. To assess the overall techno-economic and environmental feasibility of the proposed WHR system by quantifying annual cost savings, net present value (NPV), payback period, and CO₂ emission reduction in comparison with conventional hot-water production without heat recovery.

3. Materials and Methods

The overall methodological framework of this study is illustrated in Figure 1, which consists of five sequential stages: (i) development of 3D elliptical-tube helical coil geometries and CFD simulations, (ii) analysis of thermal and flow characteristics, (iii) comparative thermal performance evaluation, (iv) selection of the optimal configuration, and (v) system-level economic and environmental assessments.

This structured approach ensures a direct linkage between engineering design analysis and real-world implementation feasibility.

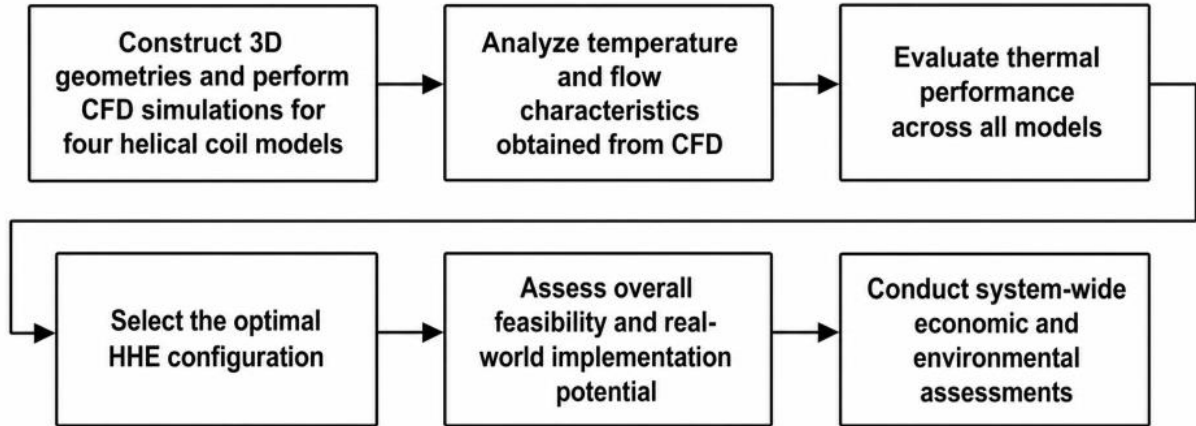


Figure 1 Overall methodological workflow of the proposed study

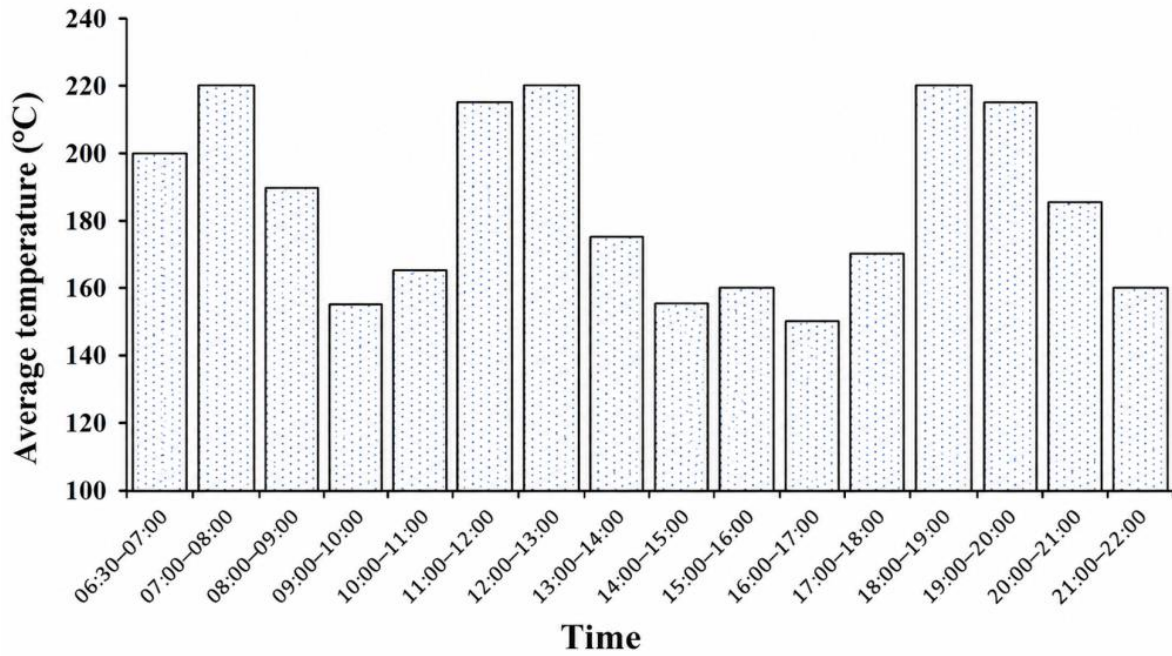


Figure 2 Hourly variation of exhaust gas temperature measured during biomass cookstove grill operation in the demonstration restaurant. The results highlight the high-grade waste-heat potential available for thermal recovery

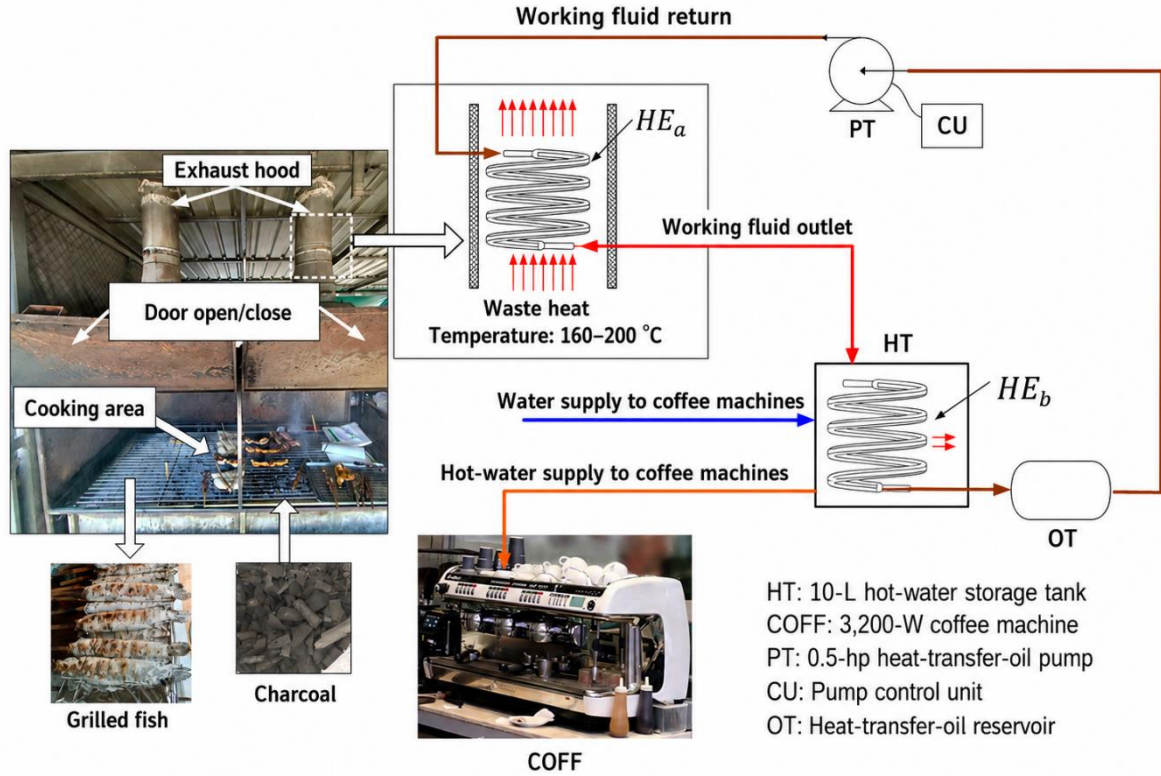


Figure 3 Schematic layout of the proposed waste-heat recovery system for biomass cookstoves, showing HE_a installed in the exhaust-hood region for flue-gas energy capture and HE_b integrated into the water-heating loop supplying a commercial coffee machine

3.1 Demonstration Study Site

In this study, operational data were collected from a medium sized restaurant located in an eastern province of Thailand. The establishment can accommodate up to 200 tables and primarily serves Thai cuisine and beverages. In addition, the restaurant operates an in-house café offering coffee and bakery products, which customers may consume on-site or purchase for takeaway. The business operates for 17 hours per day, from 07:00 to 24:00, and remains open for an average of 310 days per year.

Energy-related expenses can be categorized into two primary components: electricity consumption and cooking fuel. Among these, the cost of cooking fuel is a critical factor influencing the restaurant's overall operating costs and profit margins. The restaurant relies on two types of fuel: liquefied petroleum gas (LPG) for gas burners and charcoal for biomass cookstoves (BCS).

With regard to waste heat, cooking processes involving biomass cookstoves (BCS) generate a significant amount of unused thermal energy. As

shown in Figure 2, the exhaust gas temperature measured during grill operation remains consistently high throughout the cooking period, clearly indicating the presence of high-grade waste heat. These observations confirm that BCS constitute a highly promising and reliable source of recoverable thermal energy for secondary heat recovery applications

The design of the heat recovery system, as illustrated in Figure 3, aims to utilize the recovered thermal energy from the biomass cookstove (BCS) to produce hot water for a commercial coffee machine. The system employs two helical coil heat exchangers (denoted as HE_a and HE_b), both fabricated from bent copper tubing with an oval cross-sectional profile. This geometry was selected due to its superior heat-transfer performance compared with that of conventional circular tubes; the oval cross-section enhances secondary flow generation and induces stronger turbulence within the tube, thereby reducing the thermal boundary-layer thickness. Consequently, the Nusselt number can increase by up to 35%, while the associated pressure drop rises by no more than

5% (Ghazanfari et al., 2024). In addition, the improved uniformity of the working-fluid distribution enhances thermal stability, making the design particularly suitable for compact heat exchangers and waste heat recovery applications (Ghazanfari et al., 2024).

The operating principle of the system begins with waste heat from the BCS being transferred to the first helical coil heat exchanger (HE_a), which is installed inside the exhaust hood, as shown in Figure 3. This design builds upon the work of Kong et al. (2018), who investigated the use of helical coils for recovering waste heat from flue-gas ducts, using deionized water (DI water) as the working fluid. Their prototype demonstrated effective heat recovery from hot gases at temperatures of approximately 200°C. However, DI water requires a complex and high-cost purification process, and due to the continuous operation of the present system (310 days per year), the working fluid must exhibit high thermal stability, low volatility, and minimal maintenance requirements under prolonged high-temperature conditions. For these reasons, heat-transfer oil was selected as the working fluid, due to its superior high-temperature flow characteristics and greater thermal storage capacity compared to DI water.

The flow circuitry and heat-transfer processes of the system are shown in Figure 3. A 0.5-hp pump (PT) serves as the driving mechanism. As the working fluid enters HE_a , it absorbs waste heat from

the BCS. The heated fluid is then delivered to the second exchanger, HE_b , where the thermal energy is transferred through the coil walls to a 10 L clean-water tank, raising the water temperature prior to supplying a 3,200 W coffee machine (COFF). After releasing its heat, the working fluid flows back into a 20 L oil reservoir tank (OT), thereby completing one full operational cycle of the heat-recovery system.

3.2 Helical Coil Design Methodology

The helical coil heat exchangers HE_a and HE_b used in this study were fabricated from copper tubes with an outer diameter of 10 mm and a wall thickness of 1 mm, formed into four helical turns with an elliptical cross-section, as shown in Figure 4. The cross-sectional geometry of the tube was evaluated using the percent ovality (δ) defined in Equation (1), while the hydraulic diameter (d) was calculated according to Equation (2). Additional physical specifications of the coils are presented in Table 1.

$$\delta = \frac{(\alpha - \beta)}{\varphi} \times 100 \quad (1)$$

$$\text{where } \varphi = \frac{\alpha + \beta}{2}$$

$$d = 4 \left[\frac{\frac{\pi \beta^2}{4} + \beta(\alpha - \beta)}{\pi \beta + 2(\alpha - \beta)} \right] \quad (2)$$

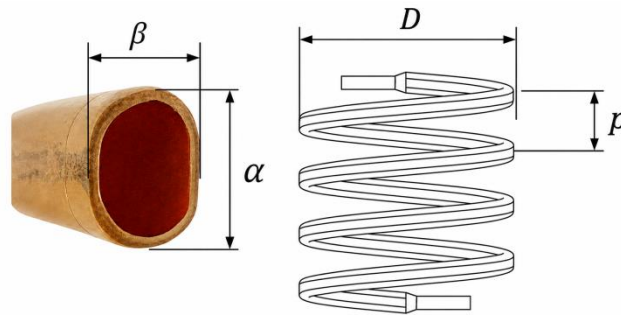


Figure 4 Geometric configuration of the elliptical-tube helical coil heat exchangers (H01–H04), highlighting differences in coil diameter, pitch, and tube ovality that govern the thermo-hydraulic behavior and heat-transfer performance

Table 1 Geometric design parameters of the four elliptical-tube helical coil heat exchanger configurations (H01–H04) assessed for biomass cookstove waste heat recovery

Model	d (m)	D (m)	d/D	p (m)	A_{coil} (m ²)	δ (%)	Node	Element
H01	0.01203	0.194	0.062	0.025	0.1172	33	671,078	1,594,003
H02	0.01203	0.194	0.062	0.050	0.1173	33	785,049	1,625,947
H03	0.01168	0.188	0.062	0.050	0.1154	50	786,721	1,859,269
H04	0.01111	0.141	0.079	0.050	0.0889	66	735,300	1,731,114

3.3 Formulation of Operating and Boundary Conditions

The evaluation of the Helical Coil Heat Exchanger (HHE) in this study is divided into two main components, as follows:

3.3.1 Thermal Performance Assessment of the HHE

The first stage involved selecting the most suitable HHE geometry from the four candidates through CFD simulations conducted under a constant surface temperature of 200°C and at mass flow rates of 0.5–2.5 L/min. The configuration exhibiting the highest heat-transfer performance was then subjected to an in-depth evaluation under operating conditions representative of the actual heat-recovery system. This advanced analysis considered the dual functional roles of the exchanger, operating as HE_a (absorbing waste heat from the BCS) and as HE_b (delivering heat to the water side).

For the HE_a configuration, CFD simulations were performed under surface temperatures of 160, 180, and 200°C across mass flow rates of 0.5–2.5 L/min.

For the HE_b configuration, the surface temperature was fixed at 30°C, with the inlet fluid temperature prescribed to be equal to the outlet temperature of HE_a , and no thermal losses along the flow path were considered.

The same coil configuration was employed for both HE_a and HE_b simulations to maintain consistent flow behavior and heat-transfer characteristics, thereby ensuring that the performance evaluation remained reliable and directly comparable. In addition, to capture flow phenomena ranging from laminar to transitional regimes while remaining aligned with the design objective of developing a compact, low complexity recovery system the working fluid flow rate was prescribed within the range of 0.5–2.5 L/min. The mass flow rate was also constrained to be identical throughout the system ($\dot{m}_{HE_a,i} = \dot{m}_{HE_b,i}$) to preserve hydraulic consistency.

3.3.2 Evaluation of Economic and Environmental Potential

Once the thermal performance of the HHE had been established, the overall feasibility of the system was assessed using net present value (NPV) and payback period (PP) as key economic indicators. Environmental performance was evaluated in parallel by quantifying the reduction in CO₂ emissions, thereby reflecting the energy benefits and long-term sustainability of the proposed heat-recovery system

within the operational context of the food-service industry.

3.4 Quantifying the Thermodynamic Properties of the System

The thermodynamic analysis of the heat-recovery system is based on the principles of mass and energy conservation to determine the rate of heat transfer and the mechanical interactions within each component of the system. The analysis is performed under steady-state conditions, assuming no thermal losses during the transport of the working fluid from HE_a to HE_b . The energy balance for each component, derived from the fundamental mass and energy balance equations, can be expressed as follows:

$$\dot{m}_{in,k} = \dot{m}_{out,k} \quad (3)$$

$$\dot{Q}_k - \dot{W}_k = \dot{m}c_p(T_{out,k} - T_{in,k}) \quad (4)$$

where $\dot{m}$, $\dot{Q}$ and $\dot{W}$ represent the mass-flow rate, heat-transfer rate, and work rate, respectively. These variables are applied in conjunction with the governing relationships expressed in Equations (3)–(5) to evaluate the mass and energy balances for each component of the system. The heat-transfer rates of HE_a and HE_b can be determined using Equation (5).

$$\dot{Q}_{HEk} = \dot{m}c_p(T_{o,k} - T_{i,k}) \quad (5)$$

The mean temperature difference (ΔT_m) is defined according to Equation (6), where T_s denotes the surface temperature of HE_a and HE_b , and T_b represents the bulk temperature of the working fluid, calculated using Equation (7). The evaluation of the convective heat-transfer coefficient follows the relationship given in Equation (8).

$$\Delta T_m = T_{s,HEk} - T_{b,HEk} \quad (6)$$

$$T_B = \frac{T_{in,HEk} + T_{out,HEk}}{2} \quad (7)$$

where $T_{s,HEk}$, $T_{b,HEk}$ denote the inlet and outlet temperatures of the working fluid for heat exchanger HE_k , respectively.

The calculation of the Nusselt number is performed according to the correlation given in Equation (8).

$$Nu_{HEk} = \frac{h_{HEk} d_{HEk}}{K} \quad (8)$$

where K is thermal conductivity (W/m·K).

For the CFD analysis, this study employed the $k - \omega$ SST (shear-stress transport) turbulence model (Lahamornchaiyakul, 2018), with the governing equations defined in Equations (9)–(19).

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i} \left((u + \sigma_k u_t) \frac{\partial k}{\partial x_i} \right) - \rho u_i' u_j' \frac{\partial u_i}{\partial x_j} - \rho \beta^* k \omega \quad (9)$$

$$\begin{aligned} \frac{\partial}{\partial x_i}(\rho \omega u_i) = & \frac{\partial}{\partial x_i} \left((u + \sigma_k u_t) \frac{\partial \omega}{\partial x_i} \right) - \frac{\alpha}{v_t} \rho u_i' u_j' \frac{\partial u_i}{\partial x_j} \\ & - \rho \beta \omega^2 + 2(1 - F_1) \rho \sigma_{\omega,2} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \end{aligned} \quad (10)$$

where the eddy viscosity is determined according to Equation (11).

$$\mu_t = \rho \frac{k}{\omega} \max \left[\frac{1}{\alpha^* a_1 \omega} \right] \quad (11)$$

$$\text{where } \Omega = \sqrt{\Omega_{ij} \Omega_{ij}} \quad (12)$$

$$\Omega_{ij} = \frac{1}{2} \left(\frac{\partial \mu_i}{\partial x_j} - \frac{\partial \mu_j}{\partial x_i} \right) \quad (13)$$

$$F_2 = \tanh \Phi_2^2 \quad (14)$$

$$\Phi_2 = \max \left[2 \frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho \omega y^2} \right] \quad (15)$$

where the constants β^* , σ_k , σ_ω are taken from the definitions provided in Equations (16)–(19).

$$\theta = F_1 \theta_1 + (1 - F_1) \theta_2 \quad (16)$$

$$F_1 = \tanh \Phi_1^4 \quad (17)$$

$$\Phi_2 = \min \left[\max \left(2 \frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho \omega y^2} \right) \left(\frac{4 \rho k}{\sigma_{\omega,2} D_{\omega}^2 y^2} \right) \right] \quad (18)$$

$$D_{\omega}^+ = \max \left[2 \rho \frac{1}{\omega \sigma_{\omega,2} \omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}; 10^{-20} \right] \quad (19)$$

The remaining constants are specified as follows:

$$\beta_1 = 0.075, \beta_2 = 0.0828, a_1 = 0.31, \alpha^* = 1$$

3.5 Economics Feasibility Assessment Tools

Although the enhancement of a heat-recovery system can substantially improve overall energy efficiency, the economic and environmental viability of such an installation remains a critical consideration for business operators. This is because the implementation of heat-recovery technology requires additional capital investment. The total investment cost is divided into two main

components: fixed cost (FC) and variable cost (VC). The fixed cost is calculated as the sum of the direct cost (DC), which includes equipment and installation expenses, and the indirect cost (IC), which covers design, supervision, and construction-related expenditures, as expressed in Equation (20).

$$FC = DC + IC \quad (20)$$

For the direct cost (DC), the principal variable is the purchased equipment cost (PEC), which represents the actual cost of the major components, namely the heat exchangers, pump, and control system. The PEC increases with system complexity in accordance with the engineering scaling factor. The indirect cost (IC), on the other hand, is calculated as a proportion of the DC and includes contingencies equal to 0.185 times the DC (Athari et al., 2022) to account for fluctuations in material and labor costs.

In addition to the capital investment, the operating cost of the heat-recovery system constitutes a critical component of the economic assessment. Using the energy cost balance approach, the total outgoing energy-related cost must be balanced against the incoming energy-related cost. This comprises equipment operation costs and maintenance expenses, as defined by Equation (21) (Nami et al., 2017).

$$C_{e,o,k} = C_{Q,k} + \sum_i C_{OP,i,k} + Z_k \quad (21)$$

Where i and o denote the directions of energy inflow and outflow, respectively, for component k within the system. To evaluate the specific energy cost associated with each component, the concept of cost per unit product energy ($\dot{C}_{p,k}$) is applied, as defined in Equation (22).

$$\dot{C}_{p,k} = \frac{C_{p,k}}{e_{p,k}} \quad (22)$$

The electrical energy cost ($C_{w,i}$) supplied to component i can be calculated using Equation (23).

$$C_{w,i} = P_i \times [C_E + Ft + \text{Vat}(7\%)] \quad (23)$$

$$\text{where } P_i = \eta_{h \rightarrow el} \times \frac{\dot{Q}_{HEk}}{1000} \times t$$

Where P_i (kWh) represents the annual electricity-equivalent energy associated with component i , derived from the recovered thermal energy. The conversion efficiency $\eta_{h \rightarrow el}$ denotes the

fraction of thermal energy that can be interpreted as equivalent electrical energy for economic evaluation purposes and is assumed to be unity in this study (Iqbal et al., 2023). The variable t represents the annual operating time (h/year). The unit electricity tariff is specified as $C_E = 4.30$ THB/kWh (0.12 USD/kWh), while Ft denotes the variable electricity tariff adjustment factor regulated in Thailand (Wongwut & Angamnaysiri, 2024). Based on the 2024 national average, Ft is specified as 0.3972 THB/kWh (0.011 USD/kWh), as reported by the Metropolitan Electricity Authority (MEA, 2025). The payback period (PP) can be evaluated using Equation (24).

$$PP = \frac{FC}{ANS} \quad (24)$$

Where FC represents the net investment cost after depreciation, and ANS denotes the annual net savings generated from system operation, calculated using Equation (25).

$$ANS = AI - C_{w,i} \quad (25)$$

Where AI represents the annual income generated from the thermal energy transferred by HE_b to the water in the hot-water tank (HT), expressed in terms of its equivalent electrical energy value. Finally, the net present value (NPV) indicator is employed to assess the economic feasibility of the investment. A positive NPV signifies that the investment is economically viable and yields returns exceeding the initial capital cost, whereas a negative NPV indicates that the expected benefits fall short of the investment requirements. The NPV can be calculated using Equation (26).

$$NPV = -(FC \times IF_0 \times RDF_0) + \sum_{i=1}^n (ANS \times IF_i \times RDF_i) \quad (26)$$

Where IF and RDF denote the inflation rate and the real discount factor, respectively, for year i within the system's operational lifetime. Both variables can be determined using Equations (27) and (28).

$$IF_i = \left(1 + \frac{R}{100}\right)^i \quad (27)$$

$$RDF_i = \left(1 + \frac{RIR}{100}\right)^{-i} \quad (28)$$

3.6 Assessment of Environmental Impacts

The CO₂ emission reduction is assessed based on the recovered thermal energy expressed as P_i

(kWh), which represents the energy associated with component i in the form of thermal energy transferred to the storage tank (TH). This energy is assumed to offset the equivalent electricity consumption of an electric heater, and the resulting CO₂ reduction is evaluated using Equation (29) with an emission factor of $v_f = 0.23029$ kgCO₂e/kWh (Battisti, 2023).

$$\varepsilon_{CO_2e} = v_f P_i \quad (29)$$

4. Results and Discussion

4.1 Model Validation

Since no mathematical model currently exists that explicitly characterizes the Nusselt number behavior for helical coils with an elliptical cross-section, this study validates the CFD results by comparing them with empirical correlations developed for circular-cross-section helical coils. The correlations employed for comparison are those proposed by Dravid et al. (1971), Pawar and Sunnapwar (2013), and Xin and Ebadian (1997).

For this purpose, the inlet working-fluid temperature was set to 200°C, and the thermophysical properties of the working fluid heat-transfer oil were determined using the Prandtl number (Pr), calculated according to Equation (30). This approach ensures a consistent and reliable basis for comparing the heat-transfer performance obtained from numerical simulations with that predicted by established empirical correlations.

$$Pr = \frac{\mu C_p}{k} \quad (30)$$

Where μ , C_p and k denote the dynamic viscosity (Pa·s), specific heat at constant pressure (kJ/kg·K), and thermal conductivity (W/m·K), respectively. These properties were assigned based on the thermophysical characteristics of the heat-transfer oil, with density, kinematic viscosity, specific heat, and thermal conductivity specified as 0.7561 g/cm³, 1.085 mm²/s, 2.469 kJ/kg·K, and 0.097 W/m·K, respectively.

The validation outcomes are presented in Figure 5(a–d), showing that the CFD-predicted Nusselt number variations for H01–H04 closely follow the experimental-based correlations of Dravid et al. (1971), Pawar & Sunnapwar (2013) and Xin & Ebadian (1997). The CFD results consistently fall between the correlations of Dravid and Pawar, confirming the expected thermal response as the Reynolds number increases. These findings are

further supported by the statistical indicators in Table 2, where the correlation of Dravid et al. (1971) provides the best agreement with the CFD predictions, with RMS errors of 1.02–11.36% and average absolute deviations of 0.19–7.43%. In contrast, the correlations between Xin & Ebadian (1997) and Pawar & Sunnapwar (2013) show

increasing overprediction at Reynolds numbers above 6,000.

These findings confirm that the CFD model employed in this study provides high accuracy and reliability for analyzing heat-transfer performance in helical coils with elliptical cross-sections under the conditions investigated.

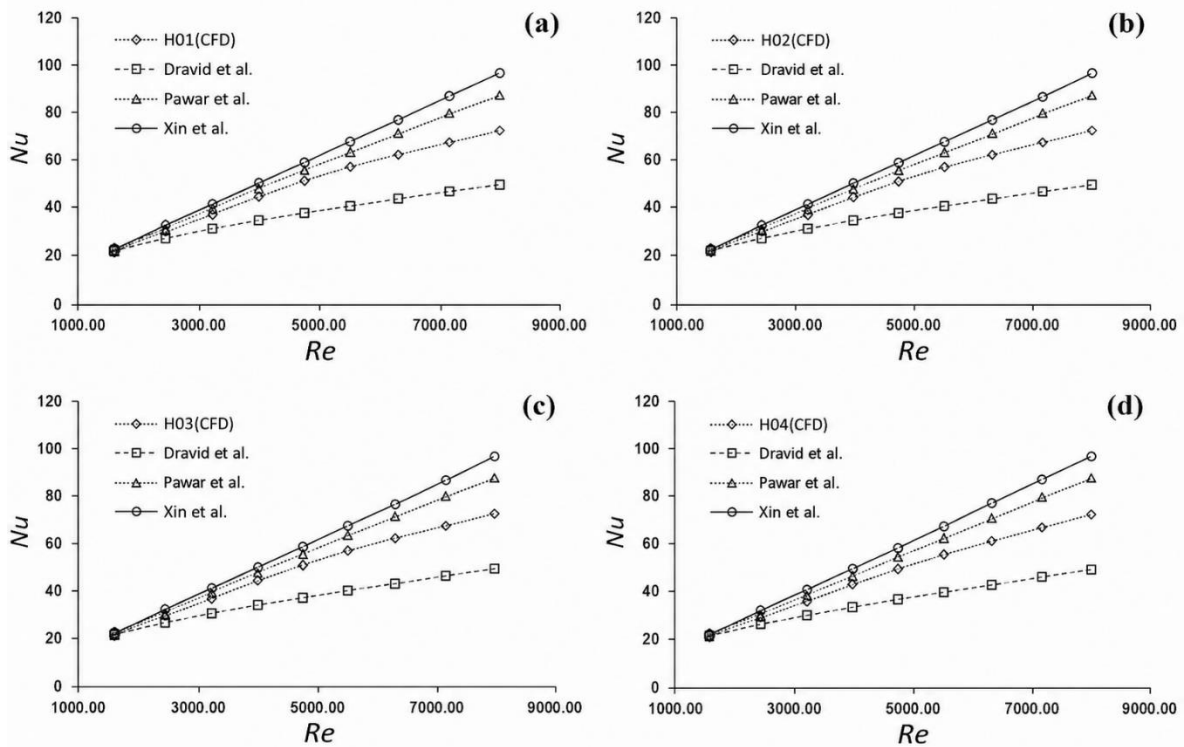


Figure 5 Comparison of CFD-predicted Nusselt number and empirical correlations for helical coil configurations H01–H04, validating the accuracy of the CFD thermal model (Dravid et al., 1971; Pawar & Sunnapwar, 2013; Xin & Ebadian, 1997)

Table 2 Statistical validation of CFD-predicted Nusselt number compared with empirical correlations for helical coils.

	Model	Dravid et al. (1971)	Pawar & Sunnapwar (2013)	Xin & Ebadian (1997)
RMS error	H01	5.19	7.23	1.10
	H02	1.84	2.94	9.14
	H03	1.02	3.16	9.58
	H04	11.36	6.99	4.28
Absolute average error	H01	16.75	10.43	7.75
	H02	17.17	9.47	6.81
	H03	17.97	7.88	4.99
	H04	10.36	14.68	16.51
Average Error	H01	−6.23	7.85	7.67
	H02	−7.04	7.32	7.14
	H03	−7.43	5.18	4.74
	H04	−0.19	12.91	17.37

Note. Empirical correlations are from Dravid et al. (1971), Pawar and Sunnapwar (2013), and Xin and Ebadian (1997).

RMS = root mean square.

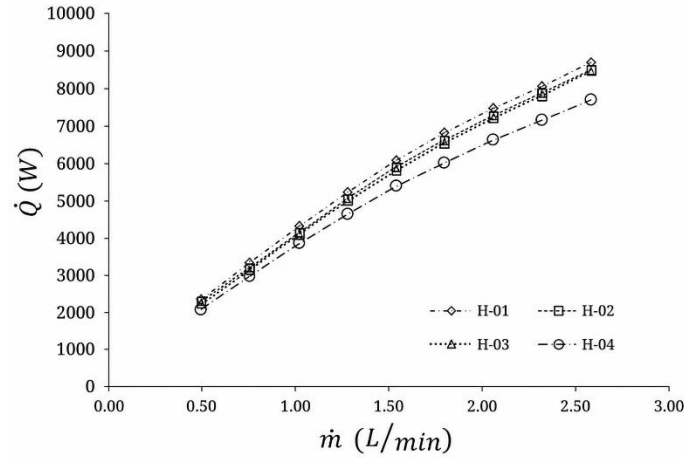


Figure 6 Heat-transfer rate as a function of working-fluid flow rate for H01–H04, demonstrating the influence of geometric variations on secondary-flow enhancement and thermal performance

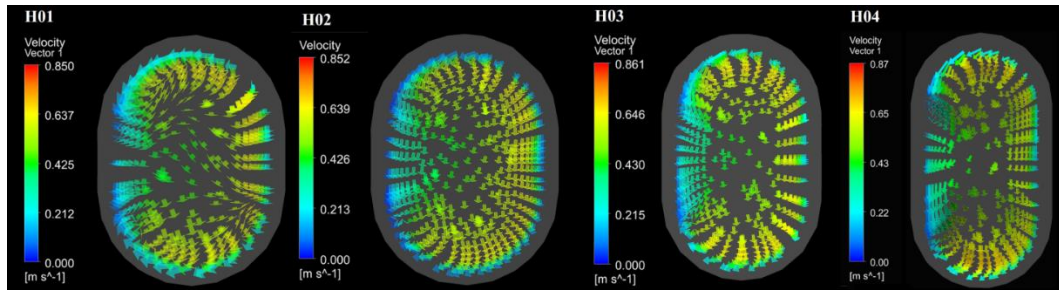


Figure 7 Cross-sectional velocity vector fields of H01–H04 at 180° from the inlet under a flow rate of 2.5 L/min and a coil temperature of 200°C.

4.2 Thermal Performance Analysis

4.2.1 Heat-Transfer Response to Physical Modifications of the HHE

Figure 6 illustrates the relationship between the heat-transfer rate ($\dot{Q}$) and the mass flow rate ($\dot{m}$) for models H01–H04. The results show that $\dot{Q}$ increases monotonically with increasing flow rate across all configurations; however, distinct performance differences arise from the geometric characteristics of each coil, as detailed below.

H01 and H02 are the two configurations with the most closely matched geometrical characteristics, sharing identical tube diameter (d), coil diameter (D), d/D ratio, and cross-sectional ovality (δ). The only difference between these two models lies in the pitch length, where H01 employs a shorter pitch (0.025 m) while H02 adopts a longer pitch (0.05 m). Although the increased pitch in H02 leads to some modification in the swirling flow pattern, as illustrated in Figure 7, the effective heat-transfer surface area of the two configurations remains nearly identical. Consequently, their heat-transfer performances are very similar over the entire range

of flow rates, with H01 exhibiting only a 2.78% higher heat-transfer rate than H02. This result indicates that altering the pitch alone is insufficient to induce a significant impact of secondary flow on the overall heat-transfer performance.

A comparison between H01 and H03 shows that the heat transfer performance of H03 is only 4.10% lower than that of H01. Although H03 features a smaller tube diameter (d) and coil diameter (D), it possesses a higher cross-sectional ovality ($\delta = 50\%$), which enhances the development of secondary flow in the near-wall region, as shown in Figure 7. However, since both models share the same d/D ratio, the modification of the secondary flow structure induced solely by increased ovality is not sufficient to cause a significant change in the overall heat-transfer performance of the coil.

In contrast, the comparison between H01 and H04 reveals the most pronounced difference among all cases. H04 exhibits the highest ovality ($\delta = 66\%$), combined with a smaller tube diameter ($d = 0.0111$ m) and the narrowest coil diameter ($D = 0.141$ m). The simultaneous reduction in D and increase in δ

results in counter-rotating vortices that occupy a larger fraction of the cross-sectional area and produce more continuous transverse flow compared with those in H01, as illustrated in Figure 7. This behavior indicates a clear modification of the radial secondary flow structure. Nevertheless, the highly oval tube cross-section in H04 promotes heat accumulation near the wall, and certain flow regions exhibit quasi-laminar characteristics. As a result, the overall heat-transfer performance of H04 is reduced, yielding a heat-transfer rate 12.83% lower than that of H01, which represents the largest deviation among all pairwise comparisons.

Based on the combined visualization results and geometric analysis, H01 demonstrates the most balanced development of secondary flow without being constrained by an increased pitch length (H02), localized ovality enhancement (H03), or an excessively narrow coil radius (H04). This behavior is consistent with the findings reported by Pourhemmati et al. (2024) and Zhang et al. (2024), who identified secondary flow as the primary mechanism governing convective heat-transfer enhancement in helical coils and highlighted its strong sensitivity to coupled geometric parameters rather than isolated changes in secondary design variables. The combined effect of these mechanisms enables H01 to consistently exhibit the highest heat-transfer performance across all flow rates, making it the most suitable configuration for application in waste-heat recovery systems.

4.2.2 Heat-Transfer Performance under Varying Surface Temperatures of The Heat-Absorbing Section (HE_a)

The temperature contour distributions of the optimal HHE configuration (H01) under different wall surface temperatures and flow rates, as presented in Table 3, provide qualitative insight into the heat-transfer behavior and flow development within the helical coil. To ensure consistency with the visualization-based nature of the CFD results, the thermal characteristics are discussed separately for each wall surface temperature of the heat-absorbing section (T_{coil, HE_a}).

1. At $T_{\text{coil}, HE_a} = 160^\circ\text{C}$

At a wall surface temperature of 160°C , the temperature contours are predominantly characterized by green and blue regions across all investigated flow rates. At the lowest flow rate ($\dot{m} = 0.5 \text{ L/min}$), noticeable temperature gradients are

observed along the coil, indicating relatively limited convective mixing and weaker thermal interaction between the wall and the working fluid. As the flow rate increases to 1.5 L/min , the temperature distribution becomes more uniform, with a gradual reduction in localized low-temperature regions. At $\dot{m} = 2.5 \text{ L/min}$, the contours exhibit the highest degree of uniformity for this temperature level, indicating enhanced convective mixing and improved thermal interaction induced by the helical geometry.

2. At $T_{\text{coil}, HE_a} = 180^\circ\text{C}$

When the wall surface temperature increases to 180°C , providing a larger temperature driving force, the overall temperature contour levels within the coil shift toward green-to-yellow regions across all flow conditions. At $\dot{m} = 0.5 \text{ L/min}$, temperature non-uniformity remains evident, although the extent of low-temperature regions is reduced compared with the 160°C case. At the intermediate flow rate of 1.5 L/min , the temperature field becomes more evenly distributed along the coil, exhibiting smooth contour transitions without pronounced hot or cold zones. At $\dot{m} = 2.5 \text{ L/min}$, the temperature contours are highly uniform, indicating that convective transport increasingly governs the thermal behavior under this operating condition.

3. At $T_{\text{coil}, HE_a} = 200^\circ\text{C}$

At the highest wall surface temperature of 200°C , the temperature contours exhibit a clear upward shift, with yellow regions occupying a larger portion of the coil. At the lowest flow rate, relatively strong temperature gradients are still observed, particularly near the inlet region. However, as the flow rate increases to $1.5\text{--}2.5 \text{ L/min}$, the temperature distribution becomes progressively more uniform along the entire coil length. Notably, the contrast in temperature patterns between different wall surface temperatures diminishes at the highest flow rate, indicating that enhanced convective mixing reduces the sensitivity of the internal temperature field to further increases in wall surface temperature.

Overall, the temperature contours in Table 3 consistently demonstrate that increasing the flow rate enhances the uniformity of the temperature distribution for all investigated wall surface temperatures. The absence of pronounced localized hot or cold regions under moderate-to-high flow conditions suggests that the geometric features of the H01 configuration specifically the pitch, d/D ratio, and tube ovality are conducive to secondary-flow-

induced mixing characteristic of helical coils. These qualitative observations provide a physically consistent basis for the subsequent quantitative

evaluation of heat-transfer performance and energy recovery.

Table 3 Temperature contour distribution of the optimal HHE configuration (H01) under different coil-surface temperatures (160–200°C) and flow rates (0.5–2.5 L/min)

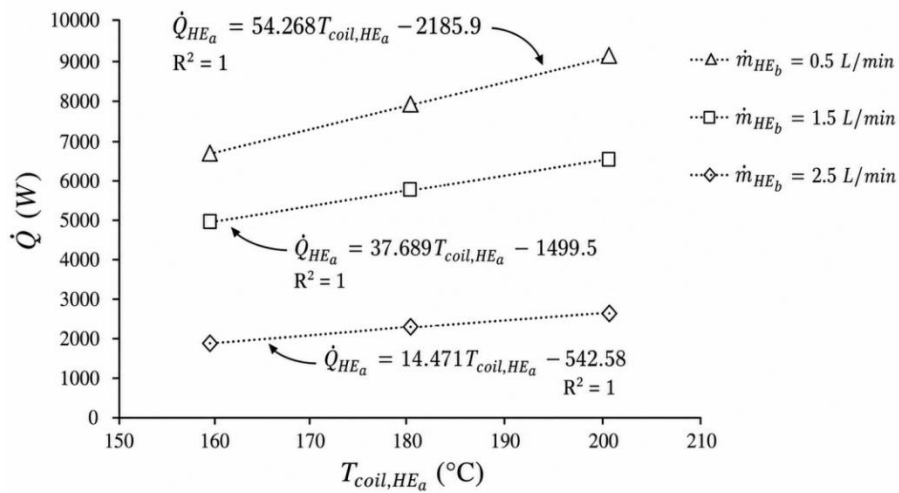
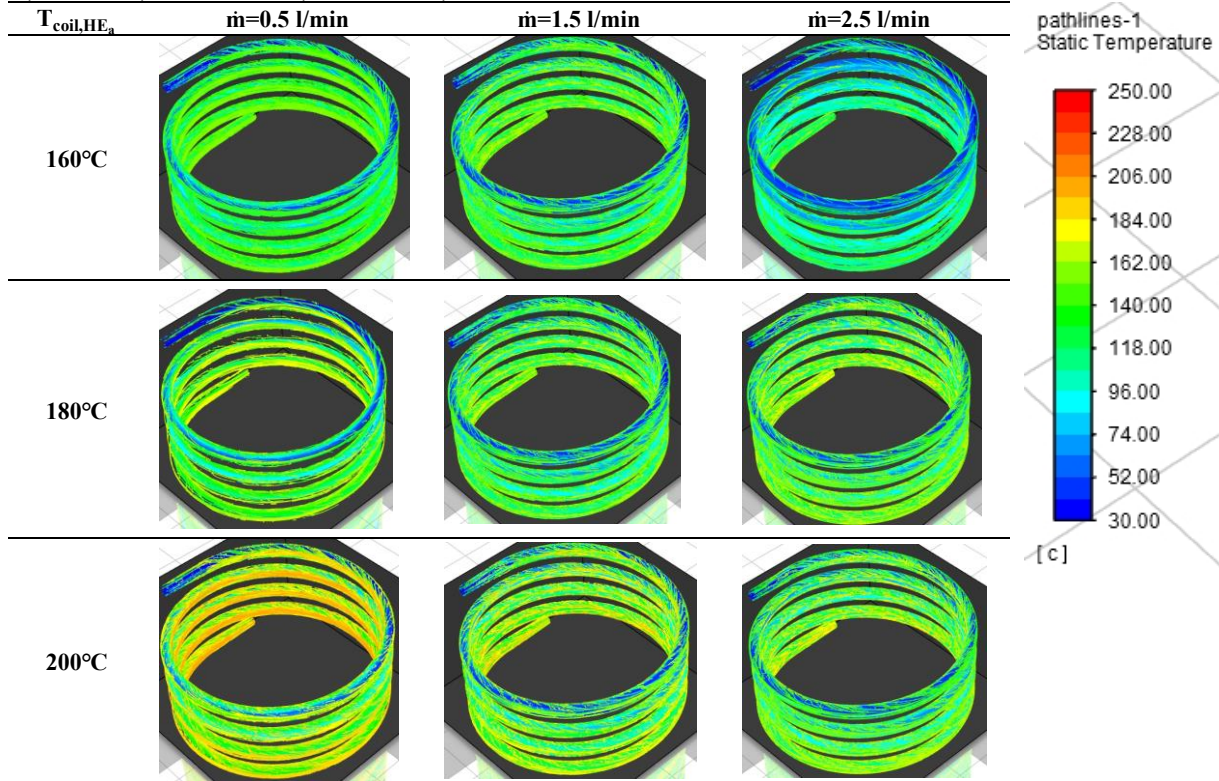


Figure 8 Variation of heat-transfer rate of HE_a with coil surface temperature under different flow rates, showing linear performance enhancement due to increased convective heat-transfer

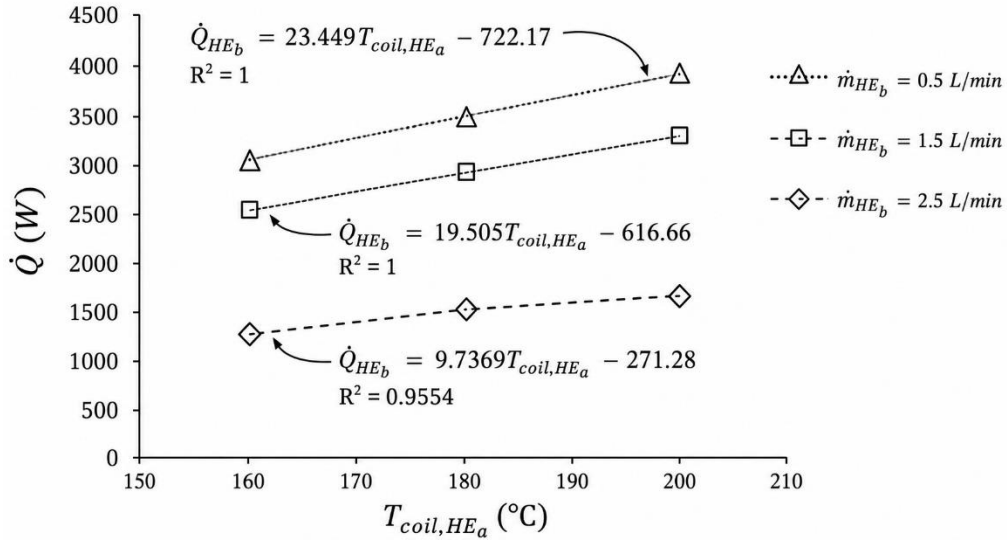


Figure 9 Heat-transfer delivery from HE_b to water storage as a function of coil surface temperature and flow rate, indicating the system's capability to generate hot water for practical food-service applications

4.3 The Role of The Helical Coil in The Heat-Recovery System

As shown in Figure 8, H01, functioning as HE_a , exhibits a strictly linear heat-transfer response to the coil surface temperature ($T_{coil, HEa}$) across all flow rates (0.5, 1.5, and 2.5 L/min). All trend lines yield $R^2=1$, indicating a perfect linear correlation between $T_{coil, HEa}$ and $\dot{Q}_{HEa}$. The rates of change in $\dot{Q}_{HEa}$ with respect to $T_{coil, HEa}$ at flow rates of 0.5, 1.5, and 2.5 L/min are 14.47, 37.69, and 54.27 W/°C, respectively. The highest rate occurs at 2.5 L/min, reflecting enhanced convective performance under elevated flow conditions due to a reduced film temperature near the wall and more effective heat transport into the working fluid.

Furthermore, at a flow rate of 2.5 L/min and a surface temperature of 200°C, H01 achieves a heat-recovery output in the range of 8,000–9,000 W, which is sufficient for applications such as hot-water generation or preheating processes in small-scale food service establishments.

As shown in Figure 9, when H01 operates as HE_b , receiving thermal energy from the high-temperature working fluid exiting HE_a and subsequently transferring this energy to the water in the storage tank, $\dot{Q}_{HEb}$ increases consistently with $T_{coil, HEa}$ at all flow rates. Although perfect linearity is

observed only at 1.5 and 2.5 L/min ($R^2=1$), the trend for 0.5 L/min despite having $R^2=0.9554$ still exhibits a clear and continuous rise.

The rates of change of $\dot{Q}_{HEb}$ with respect to $T_{coil, HEa}$ at flow rates of 0.5, 1.5, and 2.5 L/min are 9.74, 19.50, and 23.45 W/°C, respectively. These values indicate that increasing the flow rate on the HE_b side enhances the capability of the coil to release thermal energy efficiently and consistently. At the flow rate of 2.5 L/min, with $T_{coil, HEa}=200$ °C, the system attains its highest heat-transfer performance.

To support system designers, the relationships between heat-transfer performance and flow rates for both HE_a and HE_b were formulated in the form of linear predictive models. Based on the correlations between the coil surface temperature (T_{coil}) and $\dot{Q}$ in Figure 8, and between $T_{coil, HEa}$ and $\dot{Q}_{HEb}$ in Figure 9, the general form of the linear relationships can be expressed as follows:

Based on the results presented in Figures 8 and 9, This study established heat-transfer correlations at different flow rates for predicting $\dot{Q}_{HEa}$ and $\dot{Q}_{HEb}$. These correlations were formulated using the linear relationships defined in Equations (31)–(32)

$$\text{Case of } HE_a: \dot{Q}_{HEa}(\dot{m}, T_{coil, HEa}) = a_1(\dot{m}_{HEa})T_{coil, HEa} + b_1(\dot{m}_{HEa}) \quad (31)$$

$$\text{Case of } HE_b: \dot{Q}_{HEb}(\dot{m}_{HEb}, T_{coil, HEa}) = a_2(\dot{m}_{HEb})T_{coil, HEa} + b_2(\dot{m}_{HEb}) \quad (32)$$

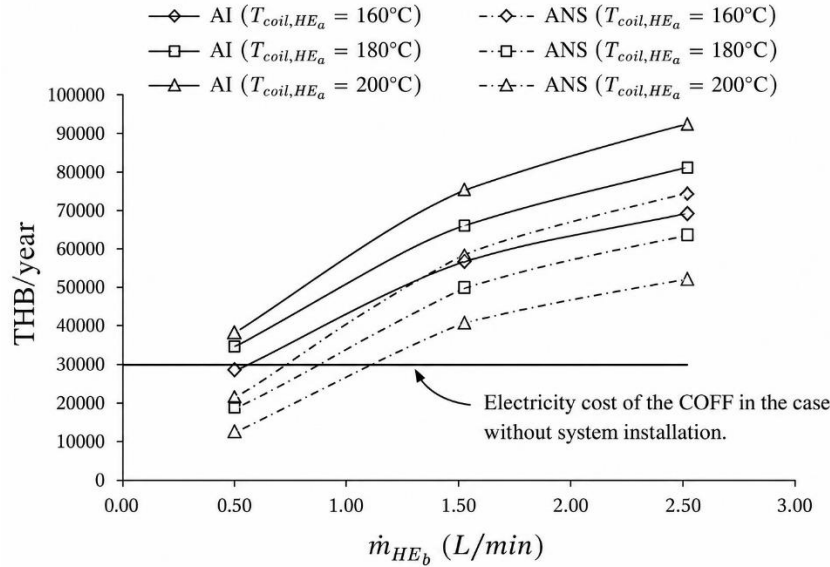


Figure 10 Annual income (AI) and annual net savings (ANS) achieved by the WHR system under different thermal operating conditions, compared with operation without heat recovery

Table 4 Evaluation of electricity-equivalent power, annual income, and net cash flow of the WHR System under Different Operating Conditions

T_{coil,HE_a} (°C)	$\dot{m}_{HE_b}$ (L/min)	P_i (kWh)	AI (THB/year)	ANS (THB/year)
160	0.50	5,453.24	27,641.40	12,162.40
	1.50	10,820.99	54,849.47	39,370.46
	2.50	13,092.34	66,362.50	50,883.49
180	0.50	6,609.41	33,501.81	18,022.80
	1.50	12,497.85	63,349.18	47,870.17
	2.50	15,105.94	76,569.04	61,090.03
200	0.50	7,135.78	36,169.86	20,690.85
	1.50	14,191.53	71,934.07	56,455.07
	2.50	17,144.34	86,901.28	71,422.27

4.4 Economic Analysis of the System

The coffee machine employed in the system has a rated power of 3,500 W and requires a total annual energy consumption of 1,289.92 kWh, resulting in an electricity cost of 27,982.06 THB per year based on 310 operating days. Without the heat-recovery system, this cost must be fully borne by the operator. An evaluation of the heat-recovery potential using H01 as both HE_a and HE_b indicates that the coil surface temperature (T_{coil,HE_a}) and the working-fluid flow rate ($\dot{m}$) exert significant influence on the economic performance. Both the annual income (AI) and annual net savings (ANS) increase consistently with higher T_{coil,HE_a} and $\dot{m}$.

As shown in Figure 10 and Table 4, at $T_{coil,HE_a} = 160^\circ\text{C}$ and $\dot{m}_{HE_b} = 0.5$ L/min, the annual net savings (ANS) are 12,162.40 THB/year. Increasing the flow rate to 1.5 L/min raises ANS to

39,370.46 THB/year (a 223% increase) and further increases it to 50,883.49 THB/year at 2.5 L/min (a total increase of 318% from the lowest condition). A similar trend is observed at $T_{coil,HE_a} = 180^\circ\text{C}$, where ANS rises from 18,022.80 THB/year at 0.5 L/min to 47,870.17 and 61,090.03 THB/year at 1.5 and 2.5 L/min, representing increases of 166% and 239%, respectively.

At the highest surface temperature of $T_{coil,HE_a} = 200^\circ\text{C}$ and $\dot{m} = 2.5$ L/min, ANS reaches 71,422.27 THB/year, which exceeds the lowest case within the same temperature group (20,690.85 THB/year at 0.5 L/min) by 245%, and surpasses the minimum value across all conditions (12,162.40 THB/year at 160°C and 0.5 L/min) by 487%. The annual income (AI) exhibits a consistent pattern: at $T_{coil,HE_a} = 200^\circ\text{C}$, AI increases from 36,169.86 THB/year at 0.5 L/min to 71,934.07 THB/year at 1.5

L/min, reaching a maximum of 81,690.28 THB/year at 2.5 L/min an increase of 126% from the lowest condition within the same group.

As shown in Figure 11, the net present value (NPV) analysis of the heat-recovery system was performed assuming a fixed capital cost (FC) of 96,869.84 THB. The analysis adopted an inflation rate (R) of 1.23% and a real interest rate (RIR) of 7.57%, based on the annual average values for 2024 reported in the Bank of Thailand Annual Report 2024 (Bank of Thailand, 2024). The results demonstrate a clear relationship between the coil surface temperature ($T_{\text{coil, HE}_a}$) and the working-fluid flow rate ($\dot{m}_{\text{HE}_b}$) in determining the financial performance of the system, as reflected in both annual net savings (ANS) and net present value (NPV). Increasing $T_{\text{coil, HE}_a}$ and $\dot{m}_{\text{HE}_b}$ enhances energy-recovery potential and significantly improves the overall economic viability of the system.

Under the low-temperature condition of $T_{\text{coil, HE}_a} = 160^\circ\text{C}$, a flow rate of $\dot{m}_{\text{HE}_b} = 0.5$ L/min yields the lowest ANS (12,162 THB/year) and results in a negative NPV (−35,436.91 THB), indicating that the system cannot achieve payback under this operating state. However, increasing the flow rate to 1.5 L/min shifts the NPV to a positive value of 101,992.51 THB, demonstrating a marked improvement in economic performance. Although this condition does not represent the optimal

operating state, it signifies the threshold at which the system becomes financially feasible.

When $T_{\text{coil, HE}_a}$ increases to an intermediate level of 180°C , the condition at $\dot{m}_{\text{HE}_b} = 0.5$ L/min yields an annual net savings (ANS) of 18,022.80 THB/year, resulting in a negative NPV (−5,835.69 THB), similar to the low-temperature condition. However, increasing the flow rate to 2.5 L/min elevates ANS to 47,870.17 THB/year and raises the NPV to 211,699.27 THB, which is more than ten times greater than that of the non-viable case at 160°C and $\dot{m}_{\text{HE}_b} = 0.5$ L/min. These results indicate that increasing the thermal input from HE_a has a substantial influence on the net financial gains achieved by HE_b .

The most economically favorable operating condition occurs at $T_{\text{coil, HE}_a} = 200^\circ\text{C}$, particularly at $\dot{m}_{\text{HE}_b} = 2.5$ L/min, which yields an annual net savings (ANS) of 71,422.27 THB/year and the highest NPV recorded in the analysis, at 263,887.97 THB. This value exceeds the lowest-performing condition ($T_{\text{coil, HE}_a} = 160^\circ\text{C}$, $\dot{m}_{\text{HE}_b} = 0.5$ L/min) by 299,324.88 THB, corresponding to an increase of 487%. These results confirm that operation under high coil-surface temperature combined with an appropriate flow rate substantially enhances the recoverable thermal energy and significantly improves the overall economic viability of the system.

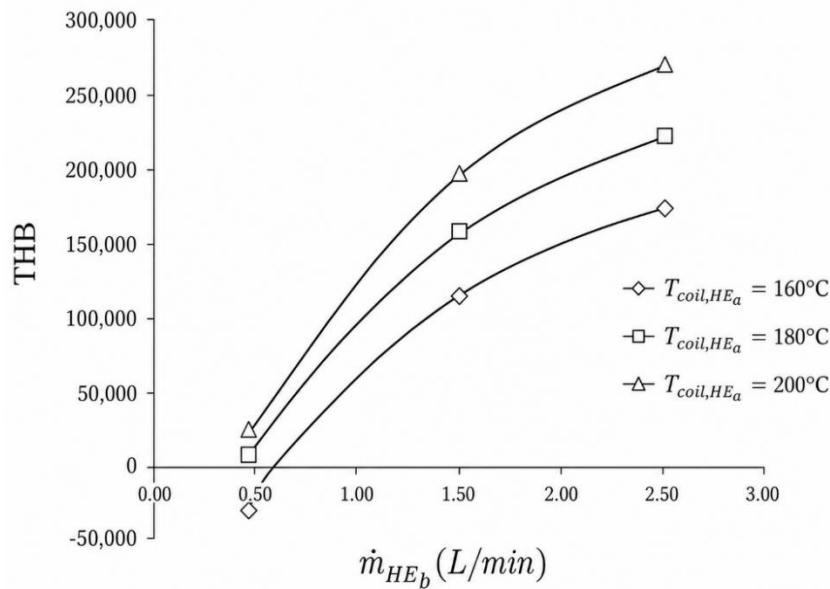


Figure 11 Net present value (NPV) of the WHR system at various coil surface temperatures and flow rates, identifying conditions under which the technology becomes financially viable

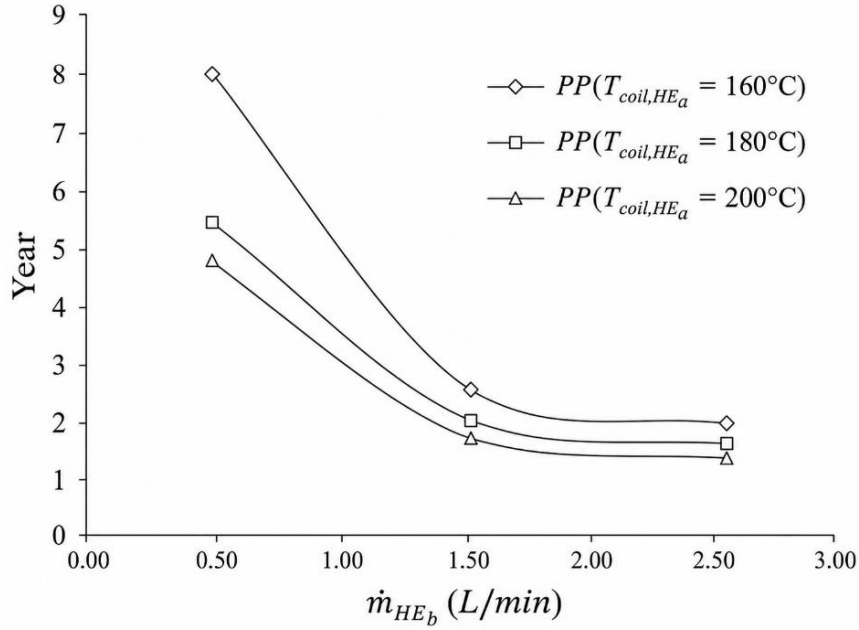


Figure 12 Payback period (PP) of the WHR system under different operating scenarios, confirming accelerated capital recovery when both flow rate and coil temperature are increased

The Payback Period (PP) analysis presented in Figure 12 demonstrates that the coil-surface temperature (T_{coil,HE_a}) and the working-fluid flow rate ($\dot{m}_{HE_b}$) are the key parameters governing the capital recovery performance of the heat recovery system. The PP decreases markedly with increasing $\dot{m}_{HE_b}$ and T_{coil,HE_a} , consistent with the corresponding rise in annual net savings (ANS) discussed in the preceding section.

Under conditions of $T_{coil,HE_a} = 160^\circ\text{C}$, the PP attains its highest value at $\dot{m}_{HE_b} = 0.5$ L/min (7.96 years). Increasing the flow rate to 1.5 and 2.5 L/min reduces the PP to 2.46 and 1.90 years, respectively, corresponding to a reduction of approximately 76%. At $T_{coil,HE_a} = 180^\circ\text{C}$, the PP decreases from 5.37 years at $\dot{m}_{HE_b} = 0.5$ L/min to 1.59 years at $\dot{m}_{HE_b} = 2.5$

L/min, a reduction of 70%. Under the highest temperature condition of $T_{coil,HE_a} = 200^\circ\text{C}$, the system exhibits its best performance, with the PP decreasing from 4.68 years at $\dot{m}_{HE_b} = 0.5$ L/min to 1.36 years at $\dot{m}_{HE_b} = 2.5$ L/min. This represents a reduction of approximately 71% and yields the lowest PP observed, which is 6.6 years shorter than the least favorable condition (160°C , 0.5 L/min), representing a reduction of more than 83%.

Therefore, it can be concluded that simultaneous increases in the inlet temperature of the working fluid and the flow rate significantly shorten the payback period of the system. This finding is crucial for the thermal and economic design of heat-recovery systems to ensure optimal cost-effectiveness.

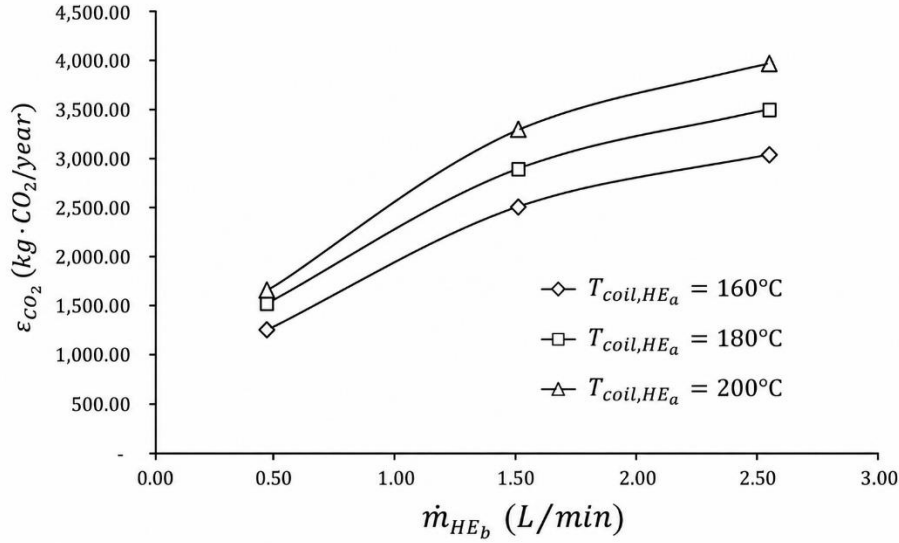


Figure 13 Reduction in CO₂ emissions resulting from displacement of electricity consumption by recovered waste heat, highlighting the strong environmental value of the system food-service applications

4.5 Environmental Impact Assessment

The results presented in Figure 13 indicate that the reduction in carbon dioxide emissions (ϵ_{CO_2e}) exhibits a direct relationship with both the working-fluid flow rate ($\dot{m}$) and the coil-surface temperature of HE_a ($T_{coil,HEa}$). As $\dot{m}$ increases from 0.5 L/min to 1.5 and 2.5 L/min, ϵ_{CO_2e} rises consistently across all tested temperatures, reflecting the corresponding increase in net recoverable heat ($\dot{Q}_{HEb}$) that the HHE can extract to replace the electric heater, driven by enhanced convective heat-transfer.

At $T_{coil,HEa} = 160^\circ\text{C}$, ϵ_{CO_2e} is 1,255.83 kg CO₂/year at $\dot{m}_{HEb} = 0.5$ L/min and increases to 2,491.97 and 3,015.03 kg·CO₂/year at $\dot{m}_{HEb} = 1.5$ and 2.5 L/min, respectively. When the coil surface temperature is elevated to 180°C and 200°C, the corresponding curves shift upward distinctly. At $T_{coil,HEa} = 200^\circ\text{C}$ and $\dot{m} = 2.5$ L/min, the system achieves an ϵ_{CO_2e} of 3,948.17 kg CO₂/year, exceeding the lowest-performing condition (160°C, 0.5 L/min) by more than a factor of three. These findings demonstrate that the system can substantially reduce CO₂ emissions, reinforcing its relevance to clean-energy objectives and carbon-reduction strategies for small-scale industrial applications.

Although the proposed H0 1 configuration demonstrates strong thermal and economic outcomes, certain implementation challenges remain. Elliptical-tube designs increase manufacturing complexity and may introduce a higher pressure

drop, requiring appropriate pump selection to maintain circulation. Safe handling of heat-transfer oil at 160–200°C also requires thermal insulation, routing protection, and operator training. Moreover, the current prototype was evaluated under a medium-sized restaurant scenario; scalability to other BCS geometries or smaller establishments may require redesign of coil dimensions and pumping power.

5. Conclusions

This study demonstrates that the elliptical-tube helical coil heat exchanger (HHE) is a highly effective solution for waste-heat recovery from biomass cookstoves used in small food-service enterprises. The key conclusions are as follows:

1. Thermal performance: The H01 configuration consistently provides the highest heat-transfer enhancement, outperforming H02, H03, and H04 by 2.78%, 4.10%, and 12.83%, respectively.
2. Operating conditions: Increasing the coil-surface temperature from 160–200°C boosts the heat-transfer rate by up to 33.40% across all tested flow rates, confirming strong dependence on temperature-driven secondary-flow development.
3. Energy benefit: Deploying H01 as both HE_a and HE_b enables the annual recovery of 17,144.34 kWh of useful heat, effectively offsetting electric heating demand in restaurant operations.
4. Economic feasibility: Cost analysis indicates a substantial annual energy-cost reduction

of 71,422.27 THB, with an attractive payback period of 1.36 years and an NPV of 263,887.97 THB.

5. Environmental impact: The system reduces CO₂ emissions by 3,948.17 kg/year, supporting practical decarbonization in small-scale commercial kitchens.

While the H01 configuration shows robust thermo-economic and environmental advantages, practical adoption may require consideration of pressure-drop-related pumping power and layout constraints when retrofitting existing cooking facilities. Future work will involve prototype deployment, measurements under dynamic restaurant operations, and long-term performance evaluation to verify actual implementation safety and reliability. Additional improvements in coil design and materials may further enhance sustainability and operational efficiency.

6. Abbreviations

Abbreviation	Full Term
A	Heat-transfer area (m^2)
ANS	Annual net cash flow from system operation (THB)
AI	Annual income generated from recovered energy (THB)
BCS	Biomass cookstove
C	Cost (THB)
CFD	Computational Fluid Dynamics
$C_{w,i}$	Electrical energy cost (THB)
d	Hydraulic diameter
D	Helical coil diameter
DC	Direct cost (THB)
Dn	Dean number
FC	Fixed capital investment (THB)
Ft	Fuel-adjustment charge (electricity)
h	Convective heat-transfer coefficient
HHE	Helical Coil Heat Exchanger
HE _a	Helical coil (heat absorption section)
HE _b	Helical coil (heat release section)
IC	Indirect cost
IF	Inflation rate
K	Thermal conductivity ($W/m K$)
m	Mass flow rate (L/min)
NPV	Net Present Value
Nu	Nusselt number
P	Electrical power (kWh)
p	Pitch distance (m)
PP	Payback period (Year)
PEC	Purchased Equipment Cost
Pr	Prandtl number

Abbreviation	Full Term
Q	Heat-transfer rate (W)
RIR	Real interest rate
RDF	Real discount factor
RMS	Root mean square error
Re	Reynolds number
T	Temperature ($^{\circ}C$)
ΔT_m	Arithmetic mean temperature difference ($^{\circ}C$)
VC	Variable cost (THB)
W	Rate of mechanical work
δ	Percent ovality
ϵ_{CO_2e}	Carbon dioxide-equivalent emission reduction ($kgCO_2/year$)
B	Bulk
HE	Helical coil
i	Inlet
o	Outlet
k	Component index
s	Surface

7. CRediT Statement

Suriya Sukarin: Conceptualization, Writing – Original Draft, Data Curation, Writing – Review & Editing, Software, Statistical Analysis, Formal Analysis, Methodology, Writing – Review & Editing.

Manusak Janthong: Conceptualization, Supervision, Validation, Project Administration.

Sathaporn Thongwik: Conceptualization, Supervision, Data Curation, Formal Analysis, Validation, Project Administration.

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