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# Performance Testing of a Shell-and-Tube Heat Exchanger with Phase Change Material for Hot Air Generator Applications

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***Abstract –*** *This study investigates the performance testing of a shell-and-tube heat exchanger designed for thermal energy storage using a phase change material (PCM), specifically sodium hydroxide. The heat exchanger used biomass-derived heat from a gasification burner, with hot air flowing within the shell and cold air passing through the tubes in a cross-flow arrangement. PCM was positioned beneath the heat exchanger cover to store excess heat during the heating process and extend the heat exchange duration during cooling. The system was tested at three cold air flow speeds (1.717, 2.965, and 5.490 m/s), while maintaining a constant hot air flow of 0.663 m/s. Results demonstrated that the PCM-enhanced heat exchanger effectively retained heat, keeping the cold air temperature above 60°C for up to 70 minutes at 1.717 m/s, compared to 40 minutes without PCM. These results suggest that incorporating PCM considerably improves heat retention and prolongs heat exchange duration.*

***Keywords*** – gasification burner, hot air generator, phase change material, shell-and-tube heat exchanger, thermal energy storage.

## 1. INTRODUCTION

Thermal Energy Storage (TES) is a critical technology for improving energy efficiency and supporting the use of renewable energy sources, particularly in systems with intermittent or fluctuating heat production such as solar or biomass energy. Storing excess thermal energy during surplus periods allows for its later use during shortages or peak demand. The application of Phase Change Materials (PCMs) has gained widespread interest as a medium for latent heat storage, owing to their ability to absorb and release large amounts of heat at nearly constant temperatures, thereby minimizing thermal fluctuations. Incorporating PCMs into thermal storage systems enhances energy storage density and maintains stable output temperatures compared to sensible heat storage materials [1], [2]. Additionally, during phase transitions, most of the heat is stored as latent heat, reducing thermal losses to the environment. For instance, Nomura *et al.* (2010) [3] reported that latent heat storage systems using high-temperature PCMs can store up to 2.75 times more energy than sensible heat systems, particularly in the context of industrial waste heat recovery.

For high-temperature applications (>300°C), sodium hydroxide (NaOH) is a notable inorganic PCM due to its high melting point (~318°C) and high latent heat of fusion. This makes it suitable for storing thermal energy from industrial processes or high-temperature renewable sources [4]. Previous studies, such as that by Nomura *et al.* (2010) [3], demonstrated the effective use of NaOH as a PCM in waste heat transport systems, while Kumar *et al.* (2021) [5] confirmed NaOH's capability for consistent heat absorption and release under constant heat flux in solar thermal applications. The selection of a PCM with a phase change temperature aligned with the target application is crucial, and NaOH offers a suitable melting range for industrial and high-grade thermal uses, along with relatively low cost compared to other PCMs. Recent works further underline NaOH's prominence for high-temperature TES. Martínez *et al.* (2025) [6] have extensively characterized NaOH and other PCMs in the 270–400 °C range, providing detailed thermophysical data and noting variations in thermal stability and material compatibility. Moreover, Jurczyk *et al.* (2024) [7] highlighted that although NaOH is affordable and abundantly available, its highly corrosive and hygroscopic nature requires careful encapsulation and moisture control in practical storage systems.

Regarding heat exchanger design for TES systems, numerous studies have focused on shell-and-tube configurations containing PCMs to enhance heat transfer and storage performance. For example, Hosseini *et al.* (2014) [8] experimentally and numerically investigated a shell-and-tube heat exchanger using paraffin as a PCM and found that heat transfer during melting and solidification is significantly affected by temperature and flow rate of the working fluid. Enhancements such as fins or internal geometry modifications can accelerate the heat transfer rate. Comparative studies have also assessed configurations like heat pipes versus shell-and-tube systems, demonstrating that design differences influence energy charge/discharge efficiency [9]. Nonetheless, shell-and-tube systems are widely adopted due to their simple construction, large heat transfer area, and flexibility in PCM integration. For these reasons, shell-and-tube units remain a focal point in recent TES research. Fan *et al.* (2025) [10] noted that the shell-and-tube design's simplicity, high PCM hold-up, and low thermal losses continue to make it a preferred

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configuration for latent heat storage. Furthermore, Wołoszyn and Szopa (2024) [11] demonstrated that employing a periodic multi-tube shell structure can dramatically speed up PCM melting and solidification, cutting the phase change duration substantially compared to a conventional single-tube design.

In the biomass energy context, capturing thermal energy from combustion or gasification processes as latent heat offers a promising strategy. Biomass gasifiers using feedstocks like rice husks or straw can generate hot gases at several hundred degrees Celsius for continuous thermal processes [12]. However, inconsistent heat supply or batch operation can lead to thermal waste if no storage is implemented. Latent heat storage using PCMs can capture excess heat during combustion and release it later to sustain operations during fuel supply interruptions or flameout periods. Accordingly, this study investigates the effectiveness of integrating NaOH-based PCM into a shell-and-tube heat exchanger connected to a biomass downdraft gasifier, comparing performance with and without PCM. The aim is to evaluate how PCM affects heat transfer efficiency, thermal storage capacity, and the continuity of thermal delivery.

Extensive research has examined the integration of PCMs into thermal systems to improve energy storage capability and temperature stability. Sharma *et al.* (2009) and Kenisarin (2010) [1], [4] reviewed diverse PCMs for solar and industrial thermal energy systems, emphasizing the potential of inorganic PCMs for high-temperature use. Nomura *et al.* (2010) [3] highlighted NaOH's effectiveness in thermal waste transport systems, while Hosseini *et al.* (2014) [8] investigated the influence of design parameters on the performance of shell-and-tube heat exchangers filled with PCMs. In the biomass sector, studies by Zainal *et al.* (2002) [12] and Osei *et al.* (2020) [13] described the potential of downdraft gasifiers as efficient renewable heat sources. However, there remains a limited number of experimental studies directly integrating high-temperature PCMs with biomass heat transfer systems. This research aims to address this gap by evaluating the performance of a NaOH-filled shell-and-tube heat exchanger for storing and releasing heat from a biomass gasification system, under varying airflow conditions.

## 2. EXPERIMENTAL METHODS

The experimental setup comprises a downdraft biomass gasifier integrated with a shell-and-tube heat exchanger. The gasifier is fueled by biomass wood pellets, which are fabricated from compressed rice straw. During operation, the gasifier combusts the biomass to produce hot producer gas and combustion by-products, which are directed into the shell side of the heat exchanger. A centrifugal blower is employed to supply combustion air to the gasifier and simultaneously drive the airflow through the heat exchanger system as shown in Figure 1.

The shell-and-tube heat exchanger is a custom-fabricated unit designed to allow cross-flow heat exchange between the hot gas (shell side) and a cool air stream (tube side). It was construct by steel tube and plate with external dimensions of width 60 cm, length 55 cm and height 15 cm respectively. Figure 2 illustrates the integrated gasifier and heat exchanger test system. The heat exchanger's shell is a rectangular insulated box with internal baffles guiding the hot gas flow across an array of horizontal tubes.

**Fig. 1. Experimental setup integrating the downdraft gasifier (black reactor on the left) with the shell-and-tube heat exchanger (steel box on the right).**

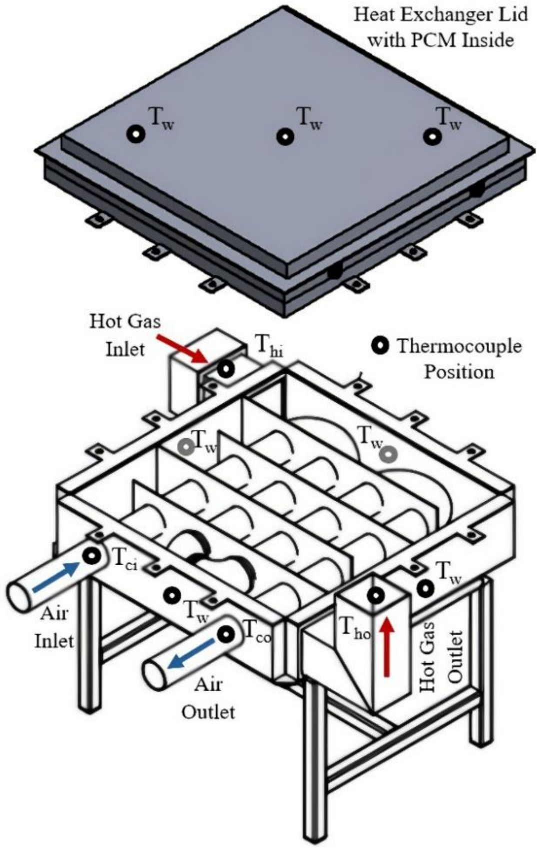


**Fig. 2. Schematic of the shell-and-tube heat exchanger with integrated PCM storage and thermocouple locations.**

The tube bundle consists of multiple steel tubes through which the cool air flows; the tubes are arranged perpendicular to the incoming hot gas flow in a cross-flow configuration. The hot gas enters the shell on one side, flows across the tubes—transferring heat—and exits on the opposite side to an exhaust stack. Meanwhile, the cool air enters the tube inlets at one end of the bundle and exits at the other end, gaining heat in the process. Temperature measurements were taken at various locations: thermocouples $T_{hi}$ and $T_{ho}$ measure the hot gas inlet and outlet temperatures; $T_{ci}$ and $T_{co}$ measure the cold air inlet and outlet temperatures; and $T_w$ indicates the wall temperatures, respectively.

The system was tested under several airflow conditions. The gasifier's air supply and fuel feed were adjusted to maintain a consistent hot gas flow velocity of about 0.663 m/s through the heat exchanger shell. On the tube side, three different cold air flow velocities were set: 1.717 m/s, 2.965 m/s, and 5.490 m/s (representing low, medium, and high cooling air flow rates). In each test, the gasifier was first ignited and allowed to heat the heat exchanger for a fixed heating period of 120 minutes (2 hours) while data were recorded. After 120 minutes, the biomass burner was shut off or its fuel was depleted, and no further heat input was provided. The airflow on both shell and tube sides, however, was maintained, and the cooling air continued to pass through the heat exchanger tubes. Temperature data logging continued for an extended period (several hours) after the heating phase to monitor the cooldown behavior and how the PCM released stored heat. Experiments were run for the two configurations:

•*Without PCM:* The heat exchanger operated as a conventional unit (the PCM chamber was empty, so only sensible heat exchange occurred).

•*With PCM:* A total of 16 kg of NaOH was installed in the exchanger lid to provide latent heat storage.

Thermocouple readings were taken at 5-second intervals at all key points (gas inlet/outlet, air inlet/outlet, PCM, and wall surfaces) and were later averaged to 1-minute intervals for analysis. Air flow rates were measured using anemometers and calibrated by standard flow measurements at the blower inlets. The heat transfer rate to the cold air was computed from the air flow mass rate and the temperature rise of the air across the exchanger. Heat losses were quantified in two ways: (1) *Wall loss* was estimated from the convective loss at the external surface of the heat exchanger, calculated as:

$$Q_{loss,wall} = hA\left(T_w - T_{amb}\right) \quad (1)$$

where $Q_{loss,wall}$ is wall heat loss rate (W). (2) *Exhaust loss* was calculated as the sensible heat remaining in the hot gas stream leaving the exchanger (based on gas flow rate and temperature drop), calculated as:

$$Q_{loss,stack} = \dot{m}_h c_{ph}\left(T_{ho} - T_{amb}\right) \quad (2)$$

where $Q_{loss,stack}$ is sensible heat loss through the stack (W). Using these measurements, the effectiveness ($\varepsilon$) of the heat exchanger was determined from the standard definition for heat exchangers:

$$\varepsilon = \frac{Q_{transfer}}{Q_{\max}} \quad (3)$$

where $Q_{transfer}$ is the actual heat transferred to the cold air, calculated as:

$$Q_{transfer} = \dot{m}_c c_{pc}\left(T_{co} - T_{ci}\right) \quad (4)$$

$Q_{max}$ is the maximum possible heat transfer rate, computed from the minimum heat capacity rate between the hot and cold streams and their inlet temperature difference:

$$Q_{\max} = \min\left(\dot{m}_c c_{pc}, \dot{m}_h c_{ph}\right)\left(T_{hi} - T_{ci}\right) \quad (5)$$

This effectiveness value ($\varepsilon$) essentially reflects the fraction of available thermal energy from the hot stream that is effectively transferred to the cold air. A higher value of $\varepsilon$ indicates better thermal performance of the exchanger.

## 3. RESULTS AND DISCUSSION

### *3.1 Thermal Behavior and Temperature Profiles*

All tests showed a similar heating and cooling profile: during the 120 min heating phase, the outlet cold air temperature gradually rose as the system warmed up. After about 2 hours, the system approached a quasi-steady outlet temperature (which varied with flow rate). Once the gasifier's heat input stopped at 120 min, the outlet air temperature began to drop. However, the rate of cooling differed significantly between the PCM-integrated exchanger and the standard exchanger.

At the lowest cold air velocity (1.717 m/s), the PCM's effect was most pronounced as shown in Figure 3. With PCM, the outlet air cooled much more slowly, remaining above 60°C for about 70 minutes into the cooldown period; without PCM it fell below 60°C after only ~40 minutes. This indicates that the PCM released its stored heat to the air, extending the useful heating duration by roughly 75%. The extended thermal output provided by the PCM is highly beneficial for applications like crop drying or space heating, where maintaining a threshold temperature for a longer time can improve performance.

At higher cold air flow rates, the benefit of the PCM was still evident but somewhat reduced. Faster airflow means more convective cooling, which can extract heat from the system (and PCM) more quickly but also means the baseline (no PCM) system itself delivers more heat during the heating phase. For the medium flow (2.965 m/s) as shown in Figure 4, the PCM configuration kept the outlet air about 10–15°C higher than the non-PCM configuration in the first 30 minutes after shutdown and maintained a small temperature advantage for over an hour.

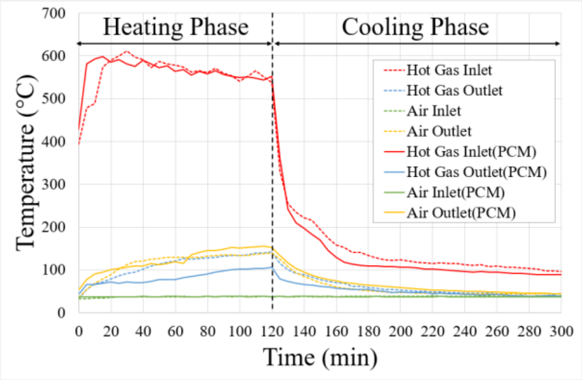


**Fig. 3. Comparison of the temperature distribution over time between heat exchangers with and without phase change material (PCM) at a cold air velocity of 1.717 m/s.**

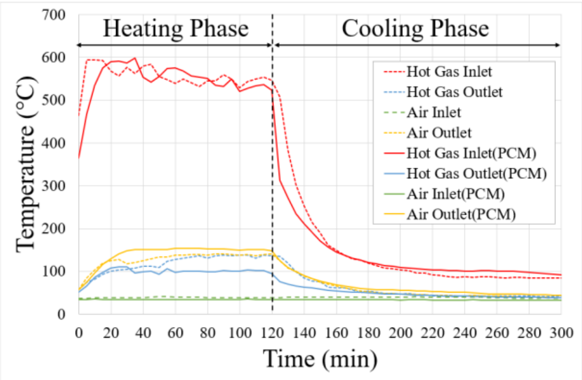


**Fig. 4. Comparison of the temperature distribution over time between heat exchangers with and without phase change material (PCM) at a cold air velocity of 2.965 m/s.**

At the highest flow (5.490 m/s) as shown in Figure 5, the cold air outlet temperatures with and without PCM were closer together (because the large airflow rapidly cools the exchanger in both cases), but the PCM still provided a slight delay in the temperature drop (outlet air stayed >60°C for a few minutes longer with PCM). In all cases, the PCM had melted completely by the end of the heating period and then solidified gradually during the cooldown, releasing stored heat to the passing air.

### *3.2 Comparison of Heat Transfer Performance during the Heating Phase*

During the 2-hour heating period, both the system with PCM and the system without PCM continuously transferred heat to the cold air. However, their outlet temperature profiles differed. In the non-PCM system, the outlet air temperature rose rapidly at the beginning and reached a quasi-steady state within 20–30 minutes. In contrast, the PCM system exhibited a slower rise in outlet air temperature initially, due to part of the heat being absorbed to melt the PCM. After approximately 30 minutes, the outlet temperature of the PCM system gradually approached that of the non-PCM system as the PCM continued to melt and became less effective at absorbing additional heat. Eventually, the molten PCM began transferring heat back to the air more effectively.

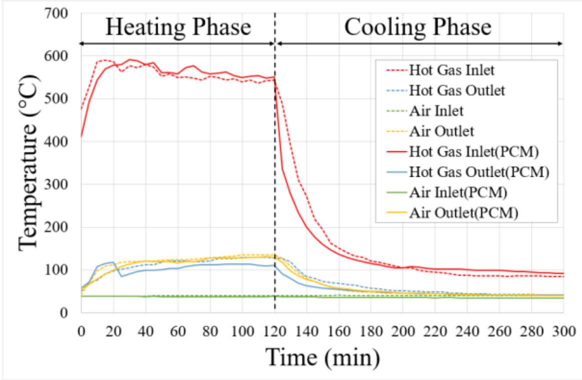


**Fig. 5. Comparison of the temperature distribution over time between heat exchangers with and without phase change material (PCM) at a cold air velocity of 5.490 m/s.**

The total thermal energy transferred to the cold air during the 2-hour heating phase is shown in Table 1. At the lowest air velocity (1.717 m/s), the PCM system delivered slightly less heat (1,371 kJ) compared to the non-PCM system (1,391 kJ), a reduction of about 1.4%, due to the PCM absorbing heat for storage. However, at higher air velocities, the PCM system outperformed the non-PCM case. At 2.965 m/s, it delivered 2,893 kJ compared to 2,491 kJ (+16% improvement), and at 5.490 m/s, it delivered 4,251 kJ compared to 4,044 kJ (+5%). This indicates that at higher airflow rates, the PCM enhances the total heat transfer, as it absorbs excess heat that would otherwise be lost, and later releases it as new air flows through the exchanger.

In this regard, PCM acts as a thermal capacitor - temporarily storing heat that cannot be immediately transferred to the air and gradually releasing it as the airflow continues. This effect is more pronounced at higher air velocities, where the residence time of air in the exchanger is short, and a portion of the hot gas energy would otherwise bypass without being utilized.

**Table 1. Thermal energy transferred to cold air during the 2-hour heating period for systems with and without PCM at different air velocities.**

| Air Velocity (m/s) | $Q_{transfer}$ with PCM (kJ) | $Q_{transfer}$ without PCM (kJ) |
|---|---|---|
| 1.717 | 1,371 | 1,391 |
| 2.965 | 2,893 | 2,491 |
| 5.490 | 4,251 | 4,044 |

### *3.3 Heat Release and Thermal Retention during the Cooling Phase*

The advantage of incorporating PCM is especially apparent during the post-heating phase (2–5 hours), when fuel input is stopped. In the non-PCM system, the outlet air temperature rapidly decreased, falling below 50°C within 30–40 minutes (depending on airflow rate). In contrast, the PCM system maintained a higher outlet temperature for a longer duration due to the release of stored latent heat during solidification.

**Table 2. Thermal energy delivered to cold air during the 3-hour post-heating period (2–5 h) for systems with and without PCM at different air velocities.**

| Air Velocity (m/s) | $Q_{transfer}$ with PCM (kJ) | $Q_{transfer}$ without PCM (kJ) |
|---|---|---|
| 1.717 | 584 | 382 |
| 2.965 | 1,053 | 727 |
| 5.490 | 1,263 | 995 |

For instance, at 1.717 m/s, the outlet air remained above 60°C for about 70 minutes in the PCM system, compared to just 40 minutes without PCM. The total heat released to the air during the 3-hour cooldown is shown in Table 2. At 1.717 m/s, the PCM system delivered 584 kJ versus 382 kJ (+53% gain). At 2.965 m/s, it released 1,053 kJ compared to 727 kJ (+45%), and at 5.490 m/s, it delivered 1,263 kJ compared to 995 kJ (+27%). These results show that PCM significantly extends useful heating time even when the absolute heat release is lower at low flow rates.

Although more heat is released at higher air velocities, the rapid airflow also extracts heat more quickly, shortening the duration of elevated outlet temperatures. For example, at 5.490 m/s, the outlet temperature in the non-PCM system dropped below 60°C within ~20 minutes after shutdown, while the PCM system extended that duration to ~35 minutes. At 2.965 m/s, the PCM maintained outlet air above 60°C for about 50 minutes versus 30 minutes without PCM.

### *3.4 System Efficiency and Energy Utilization Analysis*

The thermal effectiveness (ε) of the heat exchanger was analyzed at three cold air velocities - 1.717, 2.965, and 5.490 m/s - under configurations with and without PCM. Figure 6 presents the comparison of ε values, calculated as the ratio between the actual heat transferred to the air and the theoretical maximum.

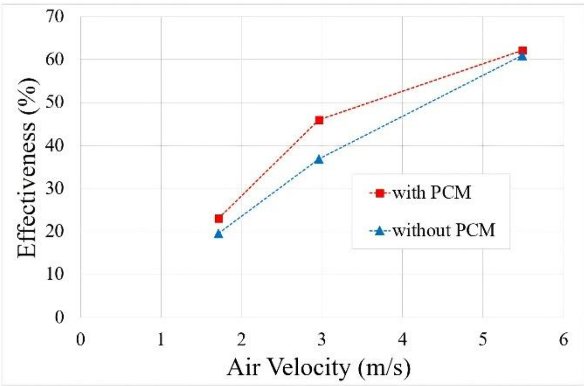


**Fig. 6. Comparison of heat exchanger effectiveness with and without PCM at different cold air velocities.**

At 1.717 m/s, effectiveness improved from 0.1958 (no PCM) to 0.2311 (with PCM), representing an 18% gain. At 2.965 m/s, the enhancement was more significant—from 0.3690 to 0.4604 (+25%). At the highest airflow (5.490 m/s), the effectiveness was 0.6098 (no PCM) and 0.6217 (with PCM), showing a marginal 2% increase.

These results demonstrate that PCM integration most effectively improves performance at moderate airflow rates, where it stores excess heat and gradually releases it, leading to smoother thermal output. At low flow rates, the benefit is constrained by slower heat transfer, while at high flow rates, the rapidly moving air limits the PCM's contribution.

Overall, the PCM-enhanced configuration improves both energy utilization and output stability, particularly valuable for applications such as drying or space heating that require sustained thermal delivery.

### *3.5 Practical Implications*

The integration of a PCM-based heat exchanger into biomass heating systems demonstrates practical benefits for energy efficiency and operational stability. In this study, the use of sodium hydroxide (NaOH) as the phase change material led to up to a 25% increase in heat recovery and extended the useful heat delivery period by approximately 70 minutes after burner shutdown. This result suggests improved thermal regulation and reduced burner cycling, particularly in applications like biomass-fuelled dryers or small-scale process heating systems.

These findings are consistent with previous research that highlights the benefits of PCM integration for high-temperature applications. For example, Nomura *et al.* [3] demonstrated effective waste heat storage using NaOH, while Kumar *et al.* [5] confirmed its heat absorption and release capabilities under steady thermal loading. Such thermal buffering behavior helps maintain a stable heat output even during intermittent operation, reducing energy losses and improving system performance. Consequently, PCM-enhanced heat exchangers offer a promising solution for advancing the efficiency and reliability of renewable thermal systems.

### *3.6 Economic Feasibility and Experimental Limitations*

The use of NaOH as a phase change material (PCM) offers economic advantages due to its low cost and high energy storage density. However, NaOH is highly hygroscopic and corrosive, requiring sealed containment and corrosion-resistant materials, which may increase system complexity and cost. Additionally, the experimental setup in this study was limited to a single charge–discharge cycle, and long-term material stability or performance under repeated thermal cycling was not evaluated. These factors should be addressed in future research to assess the system's practical viability.

## 4. CONCLUSION

This study investigated the thermal performance of a shell-and-tube heat exchanger integrated with sodium hydroxide (NaOH) as a phase change material (PCM) for thermal energy storage in a biomass gasifier system. The experimental and modeling results demonstrated that the inclusion of PCM significantly improved the system's heat storage capacity and thermal stability. Specifically, the PCM extended the duration of useful heating after burner shutdown and increased the total heat transferred to the cold air stream. The effectiveness of the heat exchanger improved by up to 25% at moderate airflow rates. These findings highlight the potential of PCM-enhanced systems to increase energy efficiency and reliability in applications with intermittent heat sources, such as biomass combustion.


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## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | = | external wall surface area ($m^2$) |
| $c_p$ | = | specific heat capacities (kJ/kg·K) |
| $h$ | = | convective heat transfer coefficient ($W/m^2K$) |
| $\dot{m}$ | = | mass flow rate (kg/s) |
| $Q_{transfer}$ | = | actual heat transferred rate(W) |
| $Q_{\max}$ | = | maximum heat transfer rate (W) |
| $T$ | = | temperature (°C) |

**Greek symbols**

| | | |
|---|---|---|
| $\varepsilon$ | = | effectiveness of the heat exchanger |

**Subscript**

| | | |
|---|---|---|
| $amb$ | = | ambient air |
| $c$ | = | cold air |
| $h$ | = | hot gas |
| $i$ | = | inlet |
| $o$ | = | outlet |
| $w$ | = | wall |

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