

Energy Saving for a Split-Type A/C Using together a Condensate Water with Air Ventilation

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Article info:

Received: 23 February 2026

Revised: 4 May 2026

Accepted: 23 July 2026

DOI:

[10.69650/rast.2026.266343](https://doi.org/10.69650/rast.2026.266343)

Keywords:

Air Conditioning
Condensate Water
Energy Saving
Ventilation Air
Split-Type

ABSTRACT

The study aims to reduce the temperature of the incoming air prior to condenser cooling by mixing and exchanging heat with ventilation air and cool condensate water. The study room was designed 2.4 m×2.9 m×2.2 m and had an installed split-type air conditioner of 12,000 BTU/hr and a fixed-speed compressor, that existing ventilation fan was used with a condensate–air heat exchange panel. The measurement condition included ambient temperature and relative humidity. Electrical consumption was from meter measurement, the study room was fixed 25°C and ambient range 28.1–38.0°C. The experiment details, Case 1, representing the air-conditioning system, and Case 2, the condensate-water and ventilation-air heat exchange system. The results of the study, for Case 1, the average coefficient of performance (COP) of 3.51, energy efficiency average ratio (EER) of 9.31. For Case 2, the average COP and EER increased to 3.69 and 10.27, respectively. The case study compared with the conventional system, the COP increased 5.23% and EER 10.69%. These demonstrate that the integration of condensate water and ventilation air for heat exchange effectively enhances the energy efficiency of the air-conditioning system and contributes to a reduction in energy consumption. The proposed method is therefore considered a practical and energy-efficient solution for improving air-conditioning performance in office buildings.

1. Introduction

Energy is a fundamental factor playing a crucial role in human life and is a key driver of the economy and national development across all sectors, including industry, services, and households. However, the continuously increasing demand for energy has raised concerns about long-term energy security and the environmental impact of inefficient energy use. Currently, air conditioning systems are one of the most widely used electricity-intensive systems in buildings and residences, especially in countries with hot and humid climates, where continuous use of air conditioning is necessary. This has led to a significant increase in the proportion of energy consumption associated with air conditioning systems. Therefore, many researchers are focusing on studying and developing ways to reduce energy consumption and improve the efficiency of air conditioning systems in various forms. Based on related research, various methods for saving energy in air conditioning systems have been presented. This research aims to address the high energy consumption of air-conditioning systems, particularly the decline in condenser efficiency at high outdoor temperatures. A pre-cooling method is therefore proposed, utilizing condensate water and ventilation air to reduce the temperature of the air entering the condenser, thereby decreasing the thermal load and overall energy consumption. The study investigates system performance, designs and experimentally tests a condensate water reuse technique, and evaluates its economic feasibility for practical implementation. The research is conducted based on a review of relevant previous studies, as follows. A 50% increase in the coefficient of performance (COP) was achieved, accompanied by a reduction in power consumption of up to 20%. Hao et al. [1] demonstrated that, across varying climatic conditions, the optimal evaporative pad

thickness in air-cooled chillers can yield energy savings between 2.4% and 14% relative to conventional air-cooled systems. Yang et al. [2] reported that integrating a direct evaporative cooler with an air-cooled chiller reduced the condensing temperature by 2.1–6.2°C, leading to a 1.4–14.4% decrease in chiller power consumption and a 1.3–4.6% increase in refrigeration capacity under head-pressure control. Gu et al. [3] proposed and evaluated a novel evaporative condenser featuring an annular elliptic finned-tube heat exchanger, with particular emphasis on the influence of cooling-pad thickness on the performance of a split-type air-conditioning system using an evaporatively cooled condenser. Evaporative cooling has been shown to significantly enhance air-conditioning performance. Specifically, it improved the coefficient of performance (COP) by up to 44% and reduced power consumption by up to 20% [4]. Similarly, retrofitting the condenser of an occupied test house with an evaporative cooling unit achieved a 20% reduction in power consumption. Islam et al. [5] performed experimental and numerical investigations of heat and mass transfer in an evaporative condenser and reported a 28% increase in COP for a commercial air-conditioning system. Hosoz and Kilicarslan [6] further examined the performance limitations of conventional air-conditioning systems. In addition to the measurable decline in thermodynamic efficiency expressed as a reduction in the coefficient of performance (COP), findings indicated that such operational inefficiencies also accelerate compressor wear, thereby shortening its operational lifespan [7]. When the ambient temperature exceeds 35°C, the coefficient of performance (COP) of most air-conditioning systems typically decreases from 2.7–3.2 to approximately 2.2–2.4. In addition, continuous operation at outdoor temperatures above 52°C substantially increases condenser pressure, which may activate protective safety limits and cause system shutdown

[8]. Although both technologies are intended to enhance heat-transfer performance, cooling pads pose a lower risk of scaling and particulate deposition, whereas water-spray systems generally offer lower installation and operating costs [9]. It has been demonstrated that augmenting spray pressure significantly enhances the heat transfer rate [10]. Air conditioning represents nearly 60% of residential energy use in the Kingdom of Saudi Arabia (KSA) [11]. It was also reported that for each 1°C rise in outdoor dry-bulb temperature, the COP and total cooling capacity of a split-type air-conditioning system decline by about 2%. One method that has garnered significant attention from researchers for improving energy efficiency and system performance in direct expansion (DX) vapor compression systems is the pre-cooling of air entering the condenser using evaporative cooling techniques. Hajidavalloo [12] evaluated the effect of evaporative pre-cooling of the condenser inlet air for a window-type air-conditioning system under high ambient conditions, reporting a 16% decrease in power consumption and a 55% increase in COP. Several studies have demonstrated that evaporative cooling can effectively enhance the performance of air-conditioning systems by reducing the condenser inlet air temperature. A previous investigation [13] showed that integrating an indirect evaporative cooler for condenser air pre-cooling in a conventional packaged air conditioner could reduce electrical energy consumption by up to 55%. Waly et al. [14] further reported that using condensate water for evaporative pre-cooling in a split-type air conditioner achieved energy savings of 5.3% during October, while daily reductions of 5% in June and 4.5% in August were observed in Beirut due to extended cooling availability from condensate. In addition, condensate-assisted evaporative cooling has been applied to window-type air-conditioning systems, demonstrating measurable performance improvements [15]. The effectiveness of direct evaporative cooling (DEC) has also been evaluated under hot-dry conditions, such as in Morocco where annual peak temperatures exceed 45°C, with simulations indicating a reduction in inlet air temperature from 44°C to 26.7°C. Al-Badri [16] reported that DEC increased the coefficient of performance (COP) by 17.8–33.24% and reduced electrical current by 0.12–0.16 A per 1°C temperature drop. For hot-humid climates, Borirak and Kotchapoom [17] demonstrated that condenser cooling using DEC increased COP from approximately 4.3 to 5.2 in a 1 TR air-conditioning unit in Thailand. Similarly, Martinez et al. [18] experimentally showed that a condensate-wetted DEC pad applied to a 1.5 TR R410A split-type air conditioner improved cooling capacity and energy efficiency ratio (EER) by 19% and 36%, respectively. Given that refrigeration and air-conditioning systems account for approximately 15% of global electricity consumption and nearly 20% in developed countries [20, 21], improving condenser-side heat rejection remains a critical pathway for reducing energy demand and associated emissions. Suri et al. (1989) [22] experimentally demonstrated that the water-cooled configuration reduced peak power demand by 40% and total energy consumption by 25%. Such systems are commonly implemented in commercial facilities, including hotels, where multiple units share a central cooling tower. However, under certain conditions, when only a few units are in operation, the cooling tower may continue functioning, resulting in significant electrical energy consumption despite the reduced load on the air-conditioning system. An open pond system has been proposed as an alternative heat sink for water-cooled air conditioners. The cooling mechanisms of an open pond include convective and evaporative heat exchange with the surrounding ambient air, as well as radiative heat exchange with the sky [23]. When implemented as a rooftop system, a designed open-pond cooling configuration was reported to reduce peak roof temperature by up to 87%. Kharrufa and Adil (2008) [24] evaluated the performance of an open pond for passive building cooling and found that it reduced the

maximum indoor temperature from 33.97°C to 28.41°C. In addition to passive applications, several studies have proposed using open ponds as heat sinks for air-conditioning systems. For example, Pohl (1999) [25] utilized a swimming pool to absorb heat rejected from an air-conditioner condenser, achieving reductions of 25–30% in electrical energy consumption and 30–35% in peak demand compared with a conventional air-cooled condenser unit. Similar approaches have also been investigated by Ngammuang et al. (2018) [26].

2. Experimental detail

2.1 Equipment installation

In Case 1, the air-conditioning system operated under normal conditions to provide a baseline for comparison. Experiments were performed in an office room (2.4 m × 2.9 m × 2.2 m), representative of a typical small office in residential and commercial buildings. This controlled indoor environment allowed for stable operating conditions and ensured repeatability of the experimental results. A standard split-type air-conditioning unit rated at 12,000 BTU/h was installed in the test room. The unit operated at a constant compressor speed without inverter control, thereby eliminating the influence of variable-speed operation on system performance. In addition, one set of ventilation fans was installed to represent typical indoor air circulation conditions during normal office operation. Under normal operating conditions in Case 1, the system functioned without any energy-saving modifications. The condensate water produced at the evaporator during the dehumidification process was discharged directly and not reused, while the ventilation air was exhausted to the outdoor environment without participating in any heat exchange process. This operating mode represents the conventional practice of standard split-type air-conditioning systems. During the experiments, the indoor air temperature was maintained at a constant setpoint of 25°C to ensure thermal comfort. Meanwhile, the outdoor air temperature varied naturally between 28.1°C and 38.0°C, reflecting typical daytime conditions in hot climates. The experiment was conducted continuously from 9:00 a.m. to 5:00 p.m. to capture the system performance under varying ambient thermal loads throughout the day.

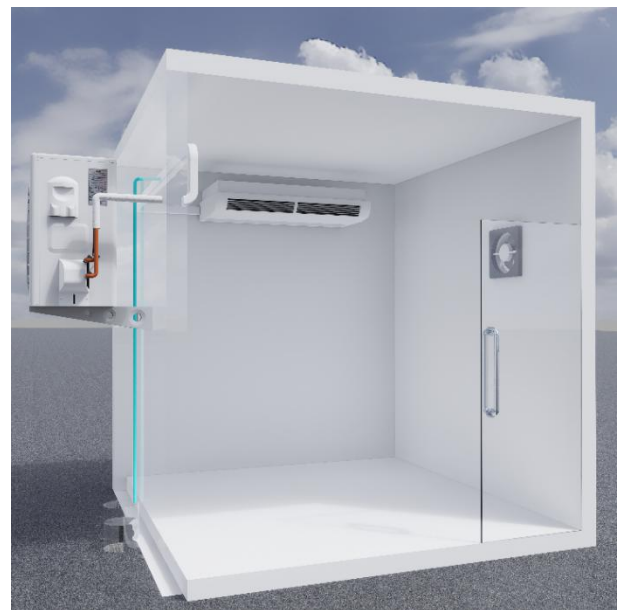


Fig. 1 Installation of the air-conditioning system operated under normal conditions.

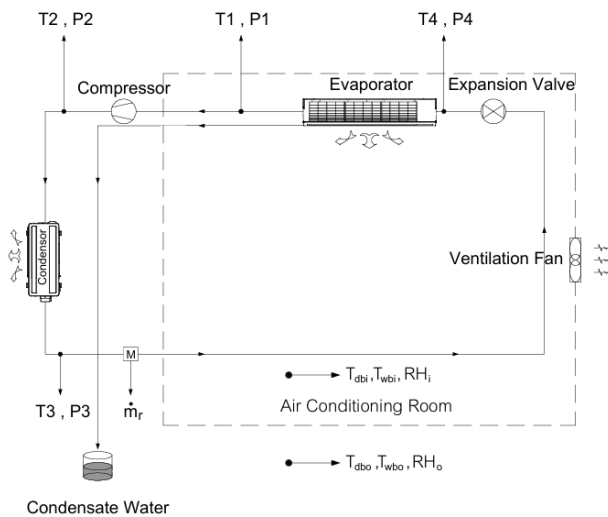


Fig. 2 Installation of the air-conditioning system operated under normal conditions diagram.

The installation configuration and layout of the air conditioning unit and ventilation fans used in Case 1 are illustrated in Figures 1 and 2. These figures provide a clear representation of the experimental setup and measurement locations, which serve as a reference for performance evaluation and comparison with the modified system in subsequent cases.

The figure 2 presents a schematic diagram of the proposed split-type air conditioning system incorporating condensate water utilization and ventilation air to enhance condenser cooling performance. The system operates on a conventional vapor-compression refrigeration cycle and comprises four main components: a compressor, condenser, expansion valve, and evaporator. The cycle begins at the compressor, where the refrigerant is compressed from a low-pressure, low-temperature state to a high-pressure, high-temperature condition (T_2, P_2). The high-energy refrigerant then enters the condenser, where it rejects heat to the surrounding environment. In this modified system, the condenser is assisted by a condensate water cooling arrangement. Condensate water generated at the evaporator during the dehumidification process is collected and directed to a copper-tube panel heat exchanger. The condensate absorbs heat from the surface of the heat exchanger and from the incoming airflow, thereby enhancing convective heat transfer. As a result, the temperature of the incoming air is reduced before it is used to cool the condenser unit. A ventilation fan is installed near the condenser to enhance airflow across the heat exchanger. The ventilation air is mixed with outdoor air before entering the condenser, which further lowers the inlet air temperature and improves heat rejection efficiency. The combined use of condensate water and ventilation air lowers the condensing temperature and pressure, thereby reducing compressor power consumption. After rejecting heat in the condenser, the refrigerant leaves at a lower temperature and pressure (T_3, P_3) and then passes through the expansion valve, where its pressure and temperature decrease (T_4, P_4). The refrigerant subsequently enters the evaporator and absorbs heat from the indoor air, providing cooling to the conditioned space. It exits the evaporator at (T_1, P_1) and returns to the compressor, completing the cycle. Indoor air conditions were monitored using dry-bulb temperature ($T_{db,i}$), wet-bulb temperature ($T_{wb,i}$), and relative humidity (RH,i), while outdoor conditions were measured as dry-bulb temperature ($T_{db,o}$), wet-bulb temperature ($T_{wb,o}$), and relative humidity (RH,o). In addition, the refrigerant mass flow rate ($\dot{m}_r$) was measured to enable detailed thermodynamic performance

evaluation, including the calculation of the coefficient of performance (COP) and energy efficiency ratio (EER).

In case 2, Modified air conditioning system developed in this research to enhance energy efficiency through improved condenser cooling. The experiment was conducted in the same office room used in Case 1, with internal dimensions of $2.4 \text{ m} \times 2.9 \text{ m} \times 2.2 \text{ m}$, in order to ensure consistency and comparability between the two experimental cases. By maintaining identical room geometry and thermal conditions, the influence of external variables was minimized, allowing the performance differences to be attributed primarily to the system modification. The air conditioning unit used in Case 2 was a split-type air conditioner with a nominal cooling capacity of 12,000 BTU/h operating at a constant compressor speed. The same ventilation fan installed in Case 1 was retained in this configuration to represent typical indoor ventilation conditions. This ensured that any observed improvement in system performance was not influenced by changes in ventilation equipment or airflow rate. Unlike the standard system, the modified system incorporated a copper coil heat exchanger installed at the condenser air inlet. The heat exchanger was designed to utilize condensate water generated by the evaporator as a cooling medium. The condensate water was circulated through a copper coil to facilitate heat exchange between the relatively cool condensate and the warmer incoming outdoor air. In addition, ventilation air was mixed with the outdoor air stream before entering the condenser. This combined approach reduced the temperature of the air supplied to the condenser, thereby improving heat rejection performance. During the experimental period, the indoor air temperature was maintained at 25°C to ensure comparable thermal comfort conditions with Case 1. The outdoor air temperature varied naturally between 28.1°C and 38.0°C , representing typical daytime conditions in hot climates. Experiments were conducted continuously from 9:00 a.m. to 5:00 p.m., covering the peak cooling-demand period and enabling evaluation under fluctuating ambient conditions. Key operating parameters, including indoor and outdoor air temperatures, relative humidity, and electrical energy consumption, were monitored throughout the tests. The installation layout and configuration of the modified system, including the copper coil heat exchanger and airflow mixing arrangement, are illustrated in Figures 3 and 4. These figures provide a detailed representation of the experimental setup and support the analysis of energy saving.

Figure 4 illustrates the schematic configuration of the experimental split-type air-conditioning system integrated with a condensate-water heat exchanger and a mechanical ventilation unit. The system operates on a conventional vapor-compression refrigeration cycle comprising a compressor, condenser, expansion valve, and evaporator. Refrigerant thermodynamic states at key locations are monitored using temperature and pressure sensors and are denoted as (T_1, P_1), (T_2, P_2), (T_4, P_4), and (T_3, P_3). The evaporator is installed inside the conditioned room to remove both sensible and latent heat from the indoor air, thereby maintaining the desired thermal conditions. Condensate water generated during the dehumidification process is collected and routed to a heat exchanger located upstream of the ventilation air stream. This heat exchanger enables indirect heat transfer between the low-temperature condensate and the incoming outdoor air. A ventilation fan draws indoor air through the heat exchanger, reducing its temperature before it enters the conditioned space. Air and water flow rates are controlled and measured, as indicated by the volumetric airflow rate ($\dot{V}_a$) and the condensate mass flow rate ($\dot{m}_r$). This pre-cooling process reduces the ventilation load imposed on the air-conditioning system. Indoor and outdoor air conditions are continuously monitored in terms of dry-bulb temperature (T_{db}), wet-

bulb temperature (T_{wb}), and relative humidity (RH). Overall, the integration of condensate heat recovery with ventilation aims to improve system energy efficiency by utilizing waste cooling potential, reducing condenser and ventilation loads, and enhancing overall thermal performance.

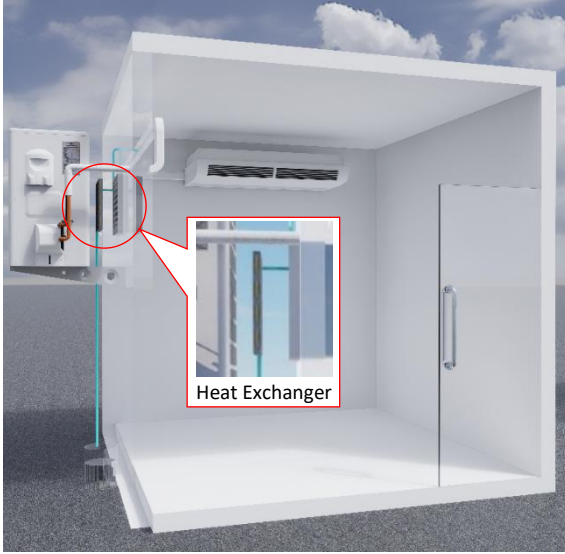


Fig. 3 Air conditioning system installation for research.

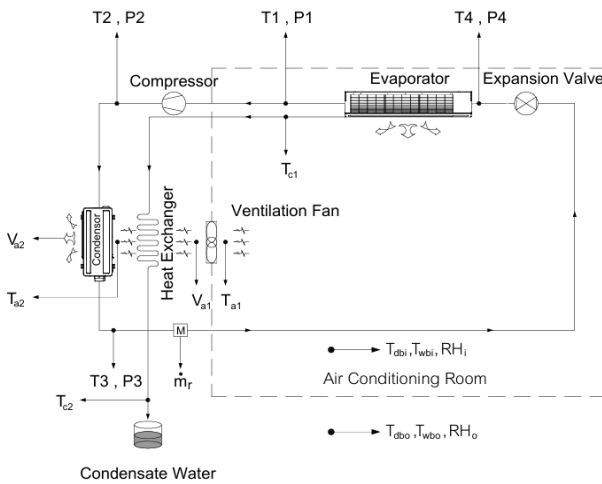


Fig. 4 Research air conditioning system installation diagram.

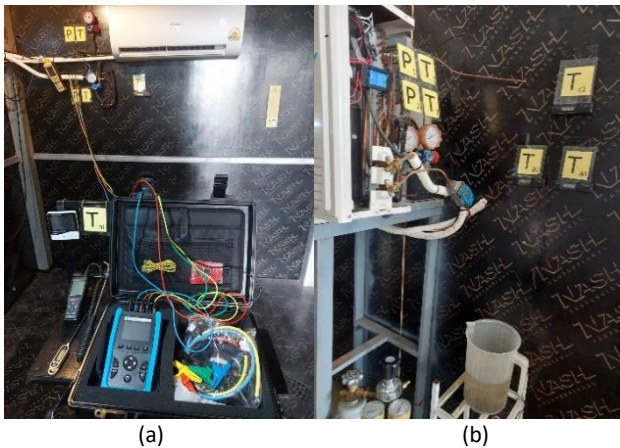


Fig. 5 Instruments and equipment in the laboratory setup: (a) inside the room, and (b) outside the room.

The installation layout of the measuring instruments and experimental equipment for the laboratory-scale air-conditioning system is illustrated in Fig. 5(a). This figure presents the arrangement

of all major components installed inside the experimental room, including the evaporator unit, temperature and pressure sensors, airflow measurement devices, and data acquisition instruments. These components were strategically positioned to ensure accurate monitoring of the thermodynamic and psychrometric parameters of the conditioned air, as well as to minimize measurement uncertainty and external disturbances during operation. Figure 5(b) illustrates the installation of the outdoor components of the experimental system, which are located outside the air-conditioning room. This configuration includes the condenser unit, compressor, condensate water collection system, heat exchanger, and ventilation fan. The outdoor setup was designed to facilitate effective heat rejection to the ambient environment and to enable the integration of the condensate-water heat exchanger with the ventilation air stream. The separation between indoor and outdoor installations reflects real-world operating conditions of a split-type air-conditioning system and ensures that the experimental results accurately represent practical applications.

2.2 Experimental procedure

After the installation of all equipment and measuring instruments was completed, the air-conditioning system was subjected to an initial operational test to verify proper functionality and experimental readiness. This commissioning process ensured that all major components of the refrigeration cycle, including the compressor, condenser, expansion valve, and evaporator, operated under stable and normal conditions. In addition, the airflow, condensate drainage, and data acquisition systems were inspected to confirm reliable and continuous measurements throughout the experimental period. Following the successful completion of the preliminary testing and system adjustment, the experimental investigation was conducted continuously from 9:00 a.m. to 5:00 p.m. under controlled operating conditions corresponding to each experimental case. During the experiment, key operational and environmental parameters were systematically recorded at regular time intervals. These parameters included electrical power consumption, indoor and outdoor air conditions, and other thermodynamic variables required for performance evaluation. The recorded data were subsequently used to calculate key performance indices of the air-conditioning system, namely the coefficient of performance (COP) and the energy efficiency ratio (EER), for both experimental cases. These indicators were then used to assess and compare the system's energy efficiency and cooling performance under different configurations, providing a quantitative basis for analysis and discussion.

The coefficient of performance (COP) is a key indicator for evaluating the efficiency of an air-conditioning system. It is defined as the ratio of useful cooling output to the electrical energy input required to produce that cooling effect. COP is dimensionless and is widely used in both experimental and analytical assessments of refrigeration and air-conditioning systems.

Mathematically, the COP of an air-conditioning system is defined as:

$$\text{COP} = \frac{Q_t}{W_t} = \frac{\dot{m}_r(h_1 - h_4)}{\dot{m}_r(h_2 - h_1)} = \frac{(h_1 - h_4)}{(h_2 - h_1)} \quad (1)$$

- COP = Coefficient of performance of air conditioning system
- Q_t = Total cooling capacity (W or kW), representing the rate at which heat is removed from the conditioned space.
- W_t = Total power consumption of air conditioning system (W or kW), including the compressor and auxiliary components such as fans or pumps, if applicable.
- $\dot{m}_r$ = Mass flow rate of refrigerant (kg/s)
- h_1 = Enthalpy of refrigerant before entering compressor (kJ/kg)

h_2 = Enthalpy of refrigerant after passing through compressor (kJ/kg)

h_4 = Enthalpy of refrigerant before entering evaporator (kJ/kg)

A higher COP indicates a more efficient air-conditioning system, as it reflects greater cooling output per unit of electrical energy consumed. In contrast, a lower COP implies reduced efficiency and higher energy use for the same cooling demand. Consequently, COP is widely applied to compare different air-conditioning systems or alternative operating configurations of the same system. In experimental studies, the cooling capacity (Q_c) can be determined using either an air-side energy balance (based on air mass flow rate and the enthalpy difference across the evaporator) or a refrigerant-side energy balance (based on refrigerant mass flow rate and the enthalpy difference across the evaporator). The power input W_t is typically measured directly using an electrical power meter. Since COP depends on operating conditions such as indoor and outdoor temperatures, humidity, and system configuration, it is usually evaluated under steady-state conditions and averaged over a specific operating period to obtain representative performance values. As a result, COP serves as a fundamental metric for assessing energy efficiency improvements and the effectiveness of energy-saving techniques in air-conditioning systems.

The energy efficiency ratio (EER) is defined as the ratio of an air-conditioning system's cooling capacity to the total electrical power input during operation. It represents the amount of cooling produced per unit of electrical power and is commonly expressed in Btu/h·W⁻¹. EER is widely used to evaluate and compare the energy performance of air-conditioning systems under specified operating conditions. In this definition, cooling capacity is reported in British thermal units per hour (Btu/h), which represents the rate of heat removal from the conditioned space, while the electrical input is expressed in watts (W), accounting for the power consumed by the compressor and auxiliary components (e.g., fans), when applicable. This unit convention follows standard practice in the air-conditioning and refrigeration industry. A higher EER value indicates that the air-conditioning system can deliver a greater cooling output for the same amount of electrical energy input, signifying improved energy efficiency. Conversely, a lower EER value implies higher energy consumption for an equivalent cooling capacity. Therefore, the EER serves as an important performance indicator for assessing system efficiency and for evaluating the effectiveness of energy-saving measures in air-conditioning applications.

$$EER = \frac{Q_E}{W_{input}} \times 3.412 \quad (2)$$

EER = Energy efficiency ratio

Q_E = Cooling capacity (Btu/hr)

Q_t = $Q_E \times 3.412$ (Watt)

W_{input} = Total work in put to system (Watt)

In experimental studies, the cooling capacity used to calculate the EER can be obtained from air-side or refrigerant-side energy balance calculations, while the electrical power input is typically measured directly using an energy or power meter. Because EER is dependent on operating conditions such as indoor and outdoor temperatures and humidity levels, it is generally evaluated under controlled or steady-state conditions to ensure meaningful comparison between different systems or operating cases. As a result, the EER is commonly employed in research and industry to assess and compare the energy performance of air-conditioning systems and to quantify the effectiveness of energy-saving strategies aimed at reducing electricity consumption.

Condensate water in an air-conditioning system is formed when moist air passes over the evaporator coil, which is below the dew point, causing water vapor to condense into liquid. The experiment showed a continuous average condensate production of 0.625 L/h throughout the testing period. This rate reflects the air humidity and latent load, with stable values due to controlled indoor conditions at 25°C. The condensate amount also depends on outdoor temperature, relative humidity, airflow rate, and heat transfer efficiency. The collected condensate can be reused for heat exchange to pre-cool air entering the condenser, improving heat rejection, reducing compressor load, and lowering overall energy consumption.

2.3 Theoretical Design of the Copper Tube Heat Exchanger

Figure 6 the design of the heat exchanger in this study is based on utilizing low-temperature condensate water to reduce the temperature of incoming air before it enters the condenser of the air-conditioning system. The fundamental principle involves heat transfer between two fluids: condensate water flowing inside the tubes and air flowing across the external surface. The heat exchanger is constructed using copper tubes with a diameter of 4.5 mm, arranged in a serpentine configuration to increase the flow path length and enhance the heat transfer area. The tubes are combined with wire fins of 1 mm diameter to further increase the effective surface area and improve convective heat transfer between the air and the tube surface.

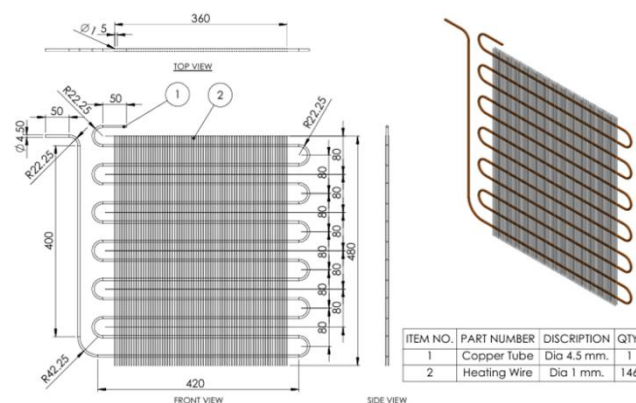


Fig. 6 Theoretical Design of the Copper Tube Heat Exchanger

In theory, heat transfer occurs when the relatively cooler condensate water flows inside the copper tubes, absorbing heat from the warmer ambient air passing over the external surface. As a result, the air temperature decreases before entering the condenser, while the condensate water temperature increases accordingly. Copper is selected as the tube material due to its high thermal conductivity, which enables efficient heat transfer, as well as its durability in practical applications. The arrangement of multiple tube passes and appropriate spacing between tubes ensures uniform airflow distribution, minimizes air resistance, and maximizes heat exchange efficiency. The design is governed by key parameters, including condensate water temperature, water flow rate, heat transfer surface area, airflow velocity, and the temperature difference between air and water. Increasing the heat transfer area and the contact time between air and the tube surface enhances the overall heat transfer rate, leading to more effective pre-cooling of the air. In summary, the proposed copper tube heat exchanger is a simple, low-energy design that effectively utilizes condensate water generated within the air-conditioning system. It serves as a practical solution to reduce the thermal load on the condenser, improve system performance, and decrease overall energy consumption.

3. Experimental results

The coefficient of performance (COP) and energy efficiency ratio (EER) of the air-conditioning system were experimentally evaluated for both operating conditions using Eqs. (1) and (2). Case 1 represents the conventional air-conditioning system, while Case 2 corresponds to the modified system proposed in this research. The calculated COP and EER values were derived from the measured experimental data and were used as key indicators to assess and compare the performance of the two systems. The experimental results for Case 1 and Case 2 are summarized in Tables 2–4, which present the measured parameters and corresponding performance indices. Variations in COP and EER under different operating conditions are further illustrated in Figs. 6–14 to highlight performance trends and differences between the baseline and modified systems. Overall, the modified configuration demonstrated improved performance relative to the standard system, as evidenced by higher COP and EER values. This comparison confirms the effectiveness of the proposed approach in enhancing system efficiency and reducing energy consumption under the tested conditions.

Weather conditions during the experimental period (9:00 a.m. to 5:00 p.m.) were continuously monitored to ensure that the results accurately captured the influence of ambient conditions on air-conditioning performance. Key outdoor parameters, including ambient air temperature and relative humidity, were recorded throughout the day because they directly affect condenser heat rejection and overall system efficiency. The outdoor air temperature exhibited noticeable daytime fluctuations, which were considered in the analysis since variations in ambient temperature influence condenser thermal loading and, consequently, the coefficient of performance (COP) and energy efficiency ratio (EER). Outdoor relative humidity was also measured to provide a complete characterization of the operating environment. The recorded weather data were used to define consistent boundary conditions for comparing the two experimental cases. Documenting ambient conditions over the full operating period improved the reliability and practical relevance of the performance evaluation.

$T_{in\ room} : T_{dbi}$	=	28.1°C
T_{wbi}	=	26.5°C
$T_{out\ room} : T_{dbo}$	=	28.1°C–38.0°C
T_{wb}	=	26.5°C–31.0°C
RH_i	=	78 - 50 %
RH_o	=	78 - 42 %
Room temperature control		
T_{dbi}	=	25.0°C
Refrigerant R-32		
$\dot{m}_r$	=	0.025 (kg/s)

Table 1 Weather conditions.

Times	T_{dbi} (°C)	T_{dbo} (°C)	%RH _o
9.00	25	28.1	78
10.00	25	30.5	69
11.00	25	33.3	63
12.00	25	36.6	58
13.00	25	37.1	54
14.00	25	38.0	49
15.00	25	36.6	45
16.00	25	35.8	42
17.00	25	33.4	46

Table 1 summarizes the weather conditions during the experiment conducted from 9:00 a.m. to 5:00 p.m., while the indoor temperature was maintained at a constant set point of 25°C. The outdoor dry-bulb temperature at 9:00 a.m. was 28.1°C, increased gradually with time, and reached a maximum of 38.0°C at 2:00 p.m., before decreasing toward the evening. The outdoor relative humidity was 78% at the beginning of the experiment and declined throughout the day, reaching a minimum of 42% at 4:00 p.m., after which it began to rise again in the evening.

T_{dbi}	=	Room temperature control value (°C)
T_{dbo}	=	Dry bulb temperature (°C)
RH_o	=	Relative humidity of outdoor (%)

Figure 7 illustrates the variation of key weather parameters during the experimental period from 09:00 to 17:00 h, including indoor dry-bulb temperature ($T_{db,i}$), outdoor dry-bulb temperature ($T_{db,o}$), and outdoor relative humidity (RH_o). Throughout the entire test, the indoor dry-bulb temperature was maintained at approximately 25°C, demonstrating that the air-conditioning system was able to provide stable indoor thermal control.

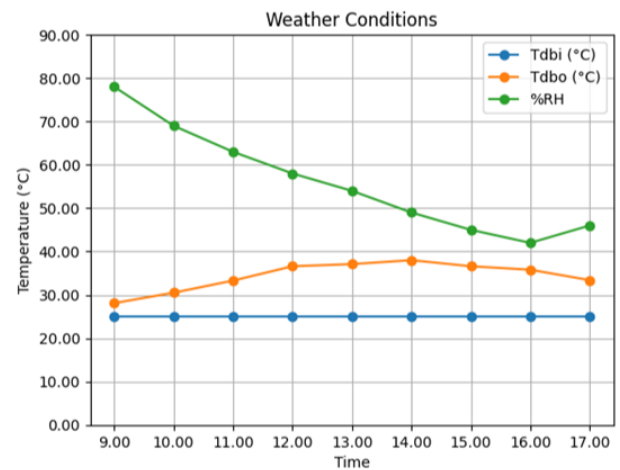


Fig. 7 Weather conditions.

Maintaining a constant indoor setpoint is essential for ensuring that performance differences between Case 1 and Case 2 are primarily attributable to the system configuration rather than variations in indoor load conditions. In contrast, the outdoor dry-bulb temperature exhibited a clear diurnal trend. It increased from approximately 28.1°C at 09:00 h to a peak value of about 38.0°C around 14:00 h, and then gradually declined toward the late afternoon. This temperature profile is representative of typical daytime operation in hot climates and imposed a progressively higher thermal load on the condenser during the peak period. Such variations are particularly relevant for evaluating condenser-side enhancement techniques because condenser heat rejection becomes increasingly constrained as ambient temperature rises, which directly affects condensing pressure, compressor power consumption, and overall system efficiency. The outdoor relative humidity showed an inverse trend relative to temperature, decreasing from approximately 78% in the morning to around 42–45% during the peak temperature period, before slightly increasing toward the end of the day. This behavior reflects common daytime weather patterns and is important because humidity influences the potential effectiveness of evaporative and condensate-assisted cooling processes, as well as the overall heat and mass transfer characteristics of the air stream. Overall, Fig. 6 confirms that the experiments were conducted under realistic and dynamically changing outdoor conditions while maintaining a controlled indoor environment. The recorded ambient conditions

therefore provide a reliable and practically relevant basis for evaluating and comparing the COP and EER of the baseline and modified air-conditioning systems.

Table 2 summarizes the coefficient of performance (COP) values measured during the experimental period from 09:00 to 17:00 h, while the indoor temperature was maintained at a constant setpoint of 25°C. This controlled indoor condition is important because it ensures that variations in COP primarily reflect changes in outdoor boundary conditions and system configuration, rather than fluctuations in indoor cooling demand. At the beginning of the test (09:00 h), the COP of the baseline system (Case 1) was 4.12, whereas the proposed system (Case 2) achieved a COP of 4.18, representing an improvement of 1.46%. As the experiment progressed toward midday, the outdoor temperature increased and a gradual decline in COP was observed in both cases. This trend is consistent with the thermodynamic behavior of vapor-compression air-conditioning systems, in which elevated ambient temperature reduces condenser heat-rejection effectiveness, increases condensing pressure, and consequently raises compressor work. These effects lead to lower system efficiency even when indoor setpoint conditions are maintained. Despite this overall reduction, Case 2 consistently exhibited higher COP values than Case 1 throughout the entire operating period.

Table 2 Results of COP in Cases 1 and 2.

Times	COP _{Case 1}	COP _{Case 2}	%COP _{Diff}
9.00	4.12	4.18	1.46
10.00	3.85	4.02	4.42
11.00	3.66	3.78	3.28
12.00	3.45	3.64	5.51
13.00	3.28	3.55	8.23
14.00	3.12	3.36	7.69
15.00	3.26	3.48	6.75
16.00	3.39	3.57	5.31
17.00	3.42	3.62	5.85
Avg.	3.51	3.69	5.23

This indicates that the proposed modification provides a sustained improvement in performance under realistic and fluctuating outdoor conditions. The difference between the two cases became more pronounced during periods of higher thermal stress. For example, at 13:00 h, when outdoor temperature was near its peak, the COP of Case 2 exceeded that of Case 1 by 8.23%. This behavior suggests that the proposed approach is particularly effective when condenser-side heat rejection becomes a limiting factor for system operation. Overall, the average COP of Case 2 was 5.23% higher than that of Case 1 over the full experimental duration. These results confirm that the proposed configuration enhances system efficiency and offers meaningful potential for energy savings, especially during high-temperature daytime operation when conventional air-conditioning systems typically experience the greatest performance degradation.

Figures 7 and 9 illustrate the time-series variation of the coefficient of performance (COP) for the baseline system (Case 1) and the proposed system (Case 2) during the operating period from 09:00 to 17:00 h, while the indoor temperature was maintained at a constant setpoint of 25 °C. Maintaining a fixed indoor setpoint is critical for isolating the influence of outdoor conditions and condenser-side modifications on system performance, thereby enabling a fair comparison between the two configurations. At the beginning of the experiment (09:00 h), the COP of Case 1 was 4.12, while Case 2 achieved a COP of 4.18, corresponding to an initial improvement of 1.46%. As the day progressed,

the outdoor air temperature increased toward midday, and a gradual decline in COP was observed for both cases.

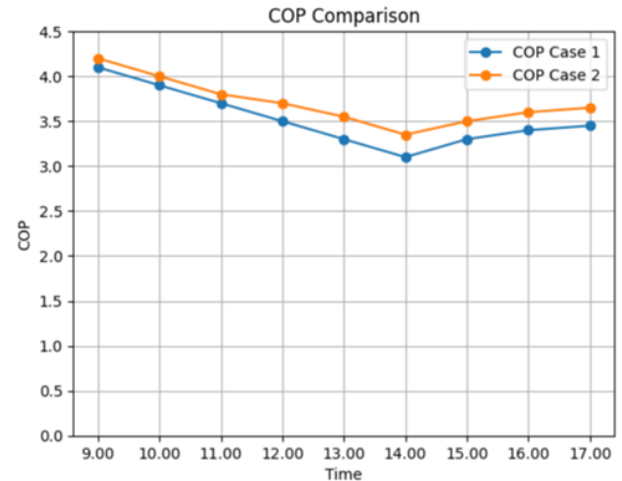


Fig. 8 Comparison of COP values in Cases 1 and 2.

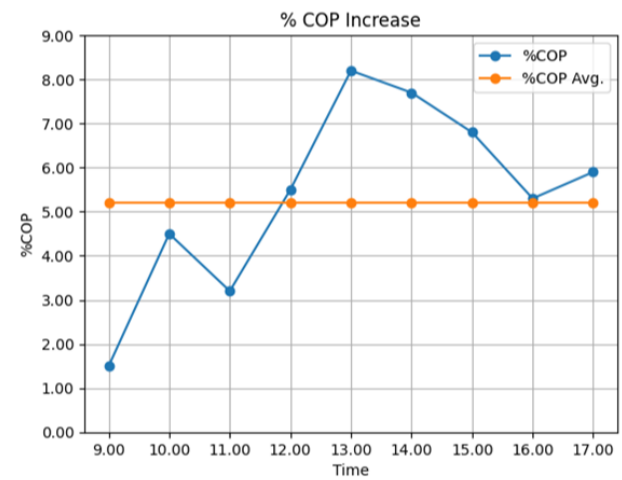


Fig. 9 % COP increase in case 2.

This behavior is expected for vapor-compression air-conditioning systems because higher ambient temperatures reduce the temperature difference available for heat rejection at the condenser, leading to increased condensing pressure and higher compressor power consumption. Consequently, the system operates at a less favorable pressure ratio, which directly lowers efficiency. Despite the overall downward trend, Case 2 consistently maintained higher COP values than Case 1 throughout the entire test period. This indicates that the proposed modification provides a sustained enhancement in condenser-side performance under realistic, time-varying ambient conditions. The improvement became more pronounced during periods of elevated outdoor temperature, when condenser heat rejection becomes increasingly constrained. For example, at 13:00 h, Case 2 achieved a COP of 3.55, exceeding Case 1 by 8.23%. This result suggests that the proposed approach delivers greater relative benefit under peak thermal loading, which is particularly relevant for hot-climate applications where air-conditioning systems experience their highest energy demand. Overall, the average COP improvement achieved by Case 2 was 5.23% over the full operating period. These findings confirm that the proposed configuration can effectively enhance system efficiency and contribute to measurable energy savings under practical daytime operating conditions.

Table 3, the total electrical power input of the air-conditioning system for Case 1 (baseline configuration) and Case 2 (proposed

configuration) during the experimental period from 09:00 to 17:00 h, together with the percentage reduction achieved by Case 2 relative to Case 1. Monitoring power input is essential because it provides a direct measure of energy demand and allows the practical impact of the proposed modification to be quantified under realistic operating conditions. The results indicate that the electrical power consumption of both cases increased from morning toward early afternoon, reaching the highest values between 13:00 and 15:00 h. This period corresponds to elevated outdoor temperatures, which impose a greater thermal load on the condenser and lead to higher condensing pressure. Under such conditions, the compressor operates at a higher pressure ratio and requires more electrical power to maintain the same cooling output. Therefore, the observed trend reflects the expected thermodynamic response of vapor-compression systems operating in hot climates. Despite this increase in demand, Case 2 consistently exhibited lower power input than Case 1 throughout the entire operating period, confirming the effectiveness of the proposed modification in reducing electricity consumption. The percentage reduction varied over time, ranging from 2.03% to 6.19%, with the maximum reduction occurring at 15:00 h, when the system experienced the highest thermal loading. This indicates that the proposed configuration provides greater benefits under peak-load conditions, when conventional systems typically suffer from the most severe efficiency degradation.

Table 3 Total work in put to system (Watt)in case 1 and 2.

Times	$W_{\text{input, Case 1}}$	$W_{\text{input, Case 2}}$	$\%W_{\text{input, Diff}}$
9.00	1,213	1,170	3.54
10.00	1,231	1,206	2.03
11.00	1,285	1,211	5.76
12.00	1,301	1,221	6.15
13.00	1,322	1,248	5.60
14.00	1,330	1,252	5.86
15.00	1,341	1,258	6.19
16.00	1,302	1,245	4.38
17.00	1,280	1,238	3.28
Avg.	1,289	1,228	4.75

Over the full experimental duration, the average power input decreased from 1,289 W in Case 1 to 1,228 W in Case 2, corresponding to a mean reduction of 4.75%. This reduction is significant from an energy-management perspective, as it demonstrates that the proposed approach not only improves performance indices such as COP and EER but also yields measurable electricity savings during the hours of highest cooling demand. Such improvements are particularly valuable for hot-climate applications where peak power consumption strongly influences operating cost and grid loading.

Figures 10 and 11 compare the total electrical power input of the air-conditioning system for Case 1 (baseline) and Case 2 (proposed) over the operating period from 09:00 to 17:00 h. As shown in Fig. 9, the power input in both cases increased from morning to early afternoon, reaching a peak at approximately 14:00–15:00 h, and then gradually decreased toward the late afternoon. This trend corresponds to the rise in outdoor air temperature during the day, which increases condenser heat-rejection demand and elevates compressor workload, thereby increasing power consumption. Notably, Case 2 consistently exhibited lower power input than Case 1 throughout the entire experimental period, confirming that the proposed modification effectively reduces electrical energy use. The difference between the two cases became more pronounced during midday, when outdoor temperatures were highest, indicating

that the proposed system provides greater benefit under elevated thermal stress conditions. Figure 10 shows the percentage reduction in power input achieved by Case 2 relative to Case 1.

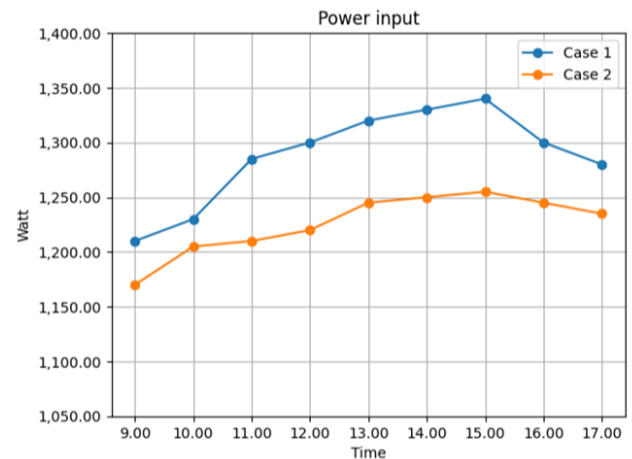


Fig. 10 Comparison of power input values in Cases 1 and 2.

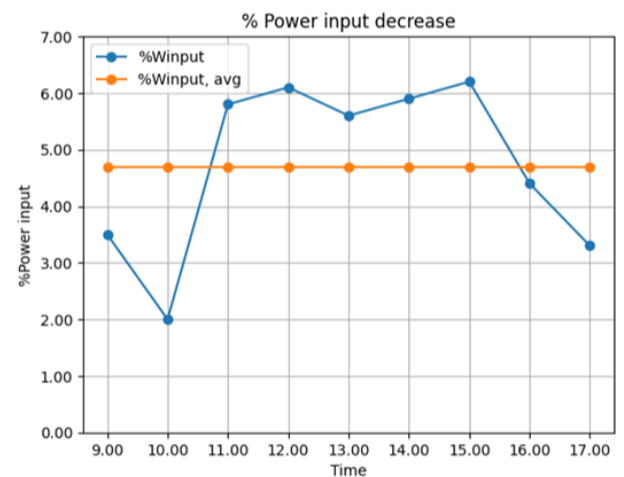


Fig. 11 Power input reduction in Case 2.

The reduction ranged from approximately 2% to 6% over the test period and reached a maximum of 6.19% at around 15:00 h, coinciding with the peak power demand of the baseline system. This behavior suggests that the proposed configuration is particularly effective during peak-load operation, when conventional air-conditioning systems typically experience the greatest efficiency degradation. The average reduction, indicated by the reference line in Fig. 10, was 4.75%, demonstrating a consistent energy-saving effect over the full operating period. Overall, the combined results of Figs. 9 and 10 confirm that the proposed system provides measurable reductions in electrical power input, directly contributing to the improved COP and EER observed in the performance evaluation.

Table 4 presents the variation in energy efficiency ratio (EER) over the experimental period from 09:00 to 17:00 h for both operating cases, while the indoor air temperature was maintained at a constant setpoint of 25 °C. Controlling the indoor condition is important because it ensures that the observed performance differences are primarily attributable to the condenser-side modification rather than variations in cooling demand. At the beginning of the test (09:00 h), the baseline system (Case 1) achieved an EER of 11.59, whereas the proposed system (Case 2) recorded a higher EER of 12.19, corresponding to a 5.19% improvement. As the day progressed and the outdoor air temperature increased toward midday, the EER

values of both cases decreased. This declining trend is consistent with the fundamental thermodynamic behavior of vapor-compression air-conditioning systems, where elevated ambient temperatures reduce the condenser's heat-rejection capacity, increase the condensing temperature and pressure, and consequently raise compressor power consumption. Despite this general reduction in performance, Case 2 consistently maintained higher EER values than Case 1 throughout the entire operating period, demonstrating that the proposed configuration delivers a stable efficiency advantage under realistic and fluctuating outdoor conditions. Notably, the performance gap widened during periods of high thermal stress. For instance, at 13:00 h—when outdoor temperature was near its peak—Case 2 exceeded Case 1 by 14.65%. This result suggests that the proposed technique becomes increasingly effective as the condenser load intensifies, which is particularly relevant for hot-climate operation where conventional systems typically experience the greatest efficiency deterioration. The enhanced performance of Case 2 can be attributed to improved condenser-side heat transfer resulting from the combined use of condensate water and ventilation air for pre-cooling. This approach reduces condenser inlet air temperature, lowers the condensing temperature, and decreases compressor work, thereby improving the overall energy conversion efficiency. Over the full experimental duration, the average EER of Case 2 was 10.69% higher than that of Case 1, confirming that the proposed modification provides meaningful efficiency gains and offers practical potential for reducing electricity consumption during daytime peak cooling demand.

Table 4 Results of EER in Cases 1 and 2.

Times	EERCase 1	EERCase 2	%EERDiff
9.00	11.59	12.19	5.19
10.00	10.67	11.37	6.58
11.00	9.72	10.65	9.59
12.00	9.05	10.17	12.42
13.00	8.47	9.71	14.65
14.00	8.00	9.16	14.40
15.00	8.29	9.44	13.79
16.00	8.88	9.78	10.13
17.00	9.12	9.98	9.44
Avg.	9.31	10.27	10.69

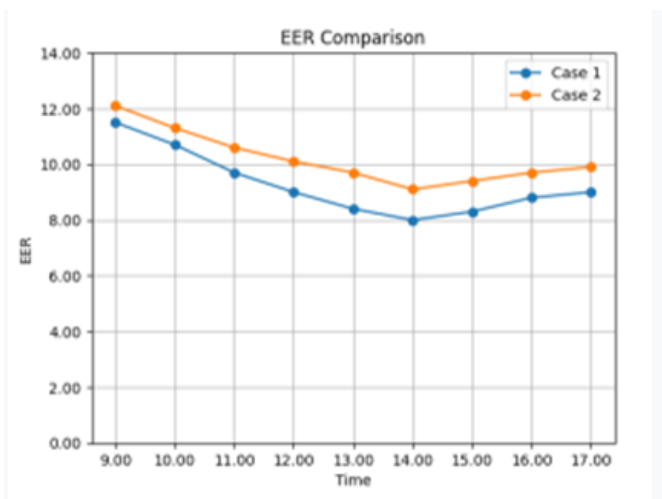


Fig. 12 Comparison of EER values in Cases 1 and 2.

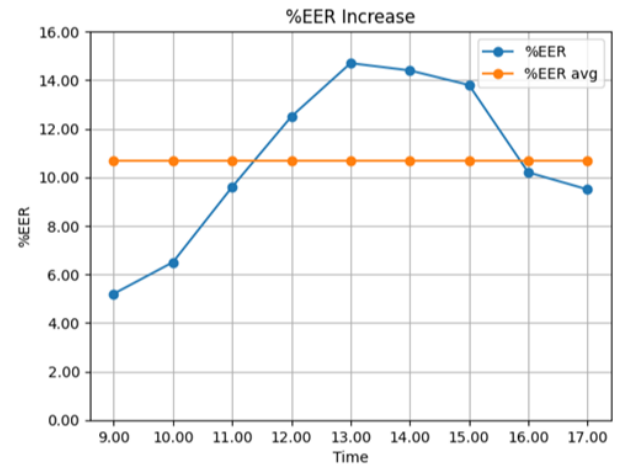


Fig. 13 Percentage increase in EER for Case 2.

Figures 12 and 13 present a comparative time-series analysis of the energy efficiency ratio (EER) for the air-conditioning system during the experimental period from 09:00 to 17:00 h, while the indoor temperature was maintained at a constant setpoint of 25°C. At the start of operation (09:00 h), the conventional configuration (Case 1) exhibited an EER of 11.59, whereas the proposed configuration (Case 2) achieved an EER of 12.19, representing a 5.19% improvement under the same indoor cooling demand. As the experiment progressed from morning to afternoon, the outdoor air temperature increased, leading to a gradual decline in EER for both cases. This trend is consistent with the thermodynamic characteristics of vapor-compression systems, in which elevated ambient temperatures reduce condenser heat-rejection effectiveness, increase condensing pressure, and consequently raise compressor power input, thereby lowering overall efficiency. Nevertheless, Case 2 consistently maintained higher EER values than Case 1 throughout the entire operating period, demonstrating the robustness of the proposed modification under fluctuating outdoor conditions. A particularly pronounced advantage was observed during periods of high thermal stress. For example, at 13:00 h, Case 2 exceeded Case 1 by 14.65%, indicating that the proposed approach becomes increasingly effective when the condenser operates under peak load conditions. This behavior can be attributed to the improved condenser-side heat transfer achieved by utilizing condensate water and ventilation air for pre-cooling, which reduces the condenser inlet air temperature, lowers the condensing temperature, and decreases compressor work. Overall, the results from Figs. 11 and 12 confirm that the proposed system provides meaningful energy-efficiency enhancement under realistic daytime operating conditions. Over the full experimental duration, the average EER of Case 2 was 10.69% higher than that of Case 1, highlighting its potential for reducing electricity consumption, particularly during hot-climate peak hours when air-conditioning systems typically experience their lowest efficiency.

4. Conclusion

This research aims to reduce the energy consumption of an air-conditioned room by improving the performance of the air-conditioning system through the utilization of ventilation air and condensate water. In Case 2, ventilation air and condensate water, which have lower temperatures than the ambient outdoor air, are used to pre-cool the outside air before it enters the condenser. The energy consumption of this proposed system is compared with that of Case 1, which represents a conventional air-conditioning system that uses only outdoor air for condenser cooling. Both of

Case 1 and Case 2 were systematically investigated by measuring and recording key operating parameters, including total electrical energy consumption, refrigerant state variables, indoor and outdoor air temperatures, and ventilation air flow rate. These data were used to evaluate and compare the overall energy performance of the two systems. The experimental procedure began with measuring and recording the ambient air temperature and relative humidity prior to system operation. An electrical energy meter was installed to continuously monitor the power consumption of the air-conditioning system throughout the experimental period. The experiments were conducted under controlled indoor conditions to ensure consistency and reliability of the results. The experimental results indicate that, in Case 1, the average coefficient of performance (COP) of the air-conditioning system was 3.51, and the average energy efficiency ratio (EER) was 9.31. In contrast, for Case 2, the average COP increased to 3.69, while the average EER increased to 10.27. Compared with the conventional system, the proposed configuration resulted in an increase of 5.23% in the average COP and 10.69% in the average EER. These findings demonstrate that the integration of condensate water and ventilation air for pre-cooling the condenser inlet air can effectively enhance the energy efficiency of the air-conditioning system and although the reduction in electricity consumption per unit may not be substantial, widespread implementation of this system could lead to significant overall energy savings. The proposed approach therefore represents a practical and energy-efficient solution for improving air-conditioning performance in air-conditioned spaces. The energy savings obtained in this study, when compared with previous studies reporting a range of approximately 5–30%, are neither the lowest nor the highest. However, the proposed approach demonstrates a balance between energy-saving performance and low investment cost, making it technically and economically appropriate. Furthermore, it shows strong potential for practical implementation to enhance energy efficiency in air-conditioning systems in a cost-effective manner.

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