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Design of an Energy-Efficient Condensing Unit of a Split-Cascade Refrigeration and Air Conditioning System for Office Use

Oginni O.T., Ajewole A.R., Olumilua A.E., Ayeye T.O. and Olagunju M. O.

Abstract The paper aimed to enhance office cooling efficiency by eliminating freezer heat discharge recycling through the design and operation of an energy-efficient condensing unit. The performance of a 200-square-foot room with a 0.352 m^2 evaporator using a total load of 1 kW, five inhabitants, two lamps, a printer, and 1 kg of water was assessed. R407C and R134a refrigerants were chosen for the freezer and air conditioner (A/C) due to their higher quality thermodynamic properties for efficiency testing. Experimental data was collected for 20 days to evaluate the performance of dual systems in terms of refrigerating capacity and COP based on work-input, evaporator, and condenser temperatures. The experiment revealed that the average work input, coefficient of performance (COP), and refrigerating effect in the freezer and air conditioner are 30.29 W and 36.49; 5.96 and 5.23; and 180.3 kJ/kg and 189.4 kJ/kg, respectively. The freezer with a centralized condensing unit had a higher COP of 12.25% compared to A/C, whereas A/C used 16.99% more energy and had a 4.8% cooling effect. The study revealed that a central outside condensing unit significantly reduced hybrid systems' energy consumption, improved cooling effects, and prevented freezer heat recycling and dispersion in the office setting.

I. Introduction

The integration of traditional refrigeration and air conditioning systems into a unified unit allows for the concurrent handling of several cooling tasks. In addition to improving comfort and energy efficiency for a variety of applications, these systems provide continuous cooling and dehumidification capabilities [1]. An expansion valve, condenser, compressor, and evaporator are the four primary parts of the whole system. Although the condenser serves as a heat exchange medium, the compressor is responsible for pressurizing the refrigerant vapour and facilitating effective heat transfer. By transferring heat from interior areas to the outside environment, it helps with cooling while in refrigeration mode. Through condensing water vapour from the air, the air conditioner, for instance, significantly contributes to the extraction of wetness from the air. By absorbing heat from the inside air and condensing fluids, the evaporator cools the space and makes it more pleasant inside. In order to guarantee appropriate refrigerant distribution and expansion, the expansion valve controls fluid pressure, temperature, and expansion, all of which improve system performance and efficiency [2]–[3]. The heat exchanger enhances energy efficiency and comfort by facilitating heat transfer between the refrigerant and the ambient air, hence augmenting the system's capabilities. These elements work together through control systems and algorithms, which include sensors and user-friendly interfaces. This allows for accurate temperature and humidity regulation to satisfy a range of comfort needs [4].

Oginni O.T., Ajewole A.R.., Olagunju M. O.
(Department of Mechanical Engineering, Bamidele Olumilua University of Education, Science and Technology Ikere-Ekiti, Nigeria)
Ayeye T.O. and Olumilua A.E.
(Department of Mechanical Engineering, Federal University Oye Ekiti, Nigeria)

Corresponding Author: oginni.olarewaju@bouesti.edu.ng

A unified air conditioning and refrigeration system saves money and transforms the way that various refrigerants are controlled. This system reduces energy consumption and improves overall comfort in residential, commercial, and industrial settings in addition to providing effective cooling thanks to its wide range of elements and sophisticated mechanisms for regulation [5]. The flexibility of hybrid refrigeration and air conditioning systems to transition between chilling and air-conditioned functions, offering both chilling and dehydrating as needed, is what defines their technique of operation. This ensures comfort in cramped spaces and maximizes system performance. [6]. When the system is in a freezer storage setting, its major objective is to chill interior spaces while preserving consistent temperature levels. The impact of cooling is produced by the chosen method of operation, which combines heat rejection, compressor operation, and evaporator duty. The air conditioning mode sets a priority for both dehumidification and cooling in order to provide the best possible comfort levels.

All refrigeration procedures are included in the mode of operation, along with humidity reduction. The evaporator lowers atmospheric humidity and makes the interior air more pleasant by eliminating moisture from the air. The vapor-compression refrigeration method, which dynamically switches between air conditioning and refrigeration to provide both cooling and dehumidification, is the foundation of a hybrid system mode. [7]-[8]. Residential structures are among the application areas for the integrated system, which simultaneously performs humidity control and cooling functions. Since these are independent mechanisms, there is less space occupied. Combined systems provide effective temperature control solutions in corporate environments, including hotels, retail centers, and workplaces. It is important to consider industrial applications, including food processing, pharmaceuticals, and electronics production, in order to ensure process efficiency and product quality [9].

The undesirable effects of freezer's condenser heat recirculation on office appliances (mal-functional), conditioned space (unsaturated cooling effect) and overall low system cooling efficiency have not been addressed [9]-[10]. The design and operational assessment of a single centralized condensing unit located outside the cooling region for hybrid air conditioning and freezing systems would eliminate the effect of recycled heat on overall efficiency.

II. Materials and Method

A. Materials

Copper connection tubes, a compressor, a condenser, a refrigerating evaporator, an air-conditioning evaporator, a single condensing unit, and building materials are required for the refrigeration and air-conditioning systems. Figures 1, 2, and 3 illustrate the system parts, which were designed with software help and in an exploded format.

B. Design parameters and analysis

The following factors were taken into account when designing the dual systems: In the case of the refrigerator cycle, the refrigerant is R134a, the refrigeration evaporator temperature is T_{re} at -10 ^oC, the air-conditioning evaporator temperature is T_{ae} at -10 ^oC, the evaporator pressure is 0.0399 MPa, the condenser temperature is T_{ch} at 40 ^oC, the condenser pressure is 1.517 MPa, the ambient

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Freezer		1
2	Two in one condensing unit (outside unit)		1
3	Air conditioner		1
4	Connecting tube	copper tube	2
5	Connecting tube	copper tube	4
6	Connecting tube	copper tube	1

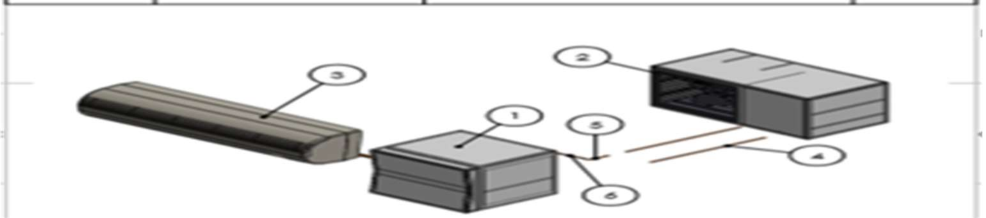

Figure 1: System and Parts Description

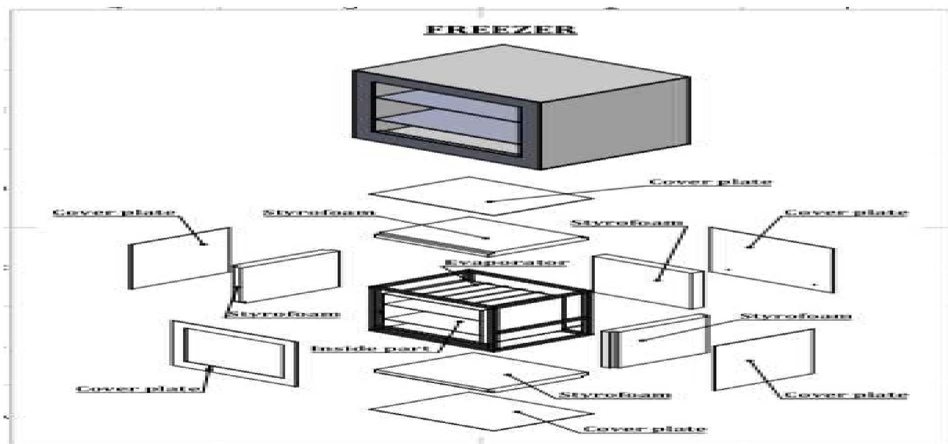

Figure 2: Exploded View of Refrigeration System

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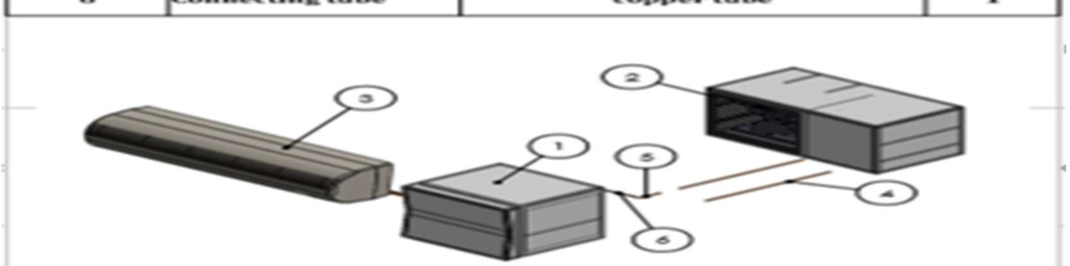

Figure 3: Orthogonal View of the dual Refrigeration System

temperature is T_o at 30 °C, and the and the refrigeration capacity is Ref at 1 kW.

All components function in a uniform state; variations in the kinetic and potential energy of the operating fluids across each part are insignificant; heat loss and pressure drop in the component-piping network are disregarded; and all throttling devices are at constant enthalpy. These design decisions were made in order to streamline the thermodynamic examination of the system.

The thermodynamic tasks at each of the main system components' intakes and outlets were examined using the mass and energy balance equations. Equations 1 and 2 represented the mass and energy relationship equations.

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

$$Q - W + \sum_{in} \dot{m}\, h - \sum_{out} \dot{m}h = 0 \quad (2)$$

Energy balances at the condenser, evaporator, compressor and throttle were contained in equations 3, 4, 5 and 6, respectively.

$$Q_H = \dot{m}(h_{2a} - h_5) \quad (3)$$

$$Q_L = \dot{m}(h_8 - h_7) \quad (4)$$

$$W_{in} = \dot{m}(h_{2a} - h_1) \quad (5)$$

$$h_3 = h \quad (6)$$

The mass balance across the system components is $\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}$

C. Determination of heat load

The four components of the heat load taken into account in the design are the product heat load, the service load, the heat absorbed from the room, and the heat carried across the cooling space's walls.

i. Heat conducted through the walls of the evaporator

The temperature differential between the interior and outside of the box, as well as the walls' thickness, surface area, and thermal resistance, all affect how much heat is transferred through the evaporator's walls. As a result, the following characteristics remain in this design:

The external dimension is 0.55 m by 0.45 m by 0.55 m, and the interior dimension is 0.50 m by 0.40 m by 0.5 m (length x breadth x depth). Surrounding temperature, to is 30 oC, temperature of ice, ti is -10 oC, change in temperature, ΔT = to–- ti is 40 oC, plate material made of stainless-steel sheet has0.001 1m thick, insulating material made of styrofoam is0.05 5m thick, surface area covered by insulator (Ains) is 1.595 m^2, {2[(0.55 x 0.45) + (0.55 x 0.55) + (0.45 x 0.55)]} as found in equation 7.

$$A_{ins} \{[(LxB) + (LxD) + (BxD)] \times 2\} m^2 \quad (7)$$

The rate of heat conduction ($\dot{Q}_{cond}$) through the walls of the refrigerated space is found to be 0.00209 kJ/s on substituting h_i = 9.3 W/m^{2}K, h_o = 22.7 W/m^{2}K, x = 0.05 m and k = 0.033 W/mK and U = 0.0328 in equations 9 and 10 [11]:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{X}{K} + \frac{1}{h_o} \quad (9)$$

$$\dot{Q}_c = \text{UA}\Delta T \quad (10)$$

The quantity of heat conducted, Q_{cond}, through the walls of the evaporator in eight hours of steady operation is estimated as 60.192 kJ as contained in equation 11 [12].

$$Q_{cond} = Q_{cond} \times t \quad (11)$$

ii. Heat gained from the room

The design took into consideration the room's dimensions, the multitude of people using it, the heat produced by gadgets and machinery, and the heat produced by lights, all while accounting for the air conditioning system's precise BTU load. Using equation 12 [13], the area of the room is determined to be 2500 BTU when taking into account an average office size of 200 square feet (20 feet by 20 feet).

$A = 31.25LW$ (12)

Five occupants (O_{cc}) in the office produced 3000 BTU as contained in equation 13.

$O_{cc} = 600N$ (13)

The average number of 2 bulbs of 10 watt was designed for and generated 85 BTU lightning (L_i) as contained in equation 14.

$L_i = 4.25T_{Li}$ (14)

Total equipment heat generated (T_{EH})] in the office with one printer of 250 watt produced 850 BTU as found in 15 and 16

$B_{TUeq} = 3.4T_{eq}$ (15)

$T_{EH} = 3.4PT_{eq}$ (16)

The overall heat load (T_{HL}) entering the air-conditioning system is 17334 KJ.

iii. Product heat load

1kg of water was considered as product load, Q_p of the system. The total heat gained of 245.012 kJ extracted from thermal load of H_a, H_{ab} and H_{bc} in cooling water from 28℃ to 0℃, changing water from 0℃ to freeze ice and further cooling from 0℃ to -5℃ were 117.236kJ, 117.236kJ and 10.54 kJ, respectively as contained in 17, 18, 19 and 20 [14] calculated as follows:

$H_a = m_w c_w \Delta T_w$ (17)

$H_{ab} = m_w L_{fw}$ (18)

$H_{bc} = m_w c_{fw} \Delta T_{fw}$ (19)

$Q_p = H_a + H_{ab} + H_{bc}$ (20)

iv. Service load

The service load of 1% heat calculated from lightning and opening of refrigerator, Q_s was 3.05204 kJ. The aggregate heat load, Q_T was 17642.25kJ per an hour. The system was designed to run steadily for a period of 8 hours, the refrigeration capacity, Q_{Ref} produced 0.612kW as stated in 21.

$Q_T = Q_c + Q_p + Q_s + T_{HL}$ (21)

D. Design of evaporator for the refrigerating system

The refrigerating chamber has a design area of 0.352 m^2 with the coil length 1.3m as found in equation 22. It has 11 turns of coil, N_{te} round the inner part of evaporator, a loading space depth of 0.5m gap between each turn of the coil, G_{coil} 0.045 m as stated in 23 and 24 [15]-[16].

$$L = \frac{A}{\pi D} \quad (22)$$

$$\text{Nte} = \frac{L}{L_{et}} \quad (23)$$

$$G_{coil} = \frac{d}{t} \quad (24)$$

E. Design of evaporator for the air conditioning system

The air conditioning system had an evaporator area of 0.352 m2 and a refrigeration capability of 0.612 kW. The coil has a 0.5 m diameter in one turn and 28 turns around the inner half of the evaporator [17].

F. Design of LTC condenser

R407C at -10ºC, the following parameters were used via inlet pressure, P1 = 0.03139MPa enthalpy, h1 = 408.2kJ/kg, entropy, s1 = 1.8037 kJ/kg-K and temperature, T1 = -10ºC = 268 K. R407C at 40ºC; the exit pressure, P_2 = 0.1517MPa, temperature, $T_2' = (273 + 40)$ K = 313 K

Entropy, $s_2 = s_1 = 1.7551$ kJ/kg-K (isentropic process), h_2 = 452.9 kJ/kg (from table at superheat level), $h_3 = h_{f3} = 263.4$ kJ/kg, $h_3 = h_4$ = 263.4 kJ/kg (Isenthalpic process).

Refrigerating effect, "Re" is found to be 189.5 kJ/kg using equation 25 [18]

$Re = h_1 - h_4$ (25)

The mass flow rate of 0.00322 kg/s was obtained as contained in (26)

$$m_l = \frac{Q_{Ref}}{Re} \quad (26)$$

From density relation, the volume of fluid flow with density ρ_l = 65.448kg/m^3 is 4.91 x 10^{-5} m^3/s. Applying continuity equation, the area was found as 9.665 x 10^{-5} m^2 with refrigerant velocity having 0.508 m/s. Inner diameter of the condenser tube is calculated as 0.123 mm using equation 27. The internal diameter of the tube was so small because of the small load; hence a 6 mm inner diameter is used [19].

$$A = \sqrt{\frac{4A}{\pi}} \quad (27)$$

G. Experimental set-up and procedure

An experimental set-up for the performance evaluation of the air conditioning system as well as the freezer, which consists of indoor and outdoor views as shown in Figures 4, 5, and 6, Two-horsepower compressors were used with the electricity supply. The system was filled with the appropriate refrigerants and ran as designed. The data collection for the experiment was done with the use of temperature and relative humidity measuring instruments. Parameters such as the initial temperature of the conditioned room before reconditioning, the corresponding temperatures of the evaporator and condenser, the conditioned space temperature, the cooling coil, and the conditioned space relative humidity. The measured parameters were utilized to estimate the psychometric properties of the air in the room. Digital dual thermometers were fitted at the inlet and outlet pipes of the compressors to measure the evaporator temperature through a sensor. Similarly, data were collected from the freezer for analysis and comparison.

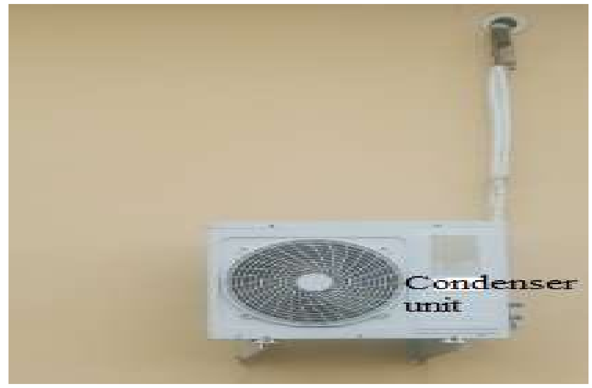

Figure 4: Outside of Central Condensing Unit

Figure 5: Inside of A/C Evaporator Unit

Figure 6: Mini Freezer with Bottle Water

III. Results and Discussion

A. Experimental results

For 20 working days, two hours a day, every two hours in the morning and afternoon, were used to record the observed values of the condenser and freezer temperatures as well as the A/C evaporators. Table 1 shows the labor input into the freezer compared to its refrigerating effects. An average of 30.29 W with 180.3 kJ/kg of input energy was present. The impact of condensing and evaporating temperatures on the system coefficient of performance is summarized in Table 2. The A/C and freezer showed an average of 5.23 and 5.96 COP, respectively. Table 3 presents the results of the average input energy (36.49 W) and cooling effect (189.4 kJ/kg) of A/C. As stated in [20], the findings showed that the use of ejectors on both systems increased the refrigeration effect and COP, which had consequences for improved cooling capacity and less compressor work.

Table 1: Freezer Work-input and Refrigerating Effect

T_e (°C)	Work-input (W)	Ref. Effect (kJ/kg)
30	30.14	180.70
20	30.23	180.50
10	30.22	180.20
0	30.34	180.10
-10	30.52	180.00

Table 2: Temperature Variations of the System on COP

T_e (°C)	T_c (°C)	$COP_{A/C}$ Morning	$COP_{A/C}$ Afternoon	COP_{Freez}
30	42	5.04	5.06	6.00
20	43	5.12	5.13	5.98
10	44	5.23	5.24	5.97
0	45	5.32	5.34	5.94
-10	46	5.41	5.36	5.92

Table 3: Summary of Work input and Refrigerating effect of A/C

T_e (°C)	T_c (°C)	Work-input into A/C Morning (W)	Work-input into A/C Afternoon (W)	Ref. Effect by A/C (kJ/kg)
30	42	37.90	37.80	189.51
20	43	37.10	37.10	189.25
10	44	36.40	36.20	189.25
0	45	35.80	35.70	189.55
-10	46	35.10	35.80	189.43

B. Impact of changes in evaporator temperature on split A/C and freezer COP

Figure 7 presents the relationship between evaporator temperature, freezer temperature, and the A/C coefficient of performance. It shows that as evaporator temperature increases, work input reduces, the refrigerating effect increases, and COP improves as a consequence. The COP determination demonstrated a relationship between the COP and the temperature of the evaporator.

C. Impact of changes in condenser temperature on split A/C and freezer COP

Figure 8 shows the correlation between condenser temperature and COP. A lower condenser temperature increases the refrigeration effect and compressor output work, which in turn raises the system COP. The COP and condenser temperature have a significant correlation, as seen by the COP estimation.

D. Impact of changes in evaporator temperature on system work-input and refrigeration effect

As indicated by Figure 9, which depicts the relationship between evaporator temperature and work input, the system's refrigerating impact increases as work input rises. The cooling effects on refrigeration devices are thus directly correlated with the temperature of the evaporator and the energy input.

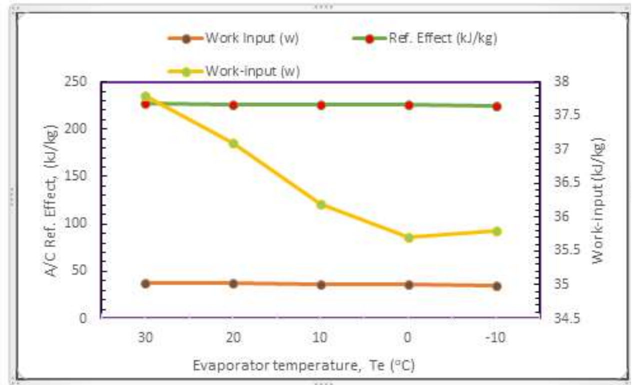

Figure 9: Effect of evaporator temperature on A/C work-input and cooling

E. Impact of changes in condenser temperature on system cooling capacity

The relationship between condenser temperature and refrigerating effect, as seen in Figure 10, suggests that when condenser temperature rises between 42 and 46 °C, refrigerating capacity rises as well.

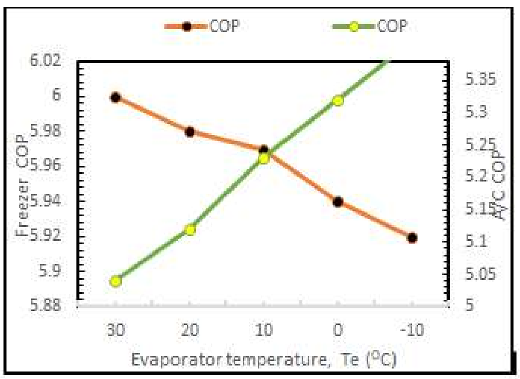

Figure 7: Effect of evaporator temperature on A/C and Freezer COPs

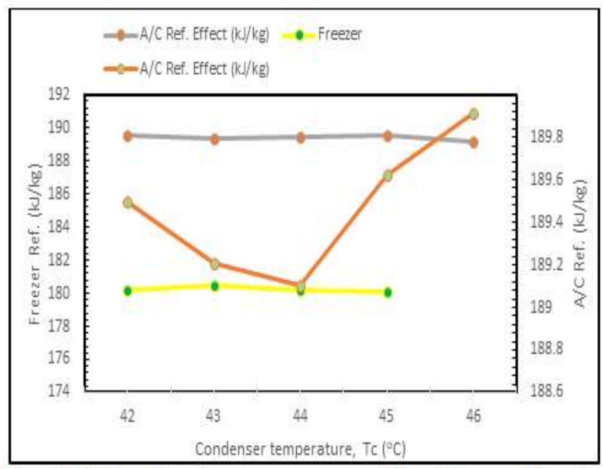

Figure 10: Effect condenser temperature on A/C freezer and refrigeration

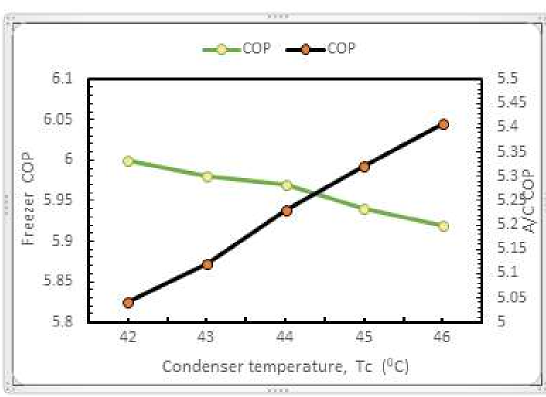

Figure 8: Effect of condenser temperature on A/C and Freezer

IV. Conclusion

A vapor compression cascade refrigeration system with a central condensing unit was designed, built, tested, and verified to improve energy efficiency and lower component costs for sustainability. The dual systems' performance evaluation revealed that the A/C system had an average of 5.22 and 5.23 in the morning and afternoon, while the freezer had 5.96. The A/C system consumed an average of 36.46 kJ/kg and 36.52 kJ/kg in the morning and afternoon, respectively, while the freezer consumed 30.29 kJ/kg. The overall refrigerating effect of both systems resulted in a 9.1 kJ/kg difference in favor of the A/C system. It was determined that the single condensing unit's assistance in reducing overall energy consumption and the system's design and performance totally eradicated the freezer's heat discharge recirculation. In addition to eliminating heat discharge recirculation by a refrigerator condenser within an office, this device reduced energy consumption and greenhouse gas emissions, improving ease of use and efficiency for residential, commercial, and industrial applications. Compared to standard standalone systems, the integrated system has several benefits, including a more room-efficient design, energy-efficient operation, comfort, cost savings, and scalability to accommodate different situations.

Nomenclature

Symbol	Meaning	Unit
U	Overall heat transfer coefficient	kW/m^2K
A	Surface area of the insulator	m^2
$\Delta T = t_o - t_i$	Change in temperature	^oC
$\dot{Q}_c$	Rate of heat transfer by conduction	W
D	Diameter of copper pipe	m
V	Volumetric flow rate	m^3/s
X	Thickness of copper tube	m
K	Thermal conductivity of insulator	kW/mK
h_i	Inside convection coefficient	kW/m^2K
h_o	Outside convection coefficient	kW/m^2K
T	Time	s
m_w	Total mass of loaded water	kg
C_w	Specific heat capacity of water	J/kgK
ΔT_w	Temperature change in cooling water to 0^{oC}	0^oC
L_w	Latent heat of water changing to frozen ice	kJ/kg
C_{fw}	Specific heat of frozen ice	kJkg.K

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