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Design and Testing of a Refrigeration System with a Heat Recovery Device for Energy Saving

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Abstract: To improve energy efficiency in industrial refrigeration systems, a heat recovery device was designed and installed on a model system. Based on theoretical analysis, the system's performance was evaluated under two operating conditions: with and without heat recovery. After the model was fabricated, experiments were conducted to measure inlet and outlet parameters over specific time intervals and operating conditions. The collected data were analyzed and compared using simulation methods to identify key factors influencing performance. Results showed that incorporating a heat recovery device increased the coefficient of performance (COP) from 3,0 to 3,44 and reduced the actual ice-making time by approximately 15 minutes compared to the system without heat recovery. This demonstrates a significant reduction in power consumption, with experimental results consistent with theoretical predictions.

Keywords: Refrigerant, Temperature, Pressure, Heat recovery device, Experimental results

I. Introduction

In the current context, energy conservation has become one of the most urgent issues, not only at the level of individual localities and cities but also on a global scale. In Vietnam as well as around the world, research and technological applications aimed at saving energy and improving energy efficiency are being prioritized and promoted [1]. The common goal is to identify effective, feasible energy-saving solutions suitable for Vietnam's practical conditions. This topic has attracted special attention from many scientists and engineers across the country.

In daily life and industrial production, sectors such as air conditioning, hot-water heating systems, food cold storage, and ice production plants consume significant amounts of electrical energy [2]. Therefore, the development of refrigeration systems that use energy efficiently and economically is a crucial and strategic factor for the sustainable development of the national economy in particular, and for the global economy in general.

From this reality, the author conducted research, theoretical calculations, and experimental model fabrication of a refrigeration system equipped with a heat recovery device. The objective was to evaluate the system's power consumption in two cases: with and without the heat recovery device. Based on the obtained results, the author proposes optimal solutions to improve system performance, enhance energy savings, and maintain operational safety and reliability.

II. Research Methodology

In this paper, the author conducts a theoretical study on a refrigeration system equipped with a heat recovery device and simultaneously develops an experimental model to verify the calculated results. After performing measurements, analyses, and evaluations of the experimental data, the author compares the findings with the theoretical model to determine the level of agreement, deviations, and energy efficiency of the system. Based on these comparisons, specific conclusions and recommendations are proposed regarding the system's potential for practical application.

2.1. Modeling of a single-stage vapor compression refrigeration cycle

The heat recovery cycle shown in Figure 1 features a heat exchanger installed between the high-pressure liquid refrigerant exiting the condenser (before entering the expansion valve) and the low-pressure vapor refrigerant leaving the evaporator (before being drawn into the compressor). The superheating (q_{qn})and subcooling (q_{ql}) processes occur within the heat recovery device (HRD).

To calculate the refrigeration cycle and select a suitable compressor for the system, it is necessary to determine the typical operating processes and the arrangement of components within the system [2, 3]. As illustrated in Figure 2, the processes occur sequentially as follows:

1'-1: Superheating of the vapor occurs in the heat recovery device;

1-2: Adiabatic compression of the vapor from low pressure and low temperature to high pressure in the compressor;

2-2': Isobaric cooling of the superheated vapor to saturated vapor in the condenser;

2'-3: Isothermal and isobaric condensation in the condenser, releasing heat to the surroundings;

3-3': Subcooling of the high-pressure liquid occurs in the heat recovery device;

3'-4: Isenthalpic throttling of the refrigerant through the expansion valve;

4-1': Isothermal and isobaric evaporation at low temperature and low pressure, absorbing heat from the refrigerated space.

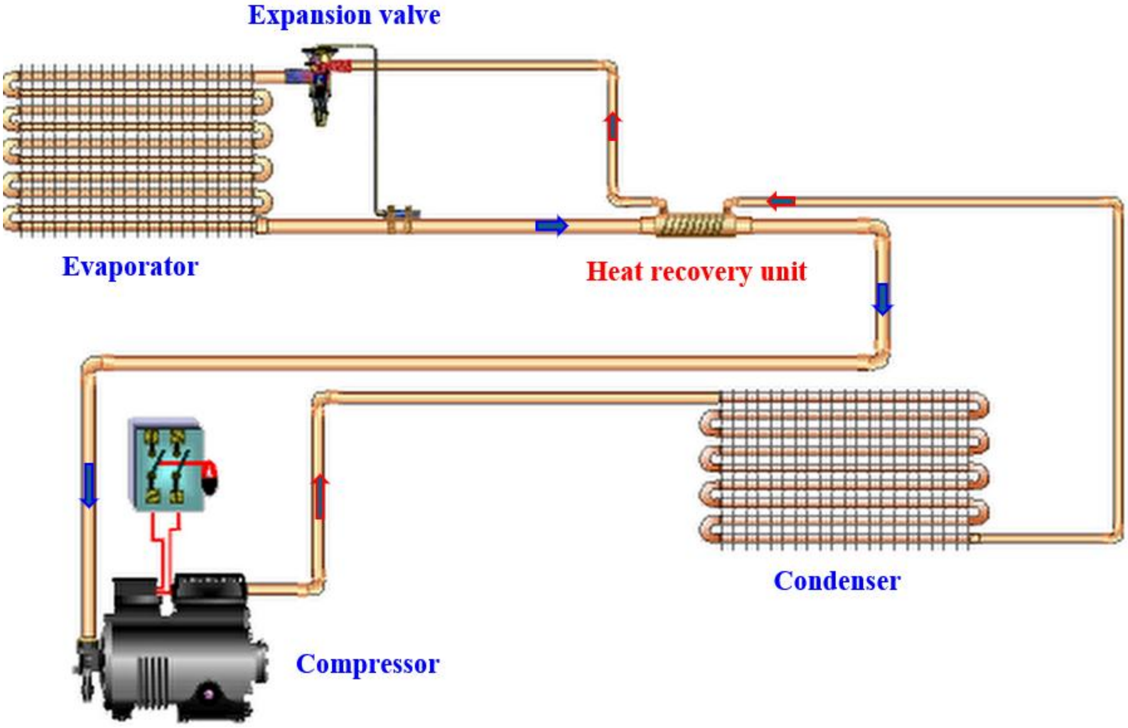

Figure 1: Schematic diagram illustrating the operating principle of the refrigeration system with an added heat recovery device.

2.2. Calculation of the cycle parameters

To evaluate the effectiveness of the refrigeration system model with the added heat recovery device (HRD), the author models the system's energy exchange using capacities and input values to meet the requirements for designing and fabricating the prototype. The required specifications are as follows: compressor cooling capacityQ_0 = 250 W; selected condensing temperature t_k = 40^0C (p_k = 10 bar), evaporation temperaturet_0 = -15^0C (p_0 = 1,64), super heat temperature t_{qn} = 10^0C;the working refrigerant is R134a; and the system operates stably and safely.

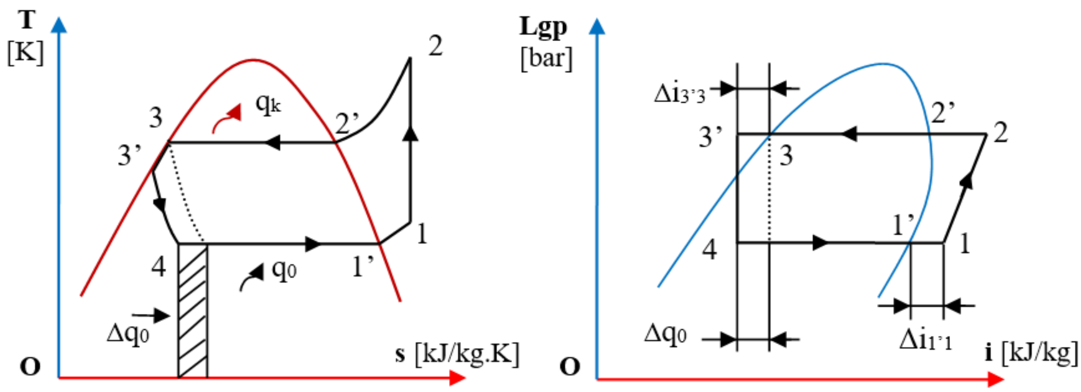

Figure 2: The T - s and Log p - i diagrams illustrate the nodal points of the refrigeration cycle

Heat balance at the heat recovery device [4] gives: $\Delta i_{ql} = \Delta i_{qn}$ meaningi_3 -$i_{3'}$ = i_1 - $i_{1'}$.Since the heat released by the liquid refrigerant $Q_h = m.c_{ph}.\ \Delta t_h$ equals the heat absorbed by the vapor refrigerant $Q_h = m.c_{ph}.\ \Delta t_h$ (neglecting environmental losses) and the mass flow rates are the same, we obtain $c_{pl}.\Delta t_l = c_{ph}.\ \Delta t_h$. Here, m is the circulating refrigerant mass flow rate (kg/s); $\Delta t_l$is the temperature difference of the low-pressure vapor at the inlet and outlet. Moreover, because the specific heat of the liquid c_{pl} is much greater than that of the vaporc_{ph}, it follows that$\Delta t_1 < \Delta t_h$. Therefore, the minimum temperature approach hΔt_{min} of the exchanger occurs at the upper end, as shown in Figure 4.According to [3, 4], $\Delta t_{min} = t_3 - t_1 \geq 5$K,which is one of the design inputs for sizing the heat recovery device in the heat-recovery cycle.

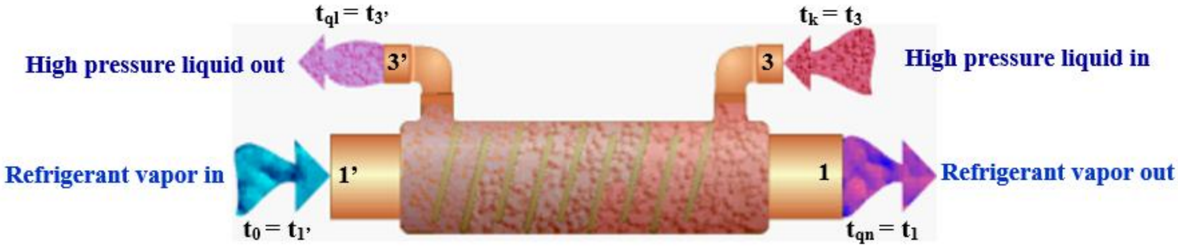

Figure 3: Flow diagram of the refrigerant through the heat recovery device

Based on the Log p - i diagram of refrigerant R134a in Appendix [6] and the given temperatures, the state parameters at the cycle's nodal points are determined as shown in the following table.

Bảng 1: State properties at the cycle's nodal points for R134a.

Nodal point	1'	1	2	3	3'	4
Temperature, ^{0}C	-15	10	72	40	35	-15
Pressure, bar	1,64	1,64	10	10	10	1,64
Enthalpy, kJ/kg	690	710	755	555	535	535

In evaluating the system's performance, the related quantities are calculated according to [1, 5] as follows:

- Specific cooling capacity of the refrigeration cycle: $q_0 = i_{1'} - i_4 = 690 - 535 = 155$ (kJ/kg) ;
- Specific compression work: $l = i_2 - i_1 = = 755 - 710 = 45$ (kJ/kg) ;
- Specific condenser heat rejection capacity: $q_k = i_2 - i_3 = 755 - 555 = 200$ (kJ/kg) ;
- Theoretical coefficient of performance (COP) with heat recovery: $\varepsilon = \frac{q_0}{l} = \frac{155}{45} = 3{,}44$;
- Theoretical coefficient of performance (COP) without heat recovery: $\varepsilon' = \frac{q'_0}{l} = \frac{135}{45} = 3{,}0$;
- Mass flow rate of vapor drawn into the compressor: $m = \frac{Q_0}{q_0} = \frac{0{,}25}{155} = 0{,}00161$ (kg/s) ;
- Specific heat transfer capacity in the heat recovery device: $q_{hn} = i_1 - i_{1'} = 710 - 690 = 20$ (kJ/kg) ;
- Heat released in the condenser: $Q_k = m.\ q_k = 0{,}00161.200 = 0{,}322$ (kW) ;
- Heat released in the heat recovery device: $Q_{hn} = m.\ q_{hn} = 0{,}00161.10 = 0{,}032$ (kW).

It should be noted that subcooling occurring in the heat recovery device increases the coefficient of performance (ε); the calculated result shows an improvement of about 13% (represented by the hatched area on the T–s diagram) compared to the case without heat recovery. In other words, it enhances the exergy efficiency of the system. Meanwhile, superheating slightly increases the compression work but ensures safe compressor operation, as the vapor entering the compressor no longer contains liquid droplets that could cause hydraulic shock.

Air flow rate required for condenser cooling: $Q_k = m_{kk}.C_p.(t''_{kk} - t'_{kk})$; with $C_p = 1$ kJ/kg.K - Specific heat capacity of air at constant pressure; If we select$t'_{kk} = 25^0$C and $t''_{kk} = 35^0$C - the inlet and outlet air temperatures of the condenser, then it follows that: $m_{kk} = \frac{Q_k}{C_p\,(t''_{kk} - t'_{kk})} = \frac{0{,}322}{1(35-25)} = 0{,}0322$ (kg/s), this provides the basis for selecting a suitable condenser fan;

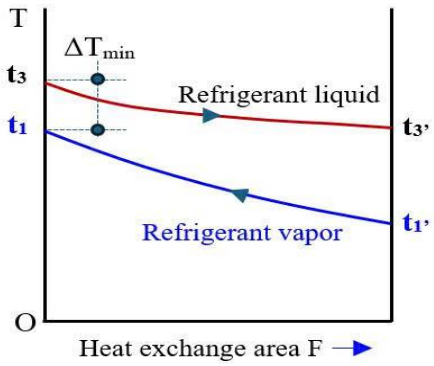

Figure 4: Temperature variation in the heat recovery device

- Heat exchange area of the condenser [2, 5]: F_k =

$\frac{Q_k}{q_k} = \frac{Q_k}{k_k.\Delta t_k}$; ($m^2$). For a forced-air condenser, the refrigerant vapor flows through the condenser tubes and releases heat to the ambient air blown across the finned outer surface by a fan. The overall heat transfer coefficient and the logarithmic mean temperature difference (LMTD) of the condenser are selected according to: [2, 4]: $\Delta t_k = 8^0K$; $k_k = 30$ W/m^2.K; the calculated result is $F_k = 1,34$ m^2;

Heat exchange area of the evaporator [2, 5]: $F_0 = \frac{Q_0}{q_0} = \frac{Q_0}{k_0.\Delta t_0}$; ($m^2$). For a forced-air evaporator, the refrigerant vapor flows through the evaporator tubes and absorbs heat from the space to be cooled, while a fan blows air across the finned outer surface of the tubes. The overall heat transfer coefficient and the logarithmic mean temperature difference (LMTD) of the evaporator are selected according to [2, 4]: $\Delta t_0 = 10^0K$; $k_0 = 12$ W/m^2.K; the calculated result is $F_0 = 2,08$ m^2.

III. Experimental Model

After completing the theoretical calculations, the author fabricated an experimental model of the refrigeration system for ice production, as shown in Figure 5, with the following technical specifications:

- Hermetic compressor with a cooling capacity of 250 W using R134a refrigerant;
- Condenser with a heat exchange area of 1,5 m² and a forced-convection cooling fan;
- Evaporator with a heat exchange area of 2,0 m² and a forced-convection cooling fan;
- Automatic expansion valve capable of adjusting according to the actual system pressure;
- Temperature relay, ammeter clamp, pressure gauges, and other auxiliary instruments.

Figure 5: Experimental model of the refrigeration system equipped with a heat recovery device

All equipment parameters were appropriately selected based on the initial data, ensuring stable and safe operation. The system was equipped with control valves, allowing it to operate in two different modes. The thermal load consisted of 2,5 kg of water contained in ice molds, with an initial water temperature of 27°C and a final ice temperature of -7°C. The expected freezing time for ice cubes [3] ranged from 2.5 to 3 hours, depending on the selected mold size.

System operation process: Low-pressure vapor generated in the evaporator is drawn into the compressor, where it is compressed to a high pressure (p_k) and high temperature (t_k), then discharged into the condenser. In the condenser, the refrigerant releases heat to the cooling medium, which is air moved by a forced convection fan, and condenses into a high pressure liquid (p_k). The liquid then passes through the expansion valve, where its pressure drops from high (p_k) to low (p_0), becoming a low pressure liquid with a low temperature (t0) before entering the evaporator. Inside the evaporator, the refrigerant absorbs heat from the ice tank, evaporates, and depending on the operating mode, either passes through the heat recovery device or bypasses it before being drawn back into the compressor. The cycle is closed and continues to repeat.

Figure 6: Ice cube trays and experimental measuring instruments

The experimental measurements were recorded simultaneously at various times throughout the day to ensure that the results accurately reflected the real operating conditions of a refrigeration system. The data are presented in the graphs shown in Figures 7 and 8. The experimental measuring instruments and the ice molds after the test run are illustrated in Figure 6.

3.1. Variation of ice water temperature over time in the two cases

The system operated stably in both modes: without heat recovery and with heat recovery. These values were continuously measured throughout the experiment and are shown in Figure 7.

When the system starts operating, the water temperature decreases rapidly because the initial temperature of the water in the ice molds is relatively high at 27°C, and most of the heat released from the water is absorbed by the refrigerant. As the water temperature drops and approaches the evaporation temperature of the refrigerant, the amount of heat absorbed by the refrigerant in the evaporator begins to decrease [7, 8]. As the ice water temperature continues to fall, the heat exchange process slows down due to the smaller temperature difference between the water and the refrigerant. However, when the ice temperature reaches -7°C, the temperature relay automatically cuts off the power supply, stopping the compressor and ending the system's operation. Experimental results show that, under the heat recovery mode, the freezing time was shortened by approximately 15 minutes compared to the mode without heat recovery.

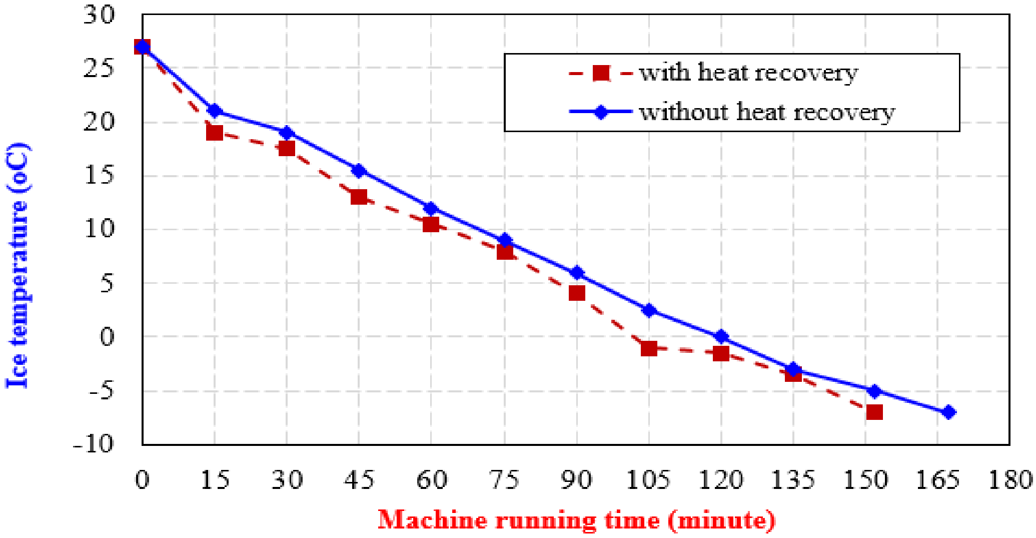

Figure 7: Variation of ice water temperature over time in the two operating modes

3.2. Variation of electric current during operation in the two operating modes

In the case where the refrigerant does not pass through the heat recovery device (HRD), the electric current remains relatively stable throughout the operation of the refrigeration system. When the refrigerant passes through the HRD, the current initially increases slightly at system startup due to the additional flow resistance in the piping, which causes a small rise in compression work [9]. However, after reaching steady-state operation, the differences in pressure and current between the two modes are minimal, indicating that the installation of the heat recovery device does not significantly affect the system's current draw. Furthermore, during system operation, the electric current exhibits minor fluctuations. This can be explained by variations in ambient temperature, which cause changes in the condensation temperature and pressure at the condenser, leading to corresponding changes in compressor work. The experimental results shown in Figure 8 indicate that the variations in current and pressure in both cases remain relatively stable, with no significant differences observed.

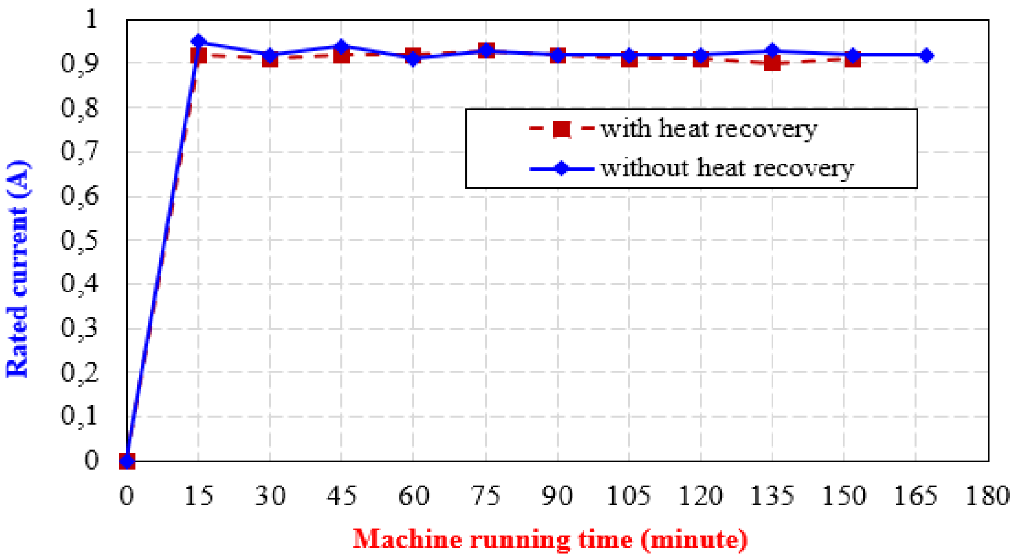

Figure 8: Representation of the variation of electric current over time in the two operating modes

IV. Conclusions

From the theoretical calculations and experimental results obtained from the model, the author has drawn the following conclusions:

- It should be noted that subcooling within the heat recovery device always increases the cooling coefficient or improves the exergy efficiency, while superheating slightly increases the compression work but ensures safe compressor operation, as the vapor entering the compressor no longer contains liquid droplets that could cause hydraulic shock in the cylinder. Therefore, the heat recovery cycle is applied only to refrigeration systems using Freon refrigerants [9, 10], where it demonstrates clear effectiveness (not applicable for NH_3 systems).
- When the system is equipped with a heat recovery device (HRD), the ice water temperature decreases more rapidly; the coefficient of performance (COP) increases from 3,0 to 3,44; the electric current variation is negligible; and the operating time of the system is shortened by approximately 15 minutes (from 167 minutes in the non-recovery mode to 152 minutes in the heat recovery mode). This indicates that the electrical energy consumption during the operation of the refrigeration system is significantly reduced.
- The addition of a heat recovery device requires a considerable increase in the initial investment cost. However, in the long term, this proves highly beneficial, as it not only saves electrical energy but also ensures safe system operation by preventing hydraulic shock, as mentioned above;
- The calculation of the parameters for the heat recovery device is quite complex; therefore, based on practical experience, the author referred to [2, 3] when designing the model to select appropriate equipment parameters that ensure consistency between the theoretical study and the experimental setup. This experimental model is retained for research and educational purposes, serving as a practical training tool for students majoring in Refrigeration and Thermal Engineering.
- The results presented above are limited to a small-capacity experimental refrigeration system model. Therefore, in future work, the author plans to extend the research to various components of industrial refrigeration systems and central air-conditioning systems in order to further evaluate the effectiveness and benefits of incorporating a heat recovery device into such systems.

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