International Journal of Recent Engineering Research and Development (IJRERD)
ISSN: 2455-8761
www.ijrerd.com || Volume 06 – Issue 03 || March 2021 || PP. 18-23

A Study on Installation of a Heat-Recovery Device in The Refrigeration System

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Abstract: In this work, a design of a refrigeration system with heat recovery is presented. Both theoretical and experimental approaches are employed to analyze and compare the energy efficiency of using and not using the heat-recovery device. Input and output parameters are collected under a specific condition and time-length. Results show that, compared to a normal system, the one with heat recovery produces a significantly greater cooling coefficient of performance. In detail, the coefficients for the former and the latter are, respectively, 2,1 and 3,45. Moreover, this leads to a considerable reduction in power consumption when heat-recovery is utilized.
Keywords: Heat-recovery device; refrigeration system; overheated air, energy-saving

1. Introduction

In recent years, energy-saving, which ensures sustainable development, has become one of the most important issues in not only Vietnam but also all around the world [1]. Indeed, this topic has been drawing attention to many researchers and scientists to reduce energy costs in various engineering fields. In developed countries, many efforts have been spent on the improvement of energy-saving devices/equipment on refrigeration and air conditioning, which consumes a large amount of power for our necessary activities, e.g., air-cooling and purification or food storing. In Vietnam, however, there still exist very few studies on this field despite the fact that for a tropical country, refrigeration systems become much more important. Amongst them, some investigations were conducted to recover heat released from the condenser of a small-capacity air conditioner; results showed that the heat-recovery process negligibly affects the stable operation of the systems. Furthermore, in our previous study [2], we designed to take advantage of waste heat by adding an air dryer and a water heater in order to increase the energy efficiency of the boiler. Nevertheless, for the refrigeration systems, no available experimental study on the addition of a heat-recovery device can be found. Therefore, in this work, we aim to design and build a refrigeration model with a heat-recovery device to estimate the power consumption and make an extensive comparison with the classical system. From this, energy-saving solutions can be provided and analyzed in detail.

This paper is organized as follows: an introduction including a description of working diagrams and determination of necessary quantities is presented in Section 1; the experimental method is introduced in Section 2; Section 3 provides results and discussions, and finally, some concluding remarks and recommendations for future work are given in Section 4.

1.1. Working and state diagrams

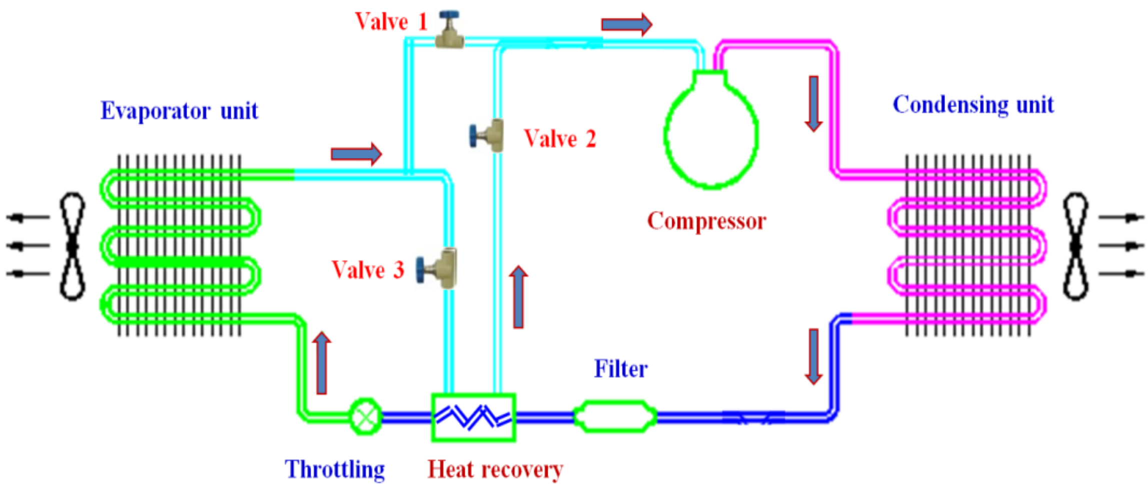

Figure 1. Schematic diagram of a refrigeration system with a heat-recovery device

It is good to be noticed that the most important improvement of the refrigeration with heat-recovery is the available heat exchange between the high-pressure liquid (after condensing and before going through the throttle) and low-pressure gas of refrigerant (before returning to the compressor) (see Fig. 1). Additionally, the overheating (q_{qn}) and subcooling (q_{ql}) are performed in the heat-recovery device [2].

As can be seen in Fig. 2, operating processes required to design a refrigeration system are, in turn, happened as follows: 1'-1: gas overheating in the heat-recovery device; 1-2: adiabatic compression of low-pressure gas; 2-2': cooling superheated gas to saturated gas; 2'-3: isothermal and isobaric condensation in the condenser and heat release to the outside environment; 3-3': liquid subcooling in the heat-recovery device; 3'-4: gas throttling through the throttle from the high pressure to low pressure; 4-1': isothermal evaporation at low temperature and low pressure and heat absorption of the environment that needs to reduce the temperature.

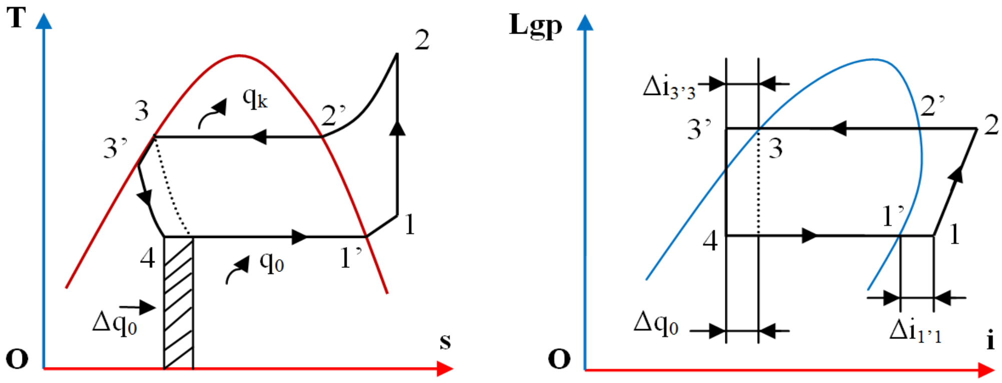

Figure 2. T-s and Lgp-i diagrams presenting nodal points

1.2. Determination of necessary quantities

In order to evaluate the efficiency of the refrigeration system with the heat-recovery, the modeling of energy exchange is carried out by the productivity and input values required for design and manufacturing. The key parameters are as follows: cooling capacity Q_0 = 0,185 kW; condensation temperature $t_k = 40^0$C; boiling temperature $t_0 = -15^0$C; superheat temperature $t_{qn} = -10^0$C; the chosen refrigerant is R134a; the stabilization and safety of this system during operation are ensured.

Considering the heat balance in the heat-recovery device in Fig. 3, we have: $\Delta i_{ql} = \Delta i_{qn}$ means $i_3 - i_{3'} = i_1 - i_{1'}$. It is noted that the heat generated by the refrigerant liquid $Q_1 = m.c_{pl}.\Delta t_1$ is equal to the one collected by refrigerant gas $Q_h = m.c_{ph}.\Delta t_h$ (heat loss to the outside environment is assumed to be negligibly small); moreover, the flow rate is preserved, leading to $c_{pl}.\Delta t_1 = c_{ph}$. Here, m is the mass flow rate of the refrigerant (kg/s); Δt_1 is the temperature difference between inlet and outlet low-pressure gas. Furthermore, since the heat capacity of a liquid c_{pl} is larger than that of a gas c_{ph}, $\Delta t_1 < \Delta t_h$. It is good to be aware that $\Delta t_{min} = t_k - t_{qn} \geq 5^0$C [1,4, 5].


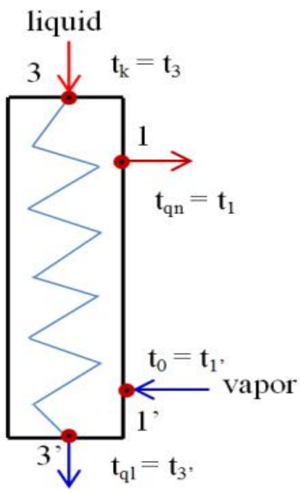

Figure 3. Heat recovery-device and refrigerant flow path in a heat-recovery device

According to Lgp-i plot of R134a [6] and determined values of temperature, state parameters at each nodal point are provided in Table 1.

Table 1. State parameters at each nodal point for R134a

Point	**1’**	**1**	**2**	**3**	**3’**	**4**
Temperature, ^{0}C	-15	-10	72	40	35	-15
Pressure, bar	1,6	1,6	10	10	10	1,6
Enthalpy, kJ/kg	690	710	755	555	535	535

Refrigeration parameters are estimated as in [7,8] and are introduced as follows:

- Specific refrigerating effect of the circle: $q_0 = i_{1'} - i_4 = 155$ (kJ/kg);
- Specific condensation heat capacity: $q_k = i_2 - i_3 = 200$ (kJ/kg);
- Specific compression work: $l = i_2 - i_1 = 45$ (kJ/kg);
- Cooling coefficient of the circle: $\varepsilon = \frac{q_0}{l} = 3{,}45$;
- Specific heat capacity at the heat-recovery device: $q_{hn} = i_1 - i_{1'} = i_3 - i_{3'} = 20$ (kJ/kg);
- Mass flow rate of the gas sucked in the compressor: $m = \frac{Q_0}{q_0} = 0{,}0012$ (kg/s);
- Thermal condensation load: $Q_k = m.\ q_k = 0{,}0012.200 = 0{,}24$ (kW);
- Air mass flow rate required to cool the condenser: $Q_k = m_{kk}.C_p.t(t''_{kk} - t'_{kk})$; with $C_p = 1$ kJ/kg.K being the air isobaric specific heat capacity; $t'_{kk} = 25^0$C and $t''_{kk} = 35^0$C being, respectively, the air temperature at the condenser’s inlet and outlet. Therefore, we have: $m_{kk} = \frac{Q_k}{C_p(t''_{kk} - t'_{kk})} = \frac{0{,}24}{1(35-25)} = 0{,}024$ (kg/s);
- Heat exchange surface of the condenser: $F_k = \frac{Q_k}{q_k} = \frac{Q_k}{k_k.\Delta t_k}$; (m^2). For the forced-air cooled condenser, the

refrigerant flowing in the tubes condenses and releases heat to the air circulated by a fan. The heat transfer coefficient and the log mean temperature difference of the condenser are, respectively, $\Delta t_k = 8^0$K and $k_k = 30$ W/m^2.K [2,4]. From this, the heat exchange surface can be estimated to be $F_k = 1{,}0$ m^2.

- Heat exchange surface of the evaporator: $F_0 = \frac{Q_0}{q_0} = \frac{Q_0}{k_0.\Delta t_0}$ (m^2). For the forced-air convection evaporator,

the refrigerant gas flows and then evaporates due to heat received from the surrounding environment. The heat transfer coefficient and the log mean temperature difference of the evaporator are, respectively, $\Delta t_0 = 10^0$K and $k_0 = 12$ W/m^2.K [2,7]. From this, the heat exchange surface can be estimated to be $F_0 = 1{,}5$ m^2.

2. Experimental Model

2.1. Experimental parameters

Based on the aforementioned results from the theoretical basis, we propose an experimental refrigeration model freezing water (see Fig. 4); its components are introduced as follows:

- A sealed compressor using R134a; its refrigerating capacity is 185W;
- A condenser with the heat exchange surface being of 1,0 m^2; forced-air convection fan is employed;
- A evaporator with the heat exchange surface being of 1,5 m^2; forced-air convection fan is employed;
- An automatically adjustable throttle valve;
- Electric control system and other auxiliary devices.

It is good to be aware that all of the technical specifications of each device have been confirmed to be reasonable with the initial calculations; furthermore, the working process is seen to be reliable and steady. The thermal load is from 2kg water contained in ice trays. The temperature of the supplied water and the freezing ice are, respectively, 26^{0}C and -6^{0}C. The freezing time is predicted to be around 2 ÷ 3 hours; this value is noted to depend on the chosen size. Furthermore, to improve the energy-saving task, a solar panel is installed to electricity (through a converter) provided to the fans.

ISSN: 2455-8761
www.ijrerd.com || Volume 06 – Issue 03 || March 2021 || PP. 18-23

Figure 4. Experimental model of a refrigeration system freezing water

2.2. Working principle of refrigeration system

The working principle of a refrigeration system is presented as following: low-pressure refrigerant gas generated from the evaporator is sucked and then pressed to high pressure (p_k) and high temperature (t_k) to become the overheated gas. After that, this so-called superheated gas is fed to the condenser; in there, the refrigerant releases heat to the cooling environment (forced-air from the fan) and then becomes a high-pressure refrigerant liquid (p_k). Under the action of a throttle valve, the pressure of the liquid is decreased from a high value (p_k) to a low one (p_0), it is then changed to a low-temperature (t_0) liquid and come to the evaporator. At the evaporator, the refrigerant receives heat from the heat sources inside the ice-maker blocks, evaporating, and can then go through a heat-recovery device (or not) before being sucked again by the compressor. The process is closed and repeated automatically.

Figure 5. Ice rays and measuring devices

3. Results and Discussions

In this part, experimental results are reported and analyzed. It is noteworthy that the experiments are carried out at various times of a day to ensure the realistic representation of the working process of a refrigeration system. It is noted that we use the following measuring devices during performing experiments: ice rays, pressure gauge, ampere pliers, and temperature relay (see Fig. 5).

3.1. Temperature variation

The system is in the mode without heat recovery (valve 1 opens, valves 2 and 3 closes) and in the mode with heat recovery (valve 1 closes, valves 2 and 3 open). These experimental values are obtained continuously and presented in Fig. 6.

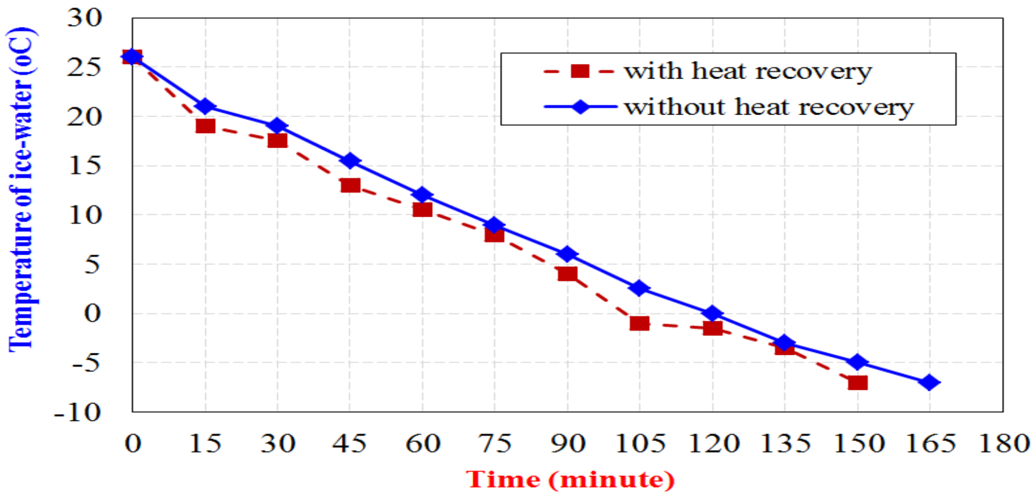

Figure 6. Variations in the temperature of ice water with time for two modes

As can be seen, the water temperature sharply decreases. This can be attributed to the high temperature of the water in rays (26^0C), resulting in a major part of released heat from the water being absorbed by the refrigerant. Moreover, the heat amount of the refrigerant is observed to be decreased when the water temperature is decreased to be nearly equal to the evaporating temperature of the refrigerant. Additionally, due to the small temperature difference between water and refrigerant, the lower temperature of water in the rays, the slower the heat exchange process is observed. However, when the temperature of ice water reaches -6^0C, the temperature relay automatically stops to provide electricity, leading to the operational interruption of the compressor.

3.2. Current intensity variation

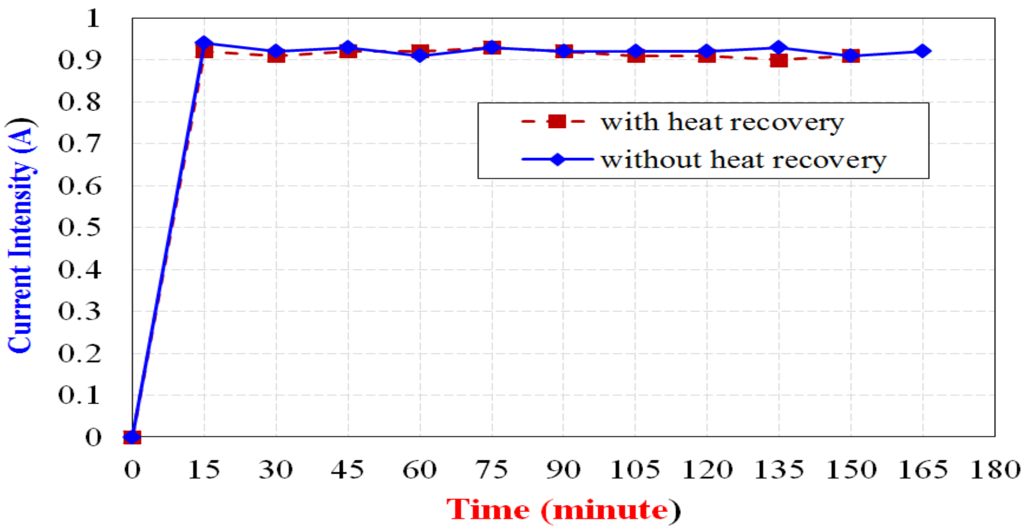

Figure 7. Variation in current intensity with time for two modes

It is noted that in the case without heat recovery, the current intensity is quite stable during operation time. It is found that in the case with the heat recovery, when the system starts, the current intensity is slightly greater; this can be attributable to the increase in pipe resistance, leading to the larger compression work [4]. However, when the system gets stable, the pressure and current intensity produced by the two modes differ insignificantly.

Therefore, we can argue that the appearance of a heat-recovery device has an inconsiderable effect on the current intensity of the system. Furthermore, when the system operates, this value can fluctuate; this is due to the change in the environment temperature, resulting in variations in the condensing temperature and compression work. Measured results are reported in Fig. 7.

4. Conclusions

From both theoretical and experimental results, several concluding remarks can be made as follows:

- It is noted that the subcooling always increases the coefficient of performance or exergy [5]; meanwhile, though leading to a slight increase in compression work, the overheat ensures the safe and stable operation of the compressor since the air sucked to it does not contain any liquids that can cause damages in the cylinder. Therefore, the heat-recovery process has only utilized the system using refrigerant of Freon [1]; the efficiency was found to be increased.

- When a heat-recovery was installed, the temperature of ice-water dropped faster; the cooling coefficient increased from 2,1 to 3,45. In addition, the current intensity was almost unchanged. The operation time shortened by about 17 minutes; specifically, the time-lengths were, respectively, 165 and 148 minutes for the case without and with heat recovery. This showed that the power consumption for the operation of the refrigeration system is significantly shorter

- The installation of a heat-recovery device could increase the initial cost. Additionally, the design calculations for a heat-recovery device were considered to be relatively complicated. Therefore, we referred [2,9] to choose suitable devices.

- Furthermore, to reduce power cost, we included a solar panel generating electricity through the converter for the operation of two fans when the weather becomes hot

- This study was performed under the experimental model with small capacity; in the future, we will extend our investigation to different devices of industrial refrigeration and air-conditioning systems.

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