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Research Article

PERFORMANCE IMPROVEMENT OF 5-KW HOUSEHOLD AIR-TO-WATER HEAT PUMP

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ABSTRACT:
Heat pump can take the advantage in higher performance than other technologies such as electric heater for producing hot water in household. In theoretical, the heat pump can transfer heat 3-4 times larger than the electric energy consumed which can identify the performance of heat pump in term of Coefficient of Performance (COP). However, the development of 5-kW household air-to-water heat pump with simple system design from previous work can achieve the COP only 2.37 following the standard testing of EN255-3 when the test condition was adapted for Thailand's weather at the ambient temperature of 25 ℃ and 70% relative humidity. Nevertheless, it still has some margin for improving the COP. Therefore, the objective of this study was to improve the performance of household air-to-water heat pump to meet the target of 10% COP increase. In this study, two improvement methods were proposed based on vapor compression system analysis and the realistic with the low investment approaches. Two methods of performance improvement were the increase of degree of subcooling at the condenser outlet by increasing the heating surface area of condensing coil, and the increase of evaporator pressure accompanied with decreasing in the degree of superheat at the evaporator outlet by changing expansion device from capillary tube to thermostatic expansion valve. The step improvement of heat pump performance, i.e. existing system, increasing the heating surface of condensing coil, using thermostatic expansion valve, and the combination of increasing the heating surface of condensing coil and changing thermostatic expansion valve, were confirmed by testing in the control room under the standard with the same condition. The results shown that the combination of both modifications can efficiently improve the COP of heat pump up to 2.64 or 11.38% higher than the existing system. In the viewpoint of the investment cost, the improvement of the full modification invested about 368 THB per 1% COP increase which was more attractive than the other such as changing the type of compressor.

Keywords: *Coefficient of Performance (COP), Air-to-Water Heat pump, Vapor compression cycle, Thermostatic expansion valve, Condensing coil*

1. INTRODUCTION

Air-to-water heat pump is high energy efficient technology in hot water supplied. Heat pump can extract heat from the air to generate the hot water. The performance of energy conversion in the heat pump, called the coefficient of

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performance (COP), could be achieved around 3 - 4 times larger than the electric energy utilization, while the energy efficiency of conventional electric heater is quite low. As a result of high energy conversion device, the use of heat pump in Thailand trends to continuously increase. The data from the year 2008-2012 shown that 1,603 heat pumps have been installed in Thailand which their capacities were in between 0.5 – 100 kW_{th} [1]. The 13-heat pumps were selected and tested the COP to present the heat pump specifications sell in Thailand. The test results illustrated that the COP of those 13-heat pumps were in the range of 2.4 - 3.7 which were below than the theoretical value. Thus, the margin of performance improvement was still available for the heat pump. Changing the type of heat exchanger to get higher efficiency and higher heat coefficient was one of the efficient method [2-3]. However, the high cost of investment was required.

Therefore, the objective of this study was to propose appropriated approaches to improve the COP of heat pump under the concept of easily practical modification and low investment cost to achieve 10% COP increase. The improvement was focused on a 5-kW air-to-water heat pump which had been developed few years ago under a simple vapor compression cycle. To examine the performance improvement, the standard EN255-3 [4] was implemented to evaluate the COP of air-to-water heat pump operation. This standard uses a tapping cycle and dynamic effects are contained in the test data to evaluate the COP instead of the steady-state testing of EN255-2 or EN14511 [5]. However, the temperature and relative humidity of test condition following EN255-3 which were determined at 15°C and 71%RH, must be adopted for Thailand in order to reflect the actual COP of heat pump when it used in Thailand where the weather is hot and humid rather than the condition of EN255-3. The surrounding condition with high temperature and high relative humidity gave the effect on the increase of COP as compare to lower temperature with lower relative humidity [6].

2. Basic of Heat Pump Technology

The principle of heat pump technology is almost the same as air conditioner and refrigerator, only the use of benefit is taken from the hot side of the condenser. The system of heat pump consists of 4 main components: compressor, condenser, expansion device, and evaporator, as shown in Fig. 1 and used the refrigerant as a working fluid to carry heat from the environment and transfer to the water.

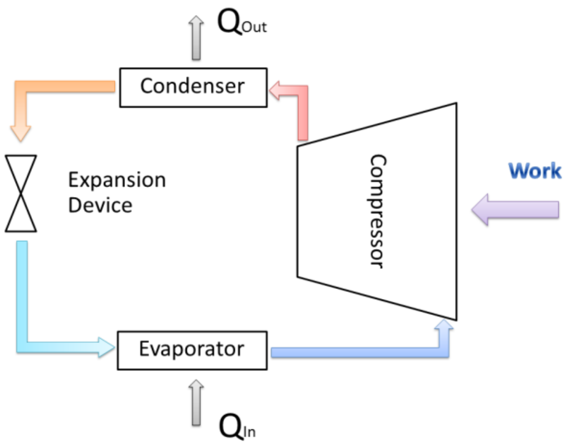

Fig. 1. System of heat pump.

3. Testing Procedure

Testing the performance of heat pump was conducted following the standard EN255-3 which has been prepared by Technical Committee RHE/17 [4]. In this study, the standard EN255-3 was thoughtfully selected as the most suitable standard for the heat pump used in Thailand. However, the test conditions must be adapted to be appropriate to Thailand's weather because the temperature and relative humidity of the surrounding play importance role on the COP of heat pump. Table 1 shows the comparison of test conditions between the original standard and the adaptation. All requirements were accompanied to construct a test room at the Department of Mechanical Engineering (ME), King Mongkut's University of Technology Thonburi (KMUTT), Thailand.

Table 1: The comparison of two test conditions [4]

Test Conditions	Standard EN 255-3	Thailand's weather
Dry bulb temperature in test room	15 °C	25 °C
Wet bulb temperature in test room	12 °C	21 °C
Feed water temperature	15°C	25 °C
Set point of water temperature at cut-off condition	Following spec.	55 °C
Set point of water temperature at working condition	Following spec.	50 °C
Flow rate of water use	15.6 l/min	15.0 l/min

The performance test includes with 5 principle stages [4] as explained below and illustrated in Fig. 2:

(1) Heating up period

Water at the temperature of 25°C is entirely fed into the storage tank. Before the heat pump is switched on, it shall be ensured that the system is in thermal equilibrium with the surrounding. The heating up time (t_h) and the electric energy input (W_{eh}) are measured during the heat pump is switched on until the compressor is switched off by the thermostat sensing the hot water temperature in the tank.

(2) Determination of the COP

After the compressor is first shut off, a single tapping is started. A half of hot water is draw from the storage tank while feed water is loaded into the tank. The heat pump is allowed to heat the water until the thermostat shuts it off again. After that another half of water volume is tapped at the same time and the heat pump is allowed to heat the water again until the thermostat shuts it off for third time. The energy content of two draw-offs, which can be evaluated from the measured flow rate and the temperature difference of water, shall not differ by more than 10%. The tapping and reheating time is measured between the two final shut offs the heat pump. The required reheating energy is determined over the second tapping period. Then the hot water tapping energy can be calculated from:

$$Q_t = \int_0^{t_t} \rho_{wh} \cdot c_{pw} \cdot q_{wh} \cdot (\theta_{wh} - \theta_{wc}) dt \tag{1}$$

where Q_t is hot water tapping energy. ρ_{wh} is density of hot water and c_{pw} is specific heat of hot water. q_{wh} is the flow rate of hot water tapping. θ_{wh} and θ_{wc} are hot and cold water temperature, respectively. t_t is tapping and reheating time.

(3) Determination of a reference hot water temperature

This period starts when the compressor is switched on for the first time after the reheating period. A large draw-off is initiated until the hot water temperature falls below 40°C as shown in Fig. 2 on the rapidly fall down of water temperature. To calculate the reference hot water, it can find from the average of the average temperature of this draw-off and that of the fifth stage.

$$\theta_{wr} = \frac{\theta'_{wh} + \theta''_{wh}}{2} \tag{2}$$

where θ_{wr}, θ'_{wh} and θ''_{wh} are reference hot water temperature, average hot water temperature during draw-off and hot water temperature at the 5th draw-off in unit of °C, respectively.

(4) Determination of standby power input

After the third stage is finished, the system is left to operate for a number of full cycles without tapping hot water. This is a stabilization period to determine the standby power input (P_{es}) of the system which should not be less than 24 hours and should compose at least one complete on-off cycle. The standby power input is measured when the first shut down after complete 24 hour until the next three complete on-off cycles are done. The duration of standby period is t_s and the required energy input is W_{es}. Then the standby power input can be evaluated by:

$$P_{es} = \frac{W_{es}}{t_s} \tag{3}$$

where P_{es} is standby power input. W_{es} is required energy input at standby and t_s is standby duration.

(5) Determination of the maximum quantity of usable hot water

After the standby period is finished, a hot water is started to draw off continuously until the hot water temperature fall below 40°C. The time from starting the draw-off until the water temperature less than 40°C is t_{max}. The maximum hot water energy is calculated by:

$$Q_{\max} = \int_0^{t_{\max}} \rho_{wh} \cdot c_{pw} \cdot q_{wh} \cdot (\theta_{wh} - \theta_{wc}) dt \tag{4}$$

where $Q_{\max}$ Maximum hot water energy (kWh)

$t_{\max}$ Time from starting draw-off until the water temperature less than 40°C (h)

After completing all test procedures, the COP of tapping period is commonly used to represent the performance of the heat pump rather than the COP from heating up period because it is more difficult to measure the energy effectively transferred to the water during the heating up period. Therefore, the COP_t of the second tapping was used to identify the performance of heat pump in this study. The coefficient of performance for tapping hot water (COP_t) can be calculated by following Eqs. (5) when Q_t is hot water tapping energy which was obtained from step (2).

$$COP_t = \frac{Q_t}{W_{es} - P_{es} \cdot t_t} \tag{5}$$

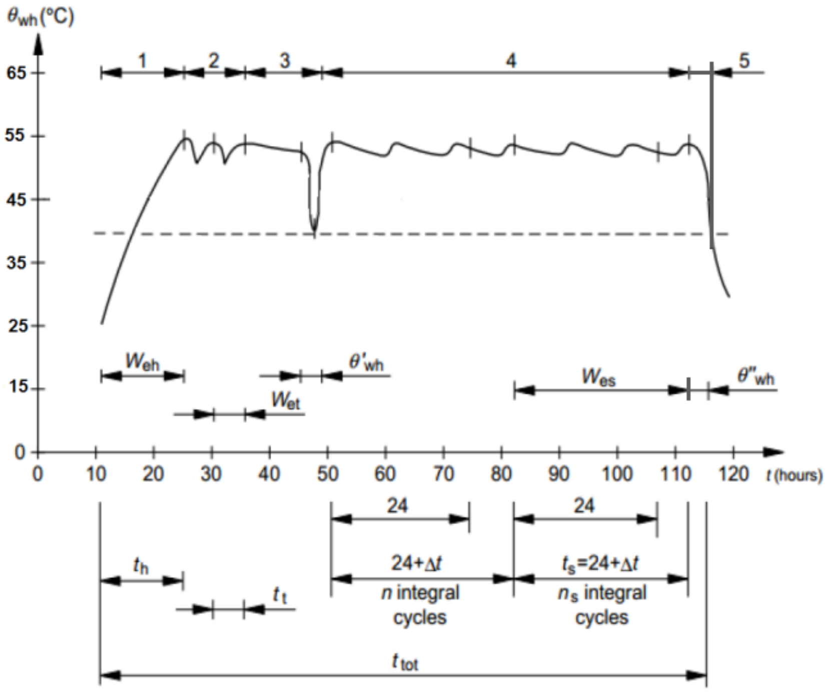

Fig. 2. The procedure of test performance EN255-3 [4].

Fig. 3 shows schematic diagram of testing room for the heat pump. Room temperature, the relative humidity, and the incoming water temperature and flow rate can be controlled following the design conditions. Data acquisition system is used to collect the experimental data, i.e. the water temperature, the energy input and etc., and recorded automatically for every 10 second.

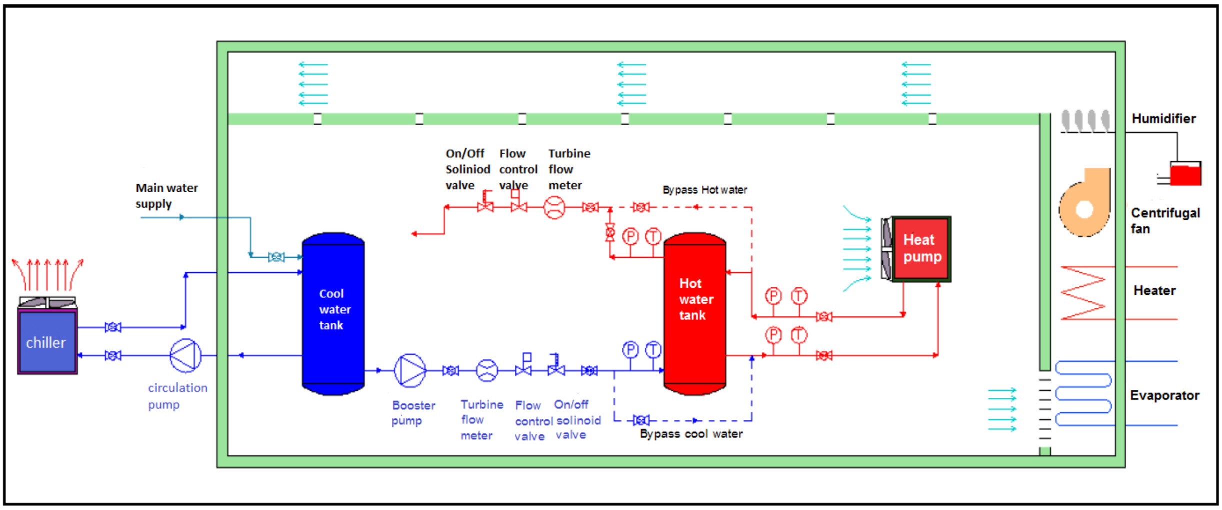

Fig. 3. Schematic diagram of testing room.

In this study, 5-kW air-to-water heat pump which was previously developed, as shown in Fig. 4, was selected for the performance improvement. The study was initially started with the standard test to confirm its performance, and then analyze the experimental data via Pressure - Enthalpy diagram to propose the method for the improvement.

A previous design of 5-kW air-to-water heat pump accomplished with 1.27-kW reciprocating compressor, condensing coil made from copper tube immersed in the storage tank, using fin tube heat exchanger as an evaporator, and using capillary tube as a flow-restricting device to reduce the pressure.

Fig. 4. A 5-kW air-to-water heat pump.

5-kW air-to-water heat pump was installed in the test room at ME, KMUTT. Then the test procedure was performed following the standard EN255-3. The room conditions were controlled at dry bulb temperature of 25^{o}C, 70 %RH. Feed water temperature was set at 25^{o}C with the flow rate of 15 liter/min. The test result showed that the COP_t of heat pump without any modification can reach at 2.37. Then using the data recorded of a specific time during the second tapping plotted on P-h diagram as illustrated in Fig. 5, it can be noticed that the degree of subcooling at the condenser outlet was very low while the degree of superheat at the compressor inlet was quite high about 12^{o}C. These two observations can give the effect on lower COP of the heat pump. In theoretical, the degree of subcooling at condenser can give the effect on the performance of vapor compression system. The COP can be increased as the increase of condenser subcooling due to the trade-off between increasing in refrigerating effect and specific compression work [7]. To obtain higher COP, therefore, the increase of condenser subcooling was significant. The degree of subcooling can be increased by increasing the area of heat transfer at the condensing coil. High degree of superheated was not appreciate in the system because it effected on the increase of compression work which resulted to the decrease of COP. Decrease of pressure evaporator was one of the method to reduce high degree of superheat as a result of increasing of refrigerating effect which can be achieved by using adjustable expansion device such as thermostatic expansion value.

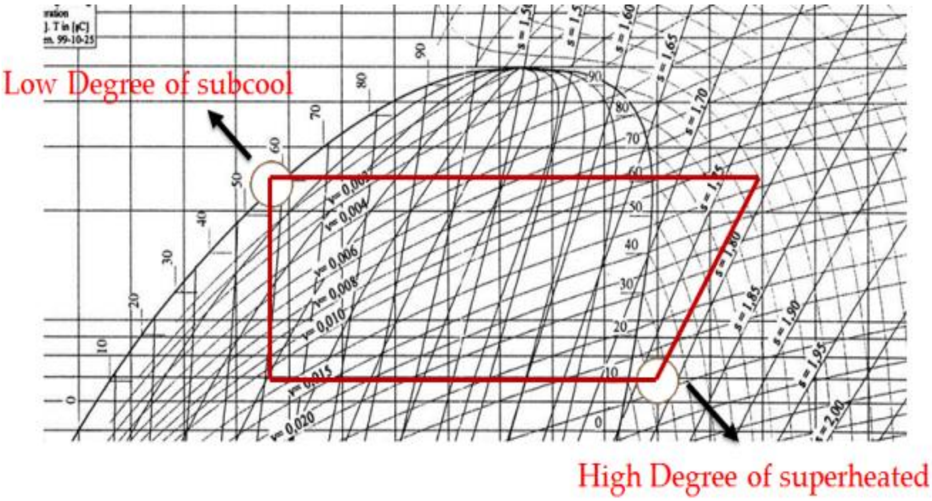

Fig. 5. Pressure – Enthalpy diagram of heat pump.

5. IMPROVEMENT

Even there are several approaches to improve the performance of heat pump such as using multistage vapor compression cycle, applying new refrigerant like R410A instead of R22 because it can absorb and release heat more efficient, design the system as hybrid system, and etc., but high investment cost are required for such approaches. Therefore, in this study, the improvements were taken into account from two observed points addressed at the initial testing.

5.1 Increasing degree of subcooling at condenser

Generally, a few subcooling is necessary to ensure that no vapor address in liquid. 3-5^{o}C degree of subcooling can be acceptable as a typical value designed for a heat pump (or refrigeration) system. According to the preliminary test, the result shows that the degree of subcooling was only 2^{o}C which means that some useful heat still remained and can be able to extract more. Original condensing coil had the diameter of 9.525 mm, the length of 15 m, and been using under the water temperature and the surface temperature of condensing coil conditions at 55^{o}C and 58^{o}C, respectively. The heat transfer rate at condensing coil during the 2nd water tapping was calculated before the improvement. Design lower of subcooling degree may cause by underestimating the heat transfer area of the condenser. To increase degree of subcooling from 2^{o}C to 4^{o}C, the surface temperature of condensing coil was determined to be 56^{o}C while the heat transfer rate and the water temperature was fixed as same as before. Dimensionless numbers, including Prandtl number (Pr), Rayleigh number (Ra), and Nusselt number (Nu), were used for the calculation. The result show that the length of new condensing coil was longer than the former one. Additional 7 m. of the condensing coil was required to obtain 4^{o}C subcooling degree which was expected to enlarge the output capacity at the heating side.

Therefore, a new design of condensing coil which increased the coil length and new pattern of fabrication was proposed to increase the heating surface area in order to achieve the degree of subcooling of 4^{o}C. This improvement provided an effect on lower discharge temperature at the exit of condensing coil. Fig. 6 shows a new design of condensing coil.

Fig. 6. New condensing coil design.

After the system modification by changing with a longer and new design condensing coil, the heat pump was tested in the test room under the same condition as a previous one. The result shows that a new system could increase the degree of subcooling from 2^{o}C to be 4-5^{o}C. It also provided the influence to the system operation by decreasing the condenser operated pressure by 3.2 bar. Consequently, the compressor consumed a lower electric energy and offered a higher COP. The COP tapping of the system improvement was 2.51. By this improvement, 2,200 THB was required as the improvement cost of the heat pump system.

5.2 Decreasing degree of superheat at evaporator

Superheat is important to assure that no liquid is entering the compressor for saving the life of the compressor because the compressor is designed to compress gas not liquid. However, too high superheat degree at the suction will impact on excessive discharge temperature, require more energy, and decrease the system capacity as the degree of superheat increases which results to the COP of the system decreases as the power input increases.

To decrease the degree of superheat, the throttle valve could be adjustable to moderate the evaporator pressure operated as full of saturated refrigerant with 8-10 degree of evaporator superheat. The originality design of this simple heat pump used a capillary tube as an expansion device which was widely used in a simple heat pump (or refrigeration) system because it was less cost and not moving part. But where a capillary tube was implemented, the cooling load cannot be more or less variation. Therefore, the use of thermostatic expansion valve was introduced to be adjustable which resulted to the decrease of superheat degree, as shown in Fig. 7.

Fig. 7. New expansion device.

The test result of this modified system shows that the degree of superheat decreased from 12^{o}C to 10^{o}C which effected to increase the evaporator pressure about 0.2 bar. A small increase of evaporator pressure affected to a lower energy of compressor and consequence to a higher COP. The testing data show that a new COP was improved to be 2.55 as compared with before the improvement. 2,000 THB was invested to improve the COP of the heat pump by changing a metering device.

5.3 Combination of new condensing coil and thermostatic expansion valve

From previous section, it shown that each improvement concept was able to succeed on the performance improvement of the heat pump system and also be practicable because of less investment cost. Therefore, the third modification was focused on combining the heat pump system with a new condensing coil design and using a thermostatic expansion valve instead of capillary tube.

The result shows that the COP of the heat pump was increased from 2.37 (before improvement) to be 2.64 (full options of the modification). In addition, it can be noticed that the degree of subcooling was increased from 2°C to 9-10°C resulted to decrease the condenser pressure by 1.7 bar, and the superheat degree was decreased from 12°C to 7-8°C which effected to a higher evaporator pressure about 0.4 bar. Fig. 8 shows P-h diagram for four systems of heat pump changing by the improvement. The whole cost for these improvements was 4,200 THB.

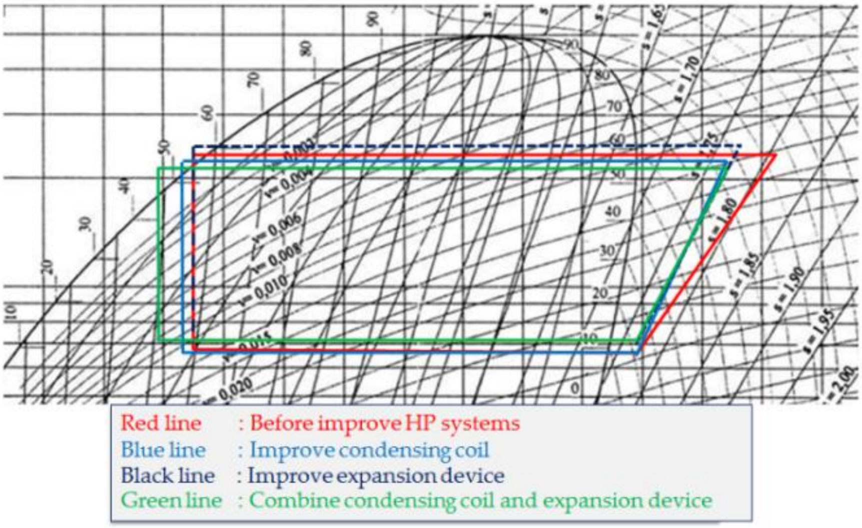

Fig. 8. Pressure - Enthalpy diagram of four heat pump systems.

5. Conclusion

By testing the heat pump system with the standard EN255-3 under Thai weather, the COP of a simple heat pump system design without any improvement was addressed only 2.37. The improvement with simple approaches were focused and found that the system can be improved by both increasing degree of subcooling and lowering superheat degree by P-h diagram analysis. To decrease the degree of subcooling, a new condensing coil was proposed and the COP can increase from 2.37 to 2.51, or 5.9% improvement with the investment cost of 2,200 THB, or 372 THB per 1% of COP improvement. While the use of thermostatic expansion device instead of capillary tube can enhance the COP up to 2.55 or 7.6% COP increased. In term of economic, 1% of COP improvement of this method used only 263 THB. If two approaches were combined together, the COP was increased to be 2.64 or 11.4% increased. The combination required more investment cost and resulted to higher unit cost of the improvement. The result shows that the system needed 368 THB for 1% of COP increment, as summarized in Table 2.

Table 2: The improvement of 5-kW heat pump

Item	Cost	COP	Increase of COP	Cost per 1% COP increased
Original	-	2.37	-	-
Condensing coil	2,200 THB	2.51	5.9 %	372 THB
Expansion Valve	2,000 THB	2.55	7.6 %	263 THB
Combined item	4,200 THB	2.64	11.4 %	368 THB

In the economic viewpoint, the result show that the change of capillary tube and using thermostatic expansion valve required less investment cost per 1% improvement of COP. To calculation the payback period of the investment, the investment cost of each case was divided by energy saving from the COP improvement. In Fig. 9, the result show that the payback period depended on the operating hour of the machine. Longer operation time, shorter payback period. However, it can obviously notice that the COP improvement by using thermostatic expansion valve was the most competitiveness. Finally, it can conclude that these improvement techniques could be more attractive than the other ways such the changing of the compressor type.

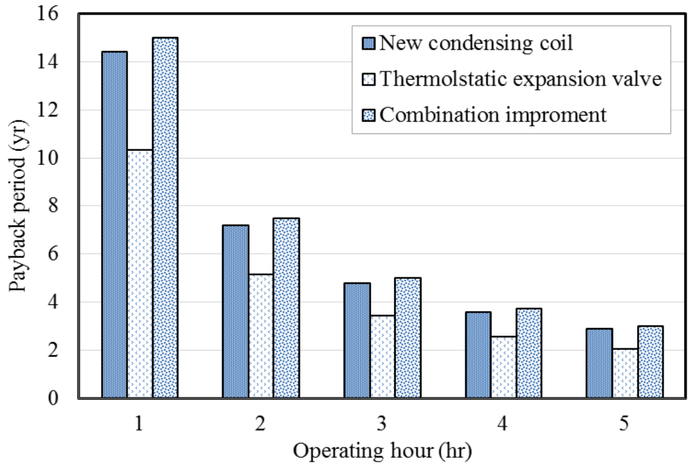

Fig. 9. Payback period of the COP improvements.

NOMENCLATURE

c_p	Specific heat, kJ/kg.°C
COP	Coefficient of performance, -
P_e	Power input, kW
q	Flow rate, m^3/s
Q	Energy, kW.h
t	Time, h
W_e	Energy input, kJ
ρ	Density, kg/m^3
θ	Hot water temperature, °C
θ'	Average temperature during draw-off, °C
Subscripts	
h	heating up
max	maximum
s	standby
t	hot water tapping
wc	cold water
wh	hot water
wr	reference hot water

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