Experimental Investigation of Closed Loop Oscillating Heat Pipe as the Condenser for Vapor Compression Refrigeration

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ABSTRACT

The aim of this article is to experimentally investigate the application of a closed loop oscillating heat pipe (CLOHP) as the condenser for a vapor compression refrigeration system. Split type air conditioner for residential use has two major disadvantages. First, it has a large pressure drop in the condenser caused by the flow of refrigerant inside a very small tube which affects compressor power and results in a decrease in the coefficient of performance (COP). Second, a large amount of heat is rejected to the surroundings since the refrigerant has to condense after passing through the condenser. To decrease pressure drop and recover heat rejection from the condensing process, this study considered using CLOHP instead of the conventional condenser in the split type air conditioner. The refrigeration capacity was set at 12,500 Btu/h (3.663 kW) with R22 as the refrigerant. The simulation of CLOHP condenser for the establishment of the optimum size of the vapor compression refrigeration system was performed using the thermo-economical method. For the optimum system size, it was found that water as the working fluid provided the highest net savings. The optimum size of the system with water as the working fluid consists of a 0.08 meter of evaporator section length, a 0.1 meter of condenser section length, pipe with an inner diameter of 2.03 millimeter, and 250 turns. Therefore, these sizes were selected to construct the CLOHP condenser. The experimental results were obtained and compared with the conventional condenser. It was found that COP of CLOHP condenser with a heat load of 800 W was decreased by about 32.4 % but the pressure drop of the CLOHP condenser was lower than that of the conventional condenser by about 91.2%. In addition, the energy efficiency rating of the CLOHP condenser was higher than the conventional condenser by about 13.4%. Finally, the outlet temperature of the cooling water which recovers heat from the condenser section of CLOHP, was increased by about 3 °C. The same trend was also observed for the heat loads of 900 W and 1,000 W.

1. INTRODUCTION

Refrigeration is the process of moving heat from one location to another by means of refrigerant in a closed refrigeration cycle. The refrigeration is developed and applied to use in various applications such as food industry, chemical industry and air conditioning for sustainable well-being. The air conditioning is commonly used in a wide range of residential and commercial buildings. Most of the air conditioner types used for this purpose are called "split type". This type of air conditioner is divided to two parts, a fan coil unit and a condensing unit which the fan coil unit is located inside the room and another one is located outside the room. The split type air conditioner based on the vapor compression refrigeration is shown in Figure 1(a). It has two disadvantages. First, it has a large pressure drop in the condenser caused by the flow of refrigerant inside a very small tube which affects compressor power and results in a decrease in the coefficient of performance (COP). Second, a large amount of heat is lost to the surroundings since the refrigerant has to condense after passing through the condenser. To reduce pressure drop and recover heat from the condensing process, in this investigation we used a Closed Loop Oscillating Heat Pipe (CLOHP) instead of the conventional condenser in split-type air conditioner as shown in Figure 1(b). The CLOHP is a heat

transfer device with very low thermal resistance, high thermal response, and can operate at low temperature difference. Many researchers studied the effects of different working fluids and fluid flow rate on the thermal effectiveness of CLOHP for air-conditioning. The studies showed that the thermal effectiveness decreases when the working fluid was changed from R134a to MP39 or the mass flow rate of cooling fluid was increased (Kammuang-lue *et al*., 2006).

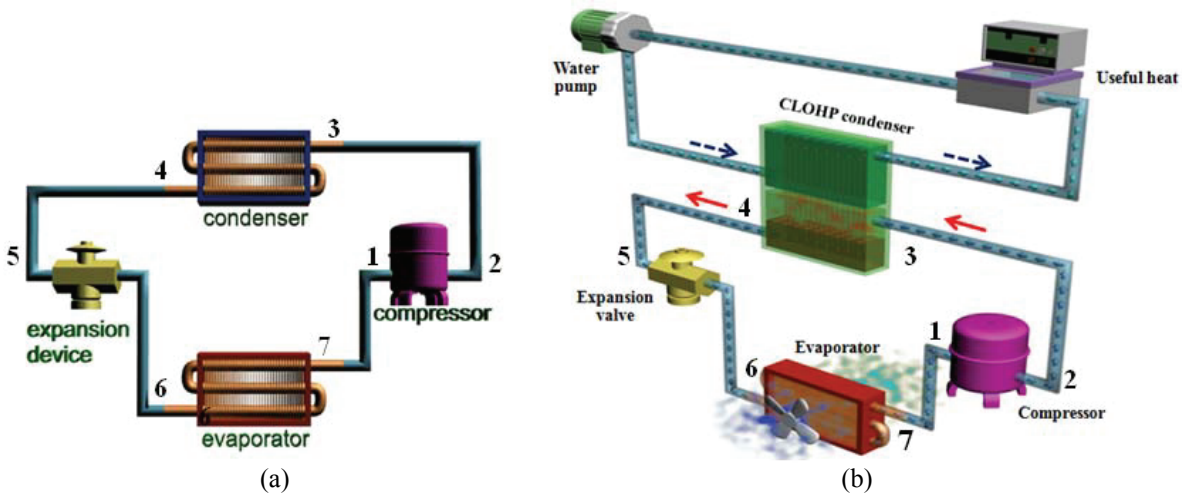


Figure 1: (a) The conventional vapor compression system
(b) The CLOHP condenser for vapor compression system

The CLOHP with check valves has been applied for reducing relative humidity in drying system and it can reduce relative humidity and achieve energy thrift (Meena *et al*., 2007). Heat rejected from a split-type residential air conditioner was recovered for clothes drying in residential buildings. The results indicated that the system was effective for its reasonably short drying duration and high energy use efficiency during air conditioning seasons (Shiming and Han, 2004). From the previous literature it can be seen that, there are no substantial studies on applying the CLOHP as a condenser in the refrigeration system to reduce pressure drop and recover heat from the condensing process. Therefore, the aim of this study is to experimentally investigate the use of a closed loop oscillating heat pipe as the condenser for vapor compression refrigeration system. Our optimization technique will be on the basis of a thermo-economical method (Soylemez, 2000; Soylemez, 2003).

2. DESIGN CONDITION AND PERFORMANCE CACULATION

2.1 Design conditions

- Dimensions of test room were 2.5×4×3 m (Width×Length×Height).
- Cooling capacity of the air conditioning unit was determined by the cooling load calculation.
- The entire test room was insulated by polystyrene to control heating load and the test room was closed while experiments were performed.
- The CLOHP condenser was designed on the basis of the optimization technique by using thermo-economical method. The optimum size of the system with water as the working fluid are 0.08 meter of evaporator section length(L_e), 0.1 meter of condenser section length(L_c), 2.03 millimeter of inner diameter(D_i) and 250 number of turns(N).

2.2 Performance calculations

The locations of the components shown in Figure 2 correspond to those shown in Figure 1 and each process was calculated as follows:

- Pressure drop in the condenser, $\Delta P_c = P_2 - P_5$ (1)
- Cooling capacity, $\dot{Q}_e = \dot{m}_r (h_1 - h_6)$ (2)
- Heat rejection rate, $\dot{Q}_c = \dot{m}_r (h_2 - h_5)$ (3)

- Isentropic compression power, $\dot{W}_{comp,isen} = \dot{m}_r (h_2 - h_1)$ (4)
- Coefficient of performance, $COP = \frac{\dot{Q}_e}{P_{comp}}$ (5)
- Energy efficiency rating, $EER = \frac{\dot{Q}_e \, (Btu/h)}{\text{Power input the system (Watt)}}$ (6)
- Electrical power input to the compressor was directly measured and also computed as,

$$\dot{W}_{comp} = \frac{\dot{m}_r w_{comp}}{\eta_{isen}} \quad (7)$$

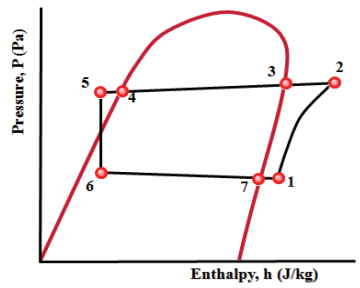


Figure 2: The P-h diagram of vapor compression system

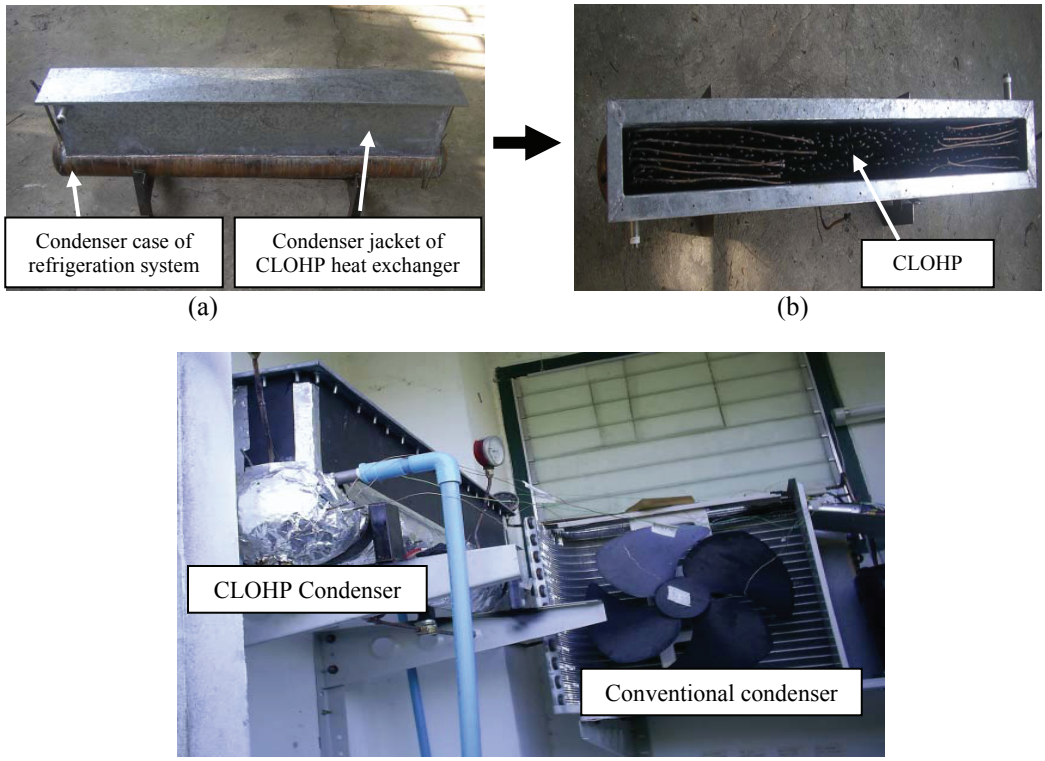
3. EXPERIMENTAL SETUP AND MEASURING EQUIPMENT

3.1 Experimental setup

The experimental setup was divided to two main parts: the conventional condenser system and the CLOHP condenser system, as shown in Figure 3(c). The conventional condenser system consists of four major components of the system, namely compressor, condenser, capillary tube and evaporator. The CLOHP condenser system uses CLOHP instead of the conventional condenser in the same refrigeration system. The CLOHP condenser is shown in Figure 3 (a) and (b), while specifications are given in Table 1.

Table 1: Specifications of the CLOHP condenser

Condenser case		**CLOHP**		**Condenser jacket**	
Material	copper	Material	copper	Material	zinc
Inner diameter	110 mm	Inner diameter (D_i)	2.03 mm	Width	90 mm
Thickness	1.2 mm	Number of turn (N)	250 turns	Length	1,020 mm
Length	1,120 mm	Evaporator section length (L_e)	0.08 m	Height	200 mm
		Adiabatic section length (L_a)	0.03 m		
		Condenser section length (L_c)	0.1 m		
		Working fluid	water		



(a) (b)

(c)

Figure 3: (a) The external CLOHP condenser
(b) The internal CLOHP condenser
(c) The experimental setup

3.2 Air conditioning unit

Cooling capacity of the air conditioning unit is 12,500 Btu/h (3.663 kW). The compressor is of reciprocating type and R22 is used as the refrigerant.

3.3 Heating loads

The test room was subjected to heating loads of 800, 900 and 1,000 W by means of an I-type electric heater. All of the heating loads were controlled by a slide regulator and were calibrated by a power clamp meter.

3.4 Cooling unit

The conventional condenser was cooled by air, while the CLOHP condenser was cooled by water. Cooling water was circulated by a water pump and the temperature of water was controlled by a cold bath. A factory calibrated rotameter was used to measure the volume flow rate of water.

3.5 Refrigerant temperature measurements

Refrigerant temperatures were measured by K-type thermocouples at four locations at the inlet of compressor, condenser, capillary tube and evaporator, respectively. The thermocouples were installed on the outside of the refrigerant copper tube surface using thermal paste to ensure good contact. Thermocouples were calibrated in a water bath with an accuracy of ±0.5°C (5-90 °C) and connected to data logger interface with a desktop computer.

3.6 Electrical power input measurement

Power input of the entire system was measured by a digital power clamp meter.

3.7 Refrigerant pressure measurements

Pressure of the refrigerant was measured by Bourdon pressure gauges at the same four locations that the refrigerant temperature was measured by the thermocouples. The pressure gauges were factory calibrated with an accuracy of ±1% (-30-120 psi for low pressure and 0-500 psi for high pressure).

3.8 Refrigerant mass flow rate measurement

The refrigerant mass flow rate was measured by a factory calibrated orifice flow meter for R22 refrigerant with an accuracy of ±5% (5-35 g/s). The mass flow rate meter was installed in the liquid line for liquid phase measurements. To ensure that the refrigerant was in the liquid phase, a sight glass was installed before orifice flow meter.

4. EXPERIMENTAL PROCEDURE

The experiments were divided to two main parts: one with the conventional condenser and the other with the CLOHP condenser. Each main experiment was divided to three sub-experiments that were conducted at the heat loads of 800, 900 and 1,000 W, respectively. In each sub-experiment, all of the data were recorded at an interval of ten minutes and a period of three hours.

4.1 The conventional condenser experiment

Before each sub-experiment was conducted, the data logger and the desk top computer were turned on to make sure all the measuring equipments were ready. Initial operating condition was a heat load of 800 W. The experimental set-up was turned on for twenty minutes to ensure that the system has reached steady state, and then all data were recorded. The refrigerant pressure, the refrigerant mass flow rate and the power input were recorded for all the locations at the same interval of three hours. The same procedure was repeated for the other two heat loads of 900 and 1,000 W.

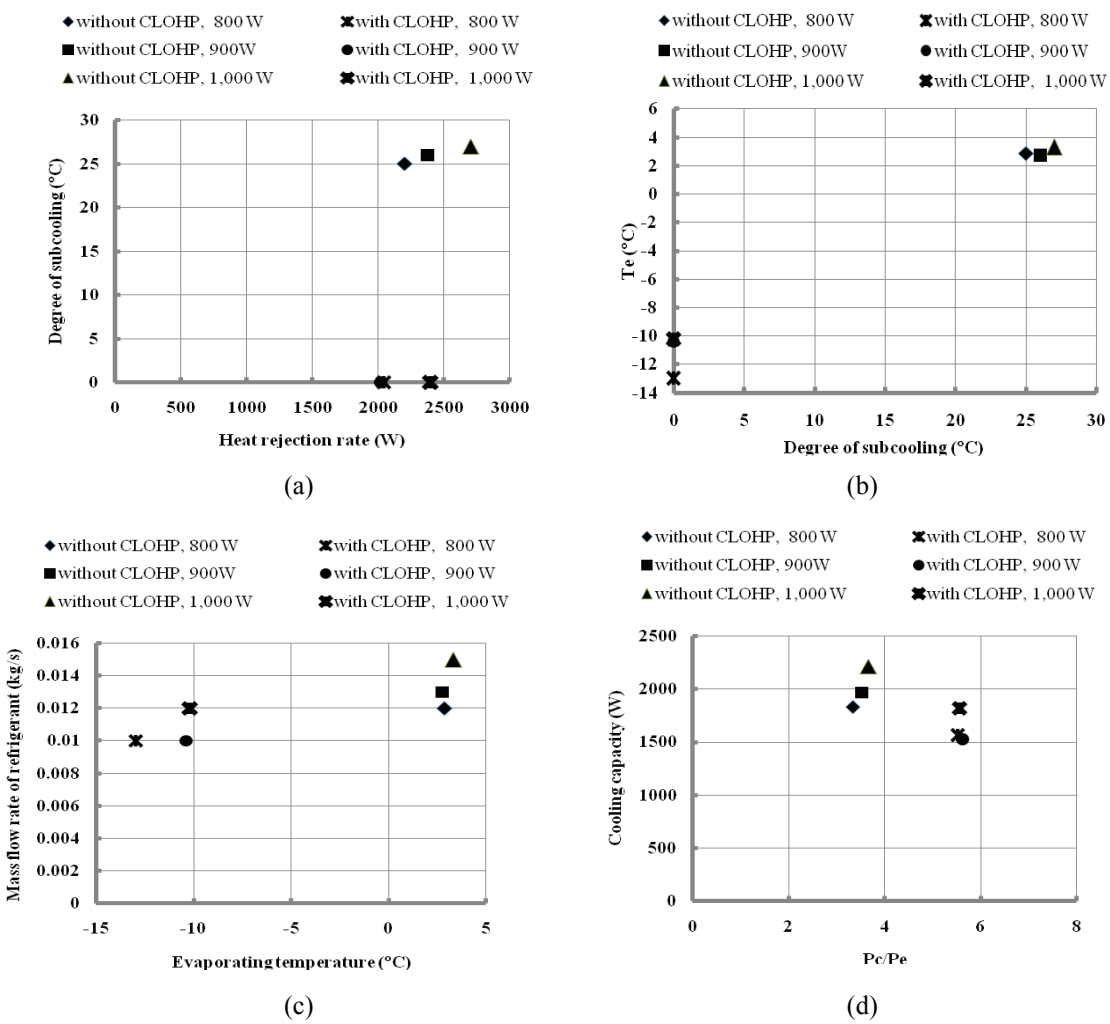
4.2 The CLOHP condenser experiment

In case of the CLOHP condenser experiment, only the conventional condenser was changed in the CLOHP condenser while other components were kept the same**.** The mass flow rate and inlet temperature of cooling water was fixed at 5 LPM and 25°C, respectively. Before all data were recorded, the sight glass was observed to ensure that the refrigerant was in the liquid phase. Then, the experimental procedure described for the conventional condenser experiments were followed.

5. RESULTS AND DISCUSSION

Figures 4 (a) to (h) show the results of experiments conducted on the conventional vapor compression system and the one with CLOHP condenser for different heat loads. Referring to Figure 4(a), for the conventional system (without CLOHP), the heat rejection rates with the heat loads of 800, 900 and 1,000 W were 2,202, 2,377 and 2,703 W, respectively. For the CLOHP condenser system with the same heat loads, the heat rejection rates were 2,042, 2,018 and 2,396 W, respectively. When these two systems were compered with the same heat loads, it was found that the heat rejection rates of the CLOHP condenser were decreased by about 7.3, 15.1 and 11.4%, respectively. In addition, the decrement of the heat rejection rate in the CLOHP condenser caused a decrease in the degree of subcooling for the refrigerant and the refrigerant was in the saturated liquid state for all of the heat loads tested, in this case the vapor fraction of the refrigerant that flow out of the capillary tube was higher. Therefore, the temperature of refrigerant at the capillary tube exit or the evaporating temperature of the CLOHP condenser system was sharply decreased as shown in Figure 4(b). From the same figure, it was found that the evaporating temperature of the conventional system with the same heat loads were 2.68, 2.75 and 3.33°C, respectively, while for the CLOHP condenser system it was -12.96, -10.38 and -10.21°C, respectively. The very low evaporating temperature of the CLOHP condenser system shown in Figure 4 (c) causes an increase in the specific volume of refrigerant for a constant compressor speed. This in turn for a constant volume flow rate of refrigerant would lead to a decrease in the refrigerant mass flow rate of the CLOHP condenser system. This observation agrees with the findings of Cabello *et al*. (2004). They experimented on a vapor compression system using R22 as the refrigerant and a condensing temperature of 45 °C. Cabello et al. (2004) in their experiments observed that the mass flow rate of refrigerant was slightly decreased when the evaporating temperature was significantly decreased. In our experiments as shown in Figure 4 (c) the mass flow rates of the conventional system with the same heat loads were 0.012, 0.013 and 0.015 kg/s, respectively while the refrigerant mass flow rates of the CLOHP condenser system were 0.01, 0.01 and 0.012 kg/s, respectively. Therefore, the refrigerant mass flow rates of the CLOHP condenser system were decreased by about 16.7, 23.1 and 20%, respectively. Figure 4 (d) shows that when the evaporating pressure of the CLOHP condenser was significantly decreased it caused the pressure ratio (Pc/Pe) of the CLOHP condenser to

increase. The pressure ratios of the conventional system with the same heat loads were 3.43, 3.52 and 3.66, respectively, while the pressure ratios of the CLOHP condenser system were 5.52, 5.62 and 5.56, respectively. Whenever the pressure ratio was increased, the mass flow rate was decreased (Cabello *et al*., 2004) because the volumetric efficiency of the compressor was decreased when the pressure ratio was increased (Gosney, 1982; Stoecker and Jones, 1982). Therefore, the cooling capacity of the CLOHP condenser system was decreased when the pressure ratio was increased (Cabello *et al*., 2004). Figure 4(d) also shows that the cooling capacity of the conventional system with the same heat loads were 1,830, 1,967 and 2,213 W, respectivly, while the cooling capacity of the CLOHP condenser system were 1,565, 1,529 and 1,818 W, respectivly. Therefore, the cooling capacity of the CLOHP condenser system was decreased by about 14.5, 22.2 and 17.9 %, respectivly. Referring to Figure 4(e), the higher compression ratio of the system caused the specific compression work in the compression process to increase (Cabello *et al*., 2004). Moreover, the higher specific compression work and lower cooling capacity of the CLOHP condenser system affected the COP and caused it to decrease, as shown in Figure 4(f). The same figure also shows that the COP of the conventional system with the same heat loads were 4.92, 4.79 and 4.51, respectively while the COP of the CLOHP condenser system were 3.28, 3.13 and 3.14, respectively. However, as shown in Figure 4(g), it can be seen that the CLOHP condenser system saved more electrical power for the system, the water pump power was much less than the condenser fan power, the EER of the CLOHP condenser system with the same heat loads was increased by about 18.9, 6.1 and 13.4%, respectively. In addition, Figure 4(h) shows that the pressure drop during the condensation process in the CLOHP condenser was sharply decreased because the refrigerant flows through a large tube during the condensing process. The pressure drop of the conventional condenser with the same heat loads were 83.5, 82.7 and 69.7 kPa, respectively, while the pressure drop of the CLOHP condenser were 6.89, 6.90 and 6.13 kPa, respectively. This decrement of pressure drop in the condenser caused a decrease in the compressor power consumption.



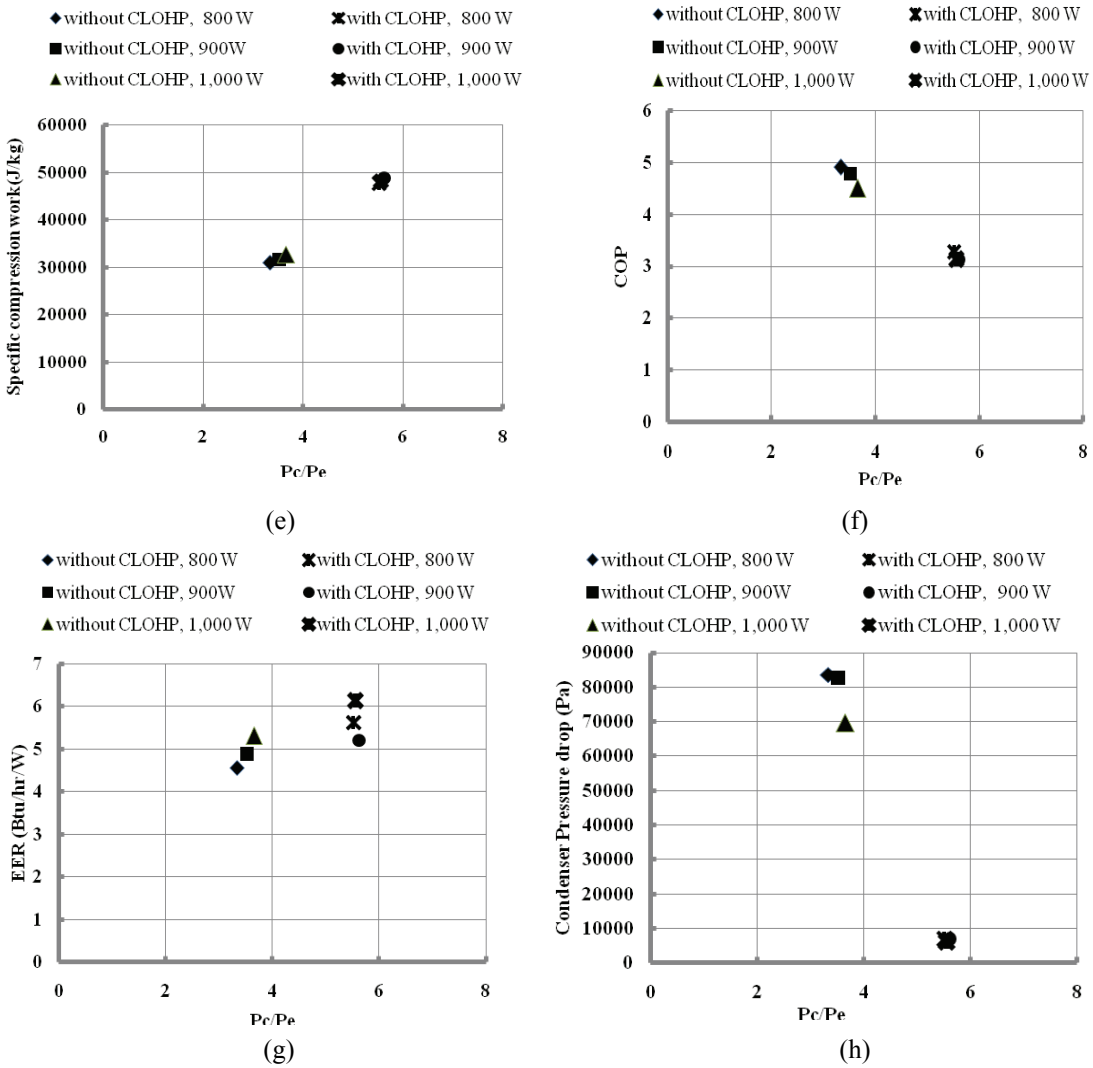


Figure 4: Comparison of experimental results between with and without CLOHP

6. CONCLUSIONS

From this study the following can be concluded:

- The COPs of the conventional system with the heat loads of 800, 900 and 1,000 W were 4.92, 4.79 and 4.51, respectively while the COP of the CLOHP condenser system with the same heat loads were 3.28, 3.13 and 3.14, respectively. The COP of the CLOHP condenser system was obviously lower than the conventional system.
- The pressure drops of the conventional condenser with the same heat loads were 83.5, 82.7 and 69.7 kPa, respectively while the pressure drops of the CLOHP condenser were 6.89, 6.90 and 6.13 kPa, respectively. Therefore, the CLOHP condenser can definitely reduce the pressure drop when compare with the conventional system.
- The outlet temperature of the cooling water which recovers heat from the condenser section of CLOHP, was increased by about 3 °C showing the possibility of recovering heat for future utilization.

To improve the COP, heat rejection rate and increase the outlet temperature of the cooling water which recover heat from the condenser section of CLOHP, in our future work will change the working fluid from water with other working fluids which have higher thermal efficiency and are environmentally safe, for example R134a.

NOMENCLATURE

COP	coefficient of performance	(-)		**Subscripts**
D	diameter	(mm)	a	adiabatic section
EER	energy efficiency rating	(Btu/h/Watt)	c	condenser section, condenser
L	length	(m)		
$\dot{m}$	mass flow rate	(kg/s)	comp	compressor
N	number of turn	(turn)	e	evaporator section
P	pressure	(Pa)	i	inner
$\dot{Q}$	heat rate	(Watt)	isen	isentropic
w	specific compression work	(J/kg)	r	refrigerant
$\dot{W}$	compressor power	(Watt)		
η	efficiency	(-)		

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