Modelling and optimization of the heat pump system for the usage of swimming pool

M J Lau1, M F Zanil1*, S Y Choong2 and J Tan3

1 Faculty of Engineering, Technology and Built Environment, UCSI University, 56000 Cheras, Kuala Lumpur, Malaysia.
2 Department of Chemical and Environment Engineering, University Putra Malaysia, 43400 UPM, Serdang, Selangor, Malaysia.
3 School of Engineering, Monash University Malaysia, Jalan Lagoon Selatan, 47500 Bandar Sunway, Selangor, Malaysia.

*Corresponding email: mohdfauzi@ucsiuniversity.edu.my

Abstract. Heat pump is a device which being used to transfer heat energy from one or more heat sources at low temperature to one or more heat sinks at high temperature simultaneously through pumping. The heat pump system used in swimming pool is an air–to-water heat pump that made up of four elements, which are evaporator, compressor, condenser and expansion valve. The overall efficiency of the heat pump in recovering the heat energy with the least of work done needed is defined as Coefficient of Performance (COP). A calculation software interface is designed using programming language of Visual Basic in Microsoft Visual Studio software to model, simulate the operation of heat pump product in actual industrial application and aid the optimization of heat pump system. The calculation software interface developed generates high accuracy COP value at 4.90, which has only 0.62% difference compared to actual COP of the heat pump product at 4.87. In order to optimize the performance of heat pump, the number of turns of outer and inner coil in condenser and the water flow rate pumped into the condenser are changed to maximize the COP within allowable cost increment. The COP is improved from 4.90 to 5.52, 5.13 and 6.89 based on each respective optimizations with cost increment controlled in between 0.5 to 2%. The final COP improved is obtained at 8.06 through combined optimization.

1. Introduction
Heat pump has gained a lot of attention from the industrial, commercial and residential areas as a promising way to provide heat energy in an environmental-friendly approach. The heat pump is recognized as an environmental-friendly technology by the European Directives which utilizes the renewable energy sources from water, ground or air for recovery of heat [1]. In recent year, attentions has been focused on the global issues in which the oil price has been shockingly increased, due to the depletion of fossil fuel energy that has been caused by the rapid development in industrialization. In 2008, the British Petroleum reported that the fossil fuels reserves may run out within some of the human lifetime if the world’s demand for energy from fossil fuels continues at the present rate in the BP Statistical Review of World Energy 2008, in which the estimated time length left for oil, natural gas and coal are 50, 70 and 250 respectively calculated in years [2]. Hence, heat pump application gains interest in becoming an alternative in providing energy source while conserving the non-renewable energy sources. Consumption of energy produced from fossil fuels contributes to environmental issues included the air pollution, global warming due to greenhouse gases, acid rain and depletion in ozone layer [3]. Therefore, the European Commission adapted a future-oriented proposal in early 2008, indents to reduce the greenhouse gases by promoting the usage of heat pump.

A thorough literature review had been studied in order to understand the heat pump operation and the processes therein in order to model and simulate the actual operation of heat pump system through
computer-aided software as well as improve the energy efficiency of heat pump to further enhance the quality of operation. Yongoua et al. [4] described the operation structure of an air-source heat pump water heater (ASHPWH) in which the heat pump system extracted aerothermal heat from outside air for sanitary hot water production. Liu et al. [5] explained the working cycle of the ASHPWH system based on the vapor compression refrigerant cycle which made up of four main processes which were evaporation, compression, condensation and throttling.

Wang et al. [6] proposed a systematic studies of continuous fin-and-tube tube heat exchangers to investigate the performance of plate-finned tube heat exchangers under dehumidifying conditions. An experimental research and operation optimization of an ASHPWH was carried out by Wu et al. [7] to optimize the design and operation strategy of ASHPWH through experimental set-up and simulation models. Robert et al. [8] produced a numerical model to describe the operation of a water-to-water heat pump system for steady-state condition in which the setup model is deterministic and consisted of distributed as well as lumped parameters. Patil et al. [9] described the design model of a helical-coil heat exchanger (HCHE) in continuous systems involved small to medium heat duties as well as the advantages possessed over double-pipe heat exchanger. In the thesis written by Kern [10], the fundamental instruction in the process heat transfer was employed in the development of heat pump model especially for evaporator and condenser parts.

The literature reviews show that numerous researchers had discussed the optimization and analysis of a heat pump system. Yongoua et al. [4] showed the combined effects of the ambient temperature and relative humidity, type and configuration of compressor, size and shape of evaporator as well as condenser and lastly the refrigerant filling quantity towards the COP of ASHPWH in South Africa. Wu et al. [7] studied the optimization analysis based on structural parameters through the increased in the outside area ratio of condenser coil to evaporator in improving the COP of ASHPWH. Zhang et al. [11] detected the important influence of the condenser coil tube length on the efficiency of an ASHPWH system.

In this paper, a heat pump product manufactured by an existing commercial heat pump supplier located in Selangor, Malaysia was modelled by designing an user-friendly and simple operation manner calculation software interface to simulate the operation of the heat pump system as precisely to the actual case. The heat pump product is an air-to-water heat pump which composed of evaporator, compressor, condenser and expansion valve that is used to heat the swimming pool water pumped into the condenser through heat recovery from ambient air at evaporator with presence of refrigerant as the heat-carrying medium. Based on the designed simulation model, the relationship between the number of turns of coil and water flow rate in condenser towards the COP of heat pump were discussed. Optimizations were carried out to maximize the COP of heat pump by involved the least additional cost due to the parameters or properties changes in heat pump. The paper is presented in such sequence as below.

2. Methodology
Programming language, Visual Basic. Net (VB.Net) built-in in Microsoft Visual Studio 2017 software was is as a modelling tool for the heat pump product. A calculation software interface using VB.Net is designed to simulate the operation of heat pump and calculate the final COP which directly indicate the efficiency and performance of the heat pump. The mathematical models of each individual components have been formulated to form the calculation structures in the software modelling to predict the operation performance of heat pump.

2.1 Evaporator Model
The evaporator used in the heat pump is a finned-tube typed heat exchanger. The material of the tube is copper whereas the material of fins is aluminium. The evaporator model of the heat pump system is presented as follow.

The energy equation of the evaporator [6] is expressed as in equation (2.1).

\[ Q_{\text{Evaporator}} = U_{\text{avg}} \times A \times F \times A_{\text{m}} \]  

(2.1)
Where $U_{o,w}$ denotes as the overall heat transfer coefficient which involves the total heat transfer coefficient at wet external fin-side and tube-side, meanwhile $\Delta i_m$ is known as the log mean enthalpy difference for the counter flow in evaporator [12], which are expressed as in equation (2.2) and (2.3).

$$\frac{1}{U_{o,w}} = \frac{1}{h_{o,w}x_{p,w}} + \frac{1}{h_{o,m}x_{p,m}}$$  \hspace{1cm} (2.2)

$$\Delta i_m = i_{a,in} + \left( \frac{(i_{a,in} - i_{a,out})}{i_{n,in} - i_{n,out}} \right) - \left( \frac{(i_{a,in} - i_{a,out})}{i_{n,in} - i_{n,out}} \right)$$  \hspace{1cm} (2.3)

Where $h_{o,w}$ is the total heat transfer coefficient at the wet external fin for the outside heat transfer with air, developed by Myers [13], whereas $h_i$ is the heat transfer coefficient at tube for the inside heat transfer with refrigerant, proposed by Gnielinski [14], which are expressed as in equation (2.4) and (2.5).

$$h_{o,w} = \frac{1}{\frac{1}{h_{o,m}} + \frac{1}{h_{o,in}}}$$  \hspace{1cm} (2.4)

$$h_i = \frac{(\frac{1}{2}) \times (R_{g} \times 1000 \times \rho_{R})}{1.07 + (12.7 \times \sqrt{\frac{T}{2} \left( \frac{1}{P} - 1 \right)^{1/2}}) \times \frac{\rho_{R}}{d_t}}$$  \hspace{1cm} (2.5)

Where $h_{o,a}$ represents the specific heat transfer coefficient of ambient air and $C_p,a$ is the specific heat of moist air at constant pressure.

### 2.2 Compressor Model.

The compressor used is a rotary-typed compressor. The compression of refrigerant gas in the compressor is assumed to be polytropic process, the energy balance of the compressor [8] is expressed as in equation (2.6).

$$W = \dot{m}_R \times \left( h_{con,in} - h_{eva,out} \right) \times \frac{n}{n-1}$$  \hspace{1cm} (2.6)

Where $n$ is the polytropic index or internal efficiency of the compressor, $\dot{m}_R$ is the mass flow rate of refrigerant pumped by compressor, whereas $h_{eva,o}$ and $h_{con,in}$ are denoted as the total enthalpy at the outlet of evaporator and inlet of condenser respectively [15], which are expressed as in equation (2.7) and (2.8).

$$h_{eva,out} = C \times T_{suction}$$  \hspace{1cm} (2.7)

$$h_{con,in} = C \times T_{delivery}$$  \hspace{1cm} (2.8)

Where $C$ represents the specific gas constant of refrigerant gas in compressor, stated by Gunther et al. [15], $T_{delivery}$ and $T_{suction}$ are the delivery and suction temperature in compressor. The total input work into the compressor of Waterco involves the addition of work required by fan motor, calculated as in equation (2.9):

$$W_{Compressor} = W \times W_{Fan Motor}$$  \hspace{1cm} (2.9)

### 2.3 Condenser Model.

The condenser used is a double-helical coil heat exchanger, in which the material of the coil is made up of titanium. The condenser model of the heat pump system is listed as follow.

The energy balance equation for condenser [9] is expressed as in equation (2.10).

$$Q_{Condenser} = U \times A \times \Delta T_c$$  \hspace{1cm} (2.10)
Where $U$ denotes the overall heat transfer coefficient in the condenser [9] whereas $\Delta t_c$ is the corrected log mean temperature difference [10], which are expressed as in equation (2.11) and (2.12).

$$\frac{1}{U} = \frac{A_0}{h_{\text{in}}} + \frac{A_0 \ln(D_u)}{2n_k A_c} + \frac{1}{h_o}$$  \hspace{1cm} (2.11)

$$\Delta t_c = \frac{(T_{ci}-T_{eo})-(T_{co}-T_{ra})}{\ln(T_{ci}-T_{eo})} \times F$$  \hspace{1cm} (2.12)

Where $h_o$ represents the heat transfer coefficient at the shell-side involved the heat transfer with swimming pool water. The equation is selected for the reynolds number higher than 10000 based on the equivalent diameter [10], which is expressed as in equation (2.13).

$$h_o = 0.36 \times Re^0.65 \times Pr_{wa}^0.5 \times k_{sw}$$  \hspace{1cm} (2.13)

$h_{io}$ denotes the heat transfer coefficient at the coil-side involved the heat transfer with refrigerant pumped from compressor [9,10], which is expressed as in equation (2.14).

$$h_{io} = [(J_H \times \frac{k}{d_{ci}}) \times (Pr)^{\frac{1}{2}} \times (1 + 3.5 \left( \frac{d_{ci}}{d_{co}} \right)^{\frac{1}{2}}) \times \left( \frac{d_{ci}}{d_{co}} \right)]$$  \hspace{1cm} (2.14)

Where $J_H$ is the Colburn Factor obtained from the graph provided by Patil et al [9].

### 2.4 Expansion Valve Model

The expansion valve used in Waterco is Thermal Expansion Valve (TXV). The expansion of liquid refrigerant in TXV is assumed to be isenthalpic process, in which the process is carried out adiabatically with no work done by the TXV [16]. Assumed isenthalpic process, there is no enthalpy change through the TXV [7]. The energy balance equations for TXV are expressed as in equation (2.15), (2.16) and (2.17).

$$Q_{TXV} = 0$$  \hspace{1cm} (2.15)

$$W_{TXV} = 0$$  \hspace{1cm} (2.16)

$$h_{TXV,in} = h_{TXV,out}$$  \hspace{1cm} (2.17)

Mass flow rate of liquid refrigerant allows to pass through the TXV to evaporator in the heat pump is expressed as in equation (2.18).

$$m_r = m_{TXV} = (k_1 + k_2 \times T_{eva}) \times \sqrt{\frac{Pr(P_{cond}-P_{eva})}{T_{eva}}}$$  \hspace{1cm} (2.18)

Where $k_1$ and $k_2$ denotes the constants which referred to Wu et al [7] for TXV.

### 2.5 Coefficient of performance

The Coefficient of Performance (COP) is used to express the overall performance of a heat pump system in terms of efficiency [147]. The expression of COP of the system performance is shown as in equation (2.19).

$$COP = \frac{Q_{\text{condenser}}}{W_{\text{Compressor}}}$$  \hspace{1cm} (2.19)

Where $Q_{\text{Condenser}}$ denotes the condensing heat transfer rate and $W_{\text{Compressor}}$ represents the electrical power consumption of compressor and fan in the heat pump system.

### 2.6 Modelling procedure

Based on the mathematical models proposed for each of the individual components in the heat pump system, a calculation software interface is created to estimate the performance of each individual
components and generate the COP for the overall system performance. Input data inserted into each individual components for the modelling and simulation of heat pump system are mostly referred to the actual industrial data, with few assumptions and references from research articles that are compatible to the heat pump system studied in this paper. The heating capacity at evaporator and condenser, the electrical consumption at compressor, refrigerant filling quantity through expansion valve as well as the COP of the system are calculated as output data.

3. Results and discussion

3.1. Verification of heat pump calculation software interface using VB.Net.

The output data for each of the individual components as well as the overall system COP calculated by the calculation software interface designed using mathematical models developed were tabulated in Table 1.

| Component       | Final Output Parameter                          | Result Generated |
|-----------------|------------------------------------------------|------------------|
| Evaporator      | Thermal energy transferred in Evaporator, \( Q_{\text{Evaporator}} \) | 7806.78 W        |
| Compressor      | Total input work in Compressor, \( W_{\text{Compressor}} \) | 2056.05 W        |
| Condenser       | Thermal energy transferred in Condenser, \( Q_{\text{Condenser}} \) | 10081.84 W       |
| Expansion Valve | Mass flow rate of refrigerant pass through TXV, \( m_{\text{TXV}} \) | 0.41 kg/s        |
| Overall Heat Pump System | Coefficient of Performance, COP | 4.90 |

The output results generated from Table 1 were compared with the actual results obtained from the heat pump product manufactured in real. The output parameters that involved in the calculation of the COP of the heat pump, which were the thermal energy transferred in condenser, \( Q_{\text{Condenser}} \) and the total input work to compressor, \( W_{\text{Compressor}} \), as well as the final COP of the heat pump system were included in the comparison as displayed in Table 2. Table 2 showed the comparison between the actual and simulated output results. The value of simulated \( W_{\text{Compressor}} \) was exactly the same as the actual \( W_{\text{Compressor}} \) with no error in difference. Simulated \( Q_{\text{Condenser}} \) was higher than the actual \( Q_{\text{Condenser}} \) by 0.69\% in error. This was due to the assumptions made on the input data which were not completely compatible with the real heat pump system studied as certain input data cannot be obtained from actual product or operation. For the overall COP of the system, simulated COP was higher than the actual COP due to the presence of error in the simulated \( Q_{\text{Condenser}} \). The percentage of error calculated for the simulated COP was 0.62\%. Since the maximum deviation between the simulated and actual results for \( Q_{\text{Condenser}} \) and COP were both not more than 1\%, the study showed good agreement between the simulation results and actual data. Therefore, the calculation software interface designed using VB.Net for the heat pump system was able to generate high accuracy results and reliable for further analysis.

| Component       | Final Result | Actual Waterco Heat Pump Model (Aquasun 50) | Visual Basic Calculation Software Interface | Percentage of Error (%) |
|-----------------|--------------|---------------------------------------------|--------------------------------------------|------------------------|
| Evaporator      |              |                                             |                                            |                        |
| Compressor      |              |                                             |                                            |                        |
| Condenser       |              |                                             |                                            |                        |
| Expansion Valve |              |                                             |                                            |                        |
| Overall Heat Pump System |              |                                             |                                            |                        |


Compressor

|                  | Total input work in Compressor, \( W_{\text{Compressor}} \) | \( W_{\text{Compressor}} \) | 2056 W | 2056.05 W | 0 |
|------------------|-------------------------------------------------------------|-----------------|--------|-----------|---|

Condenser

|                  | Thermal energy transferred in Condenser, \( Q_{\text{Condenser}} \) | \( Q_{\text{Condenser}} \) | 10012.72 W | 10081.81 W | 0.69 |

Overall Heat Pump System

|                  | Coefficient of Performance, COP | COP | 4.87 | 4.90 | 0.62 |

3.2. Optimization of heat pump based on structural parameters and thermodynamic properties

In order to maximize the COP of the heat pump system with the least additional costing involved, the factors affecting the performance of the heat pump system were studied and the heat pump was optimized based on the selected parameters or properties. The heat pump calculation software interface designed using VB.Net was used to observe the optimization results. The optimizations that were carried out based on the geometric parameters, which were the number of turns of outer and inner coil in condenser as well as the thermodynamic properties, which was the water flow rate pumped into condenser were further discussed in following section.

3.2.1. Optimization analysis based on number of turns of helical coil in condenser.

As mentioned above, the condenser used to manufacture the heat pump product from Waterco was double-helical coil heat exchanger which consisted of outer and inner loop of coil. The optimization frameworks constructed involved simulation on No and Ni respectively were outlined as follow.

Optimization framework involved \( N_o \) (\( N_i \) = Fixed Variable)

Objective Function: Maximize COP: \( f (N_o, N_i, C_i) \)

Subjected to: Number of Turns of Outer Coil, \( N_o: 12 \leq N_o \leq 18 \)

Number of Turns of Inner Coil, \( N_i = 11 \)

Cost Increment: \( C_i \leq 1\% \)

As shown in Figure 1, the initial \( N_o \) was 12 turns in the condenser of the heat pump. When the value of \( N_o \) increased, the total length of helical coil needed to form \( N \) turns, \( L_c \) increased obviously provided that the \( N_i \) remained the same at 11 turns. In Figure 2, when the \( L_c \) increased due to increasing in \( N_o \), the condenser heating capacity, \( Q_{\text{Condenser}} \) increased significantly due to the increased in the total heat transfer surface area available at coil-side, \( A \) when longer \( L_c \) was used, which can further improved the COP of heat pump system effectively. However, the enhancement of the COP can only be done within the cost increment, \( C_i \) required due to the changes in \( L_c \) which was allowed by Waterco. The optimizations performed involved the \( N_o \) in condenser were accepted for \( C_i \) estimated at or below 1\% so the optimization was economically feasible, which was displayed as Figure 3. The recommended \( N_o \) after optimized was 14 turns to maximize COP from initial 4.90 to 5.20. 6.12 \% increased of COP had achieved at 1\% cost increment.
Optimization framework involved $N_i$ ($N_o = \text{Fixed Variable}$)

Objective Function: Maximize COP: $f\left(N_o, N_i, C_f\right)$

Subjected to:

- Number of Turns of Inner Coil, $N_i$: $11 \leq N_i \leq 17$
- Number of Turns of Outer Coil, $N_o = 12$
- Cost Increment: $C_i \leq 1\%$

Referred to Figure 4, the initial $N_i$ was 11 turns in the condenser of the heat pump. When the value of $N_i$ increased, $L_c$ increased as well, same concept as the optimization involved $N_o$, provided that now the $N_o$ remained the same at 12 turns. In Figure 5, when the $L_c$ increased due to increasing in $N_i$, the condenser heating capacity, $Q_{\text{Condenser}}$ increased significantly due to the increased in $A$ when longer $L_c$ was used, which in turn improved the COP of heat pump system. However, the enhancement of the COP through optimization involved the $N_i$ in condenser can only be accepted for $C_i$ estimated at or below 1\% as recommended by Waterco so the optimization was economically feasible, which was displayed as Figure 6. The recommended $N_o$ after optimized was 13 turns to maximize COP from initial 4.90 to 5.13. 4.69 \% increased of COP had been achieved at 0.80\% cost increment.
3.2.2. Optimization analysis based on water flow rate pumped into condenser. In this section, swimming pool water flowed through the shell-side of condenser, $q_w$ which was pumped by separate water pump was optimized to maximize the COP of the heat pump system. In this case, geometric parameters such as the $N_O$ and $N_i$ in condenser were acted as the fixed variables. The optimization framework constructed involved simulation on $q_w$ was outlined as follow.

**Optimization framework involved $q_w$ ($N_O$ & $N_i$ Fixed Variables)**  
Objective Function: Maximize COP: $f (q_w, N_O, N_i, C_I)$  
Subjected to: Water Flow Rate, $q_w$ (m$^3$/s): 0.0019 $\leq q_w \leq$ 0.0046  
Number of Turns of Outer Coil, $N_O = 12$  
Number of Turns of Inner Coil, $N_i = 11$  
Cost Increment: $C_I \leq 1.5\%$
In Figure 7, the initial $q_w$ pumped into the heat pump was 0.0019 m$^3$/s. When the value of $q_w$ increased, the condenser heating capacity, $Q_{Condenser}$ increased significantly, which later can increased the COP of heat pump system. This was because the increased in $q_w$ could improve the heat transfer coefficient at shell-side of condenser, $h_o$, which indicated that the outside heat transfer with swimming pool water in condenser had been improved. However, the enhancement of the COP through optimization involved $q_w$ in condenser can only be accepted for $C_I$ estimated at or below 2% as recommended by Waterco based on the change of the capacity of water pump required to support the increased $q_w$ so the optimization was economically feasible, which was displayed as Figure 8. The recommended $q_w$ after optimized was 0.0037 m$^3$/s to maximize COP from initial 4.90 to 6.89. 64.49% increased of COP had been achieved at 1.50% cost increment.
3.2.3. Final Optimization Results. The summary of results obtained from the optimization works completed as above were tabulated in Table 3.

Table 3. Summary of Optimization Results

| Parameter / Variable Optimized | Optimized Component: Condenser | Parameter / Variable Value Before Optimized | Parameter / Variable Value After Optimized | COP Value Before Optimized | COP Value After Optimized | Cost Increment Before | Cost Increment After |
|--------------------------------|--------------------------------|---------------------------------------------|---------------------------------------------|----------------------------|----------------------------|-----------------------|-----------------------|
| Number of Turns of Outer Helical Coil, NO | Condenser | 12 turns | 14 turns | 4.90 | 5.52 | 0.20 % | 1.00 % |
| Number of Turns of Inner Helical Coil, Ni | Condenser | 11 turns | 13 turns | 4.90 | 5.13 | 0.20 % | 0.80 % |
| Water Flow Rate into Condenser, qw | Condenser | 0.0019 m³/s | 0.0037 m³/s | 4.90 | 6.89 | 0.00 % | 1.50 % |

The three independent optimization works were combined to form and proposed as the final solution for the optimization of this heat pump product studied, which was shown in Table 4. The percentage of increased in COP obtained from the final combined optimization was 64.49% as the COP increased from 4.90 to 8.06.

Table 4. Final result of the combined optimization for heat pump

| Number of Turns of Outer Helical Coil, NO | Number of Turns of Inner Helical Coil, Ni | Water Flow Rate Pumped into Condenser, qw | COP Value from Combined Optimization |
|-----------------------------------------|-----------------------------------------|------------------------------------------|----------------------------------------|
| Before Optimized | After Optimized | Before Optimized | After Optimized | Before Optimized | After Optimized | Before Optimized | After Optimized |
| 12 turns | 14 turns | 11 turns | 13 turns | 0.0019 m³/s | 0.0037 m³/s | 4.90 | 8.06 |

4. Conclusion
In this paper, a calculation software interface was designed by using programming language, VB.Net in Microsoft Visual Studio 2017 to model and simulate the actual operation of the heat pump product manufactured by the existing commercial heat pump supplier located in Selangor, Malaysia. The mathematical models of each individual components which made up the main core of the heat pump system were developed and acted as the calculation structures of the software interface to calculate the output data for each individual components as well as the COP of heat pump system. This paper proposed the simulation and optimization of heat pump with the purpose to maximize the COP within acceptable costing involved. Based on the analysis and discussions of the results obtained, some conclusions are made as follow.

(a) The heat pump calculation interface designed using VB.Net was able to calculate the output results with high accuracy compared to the actual operation data in fast response, simple operation manner and required no cost. The difference between the simulated and actual values for \( W_{\text{Compressor}} \), \( Q_{\text{Condenser}} \) and COP were 0%, 0.69% and 0.62% respectively, which were observed to be significantly small. This proved that the mathematical models used behind the calculation of the simulated results were suitable and highly accurate to represent the real heat pump operation.
For the optimization of COP involved \( q_w \) with no changes in \( N_o \) and \( N_i \), the \( q_w \) was proposed to be optimized from 0.0019 m\(^3\)/s to 0.0037 m\(^3\)/s to increase the COP from 4.90 to 6.89 with 1.5 \% of cost increment. The percentage of COP increased was 40.61 \%.

(c) Final solution which involved the combination of all three optimizations done was proposed, in which the COP can be maximized from 4.90 to 8.06, with 64.49 \% increased by optimized the \( N_o \) to 14 turns, \( N_i \) to 13 turns and \( q_w \) to 0.0037 m\(^3\)/s at the same time.

References
[1] Borre A 2011 Definition of heat pumps and their use of renewable energy sources REHVA Journal 38-9
[2] Ruhl C 2008 BP statistical review of world energy London: BP
[3] Aksoy N, Gok O, Mutlu H, and Kilinc G 2015 CO2 Emission from Geothermal Power Plant in Turkey In Proc. World Geothermal Congress
[4] Yongoua J, Tangwe S and Simon M 1990 A review on the performance assessment and optimization techniques of air source heat pump water heaters used in South Africa Heat Recovery Systems & CHP 437-46
[5] Liu F, Ma Y, and Zhuang R 2008 An Experimental Study on the Heat Pump Water Heater System with Refrigerant Injection Int. Refrigeration and Air Conditioning Conf. p 5
[6] Wang C, Hsieh Y, and Lin Y 1997 Performance of plate finned tube heat exchangers under dehumidifying conditions Journal of Heat Transfer 109-17
[7] Wu I, Wang R, Li S and Guo J 2011 Experimental research and operational optimization of an air source heat pump water heater Applied Energy 4128-38
[8] Robert S, Laszlo G and Igor F 2015 Optimization of heat pump system Energy 45-54
[9] Patil R, Shende B and Ghosh P 1982 Designing a helical-coil heat exchanger Chemical Engineering 85-8
[10] Kern D 1950 Process Heat Transfer (New York: McGraw-Hill)
[11] Zhang J, Wang R and Wu J 2007 System optimization and experimental research on air source heat pump water heater Applied Thermal Engineering 1029-35
[12] Yoo S, and Lee D 2004 An Experimental Study on Performance of Automotive Condenser and Evaporator Int. Refrigeration and Air Conditioning Conf. (Purdue: Purdue University) pp 1-7
[13] Myers R 1967 The Effect of Dehumidification on the Air-Side Heat Transfer Coefficient for a Finned-Tube Coil M.S. Thesis (Minneapolis: University of Minnesota)
[14] Gnielinski V 1976 New equation for heat and mass transfer in turbulent pipe and channel flow International Chemical Engineering 359-68
[15] Gunther E, Hiebler S, Mehling H and Redlich R 2009 Enthalpy of phase change materials as a function of temperature: required accuracy and suitable measurement Methods International Journal of Thermophysics 1257-69
[16] Helsdon R 1965 Introduction to Applied Thermodynamics : Adiabatic Throttling Process 1st Edition
[17] Clara V, David D, Filip L, Jan V and Lieve H 2012 Multi-objective optimal control of an air-to-water heat pump for residential heating Building Simulation 281-91

Acknowledgements
This study is supported by Waterco (M) Sdn. Bhd. Authors are grateful to Dr. Chin Swee Boon, Engineering Consultant from Waterco as well as Mr. Saw Soon King, Senior Executive and Product Engineer from Waterco for the technical support and advices throughout the study.