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Refrigerant circuitry optimization of an outdoor unit heat exchanger based on partial load operating conditions

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Abstract

Finned tube heat exchangers have been widely used as a condenser and an evaporator in a residential air conditioning and heat pump system. These heat exchangers are mostly designed based on peak load operating points. Heat pump systems operate considerably for a longer period of time at partial load operating conditions rather than peak load design point. This study investigated refrigerant circuitry optimization based on rated and partial load operating conditions. The achieved optimal refrigerant circuitry for each operating condition is different from the other due to different outdoor conditions. The results revealed that significant performance improvement of 4.70% and 2.40% have been achieved in average heat exchanger capacity using optimized refrigerant circuitry based on the highest weighted partial load operating condition rather than peak load design point in cooling and heating modes respectively.

Keywords: Finned tube heat exchanger; refrigerant circuitry; optimization; partial load; operating condition

1. Introduction

The International Energy Agency (2013) has stated that air conditioning (AC) and heat pump systems account for almost 700 million metric tons of CO_2 equivalent direct (7%–19%) and indirect emissions (74%) per year, which is responsible for global warming and ozone depletion. This percentage is expected to increase drastically in the next half-century as the demand for AC and heat pump systems around the globe is increasing day by day due to the rapid rise in temperature and humidity [1]. Direct emissions can be reduced by introducing low GWP refrigerants, whereas, indirect emissions can be controlled using more efficient heat pump systems [2].

One of the main components of a residential air conditioning (AC) and heat pump system are the finned tube heat exchangers that are used as a condenser or evaporator. Thermo-hydraulic performance of these heat exchangers is influenced by several geometric parameters including tube length, tube diameter, number of tube rows, tube pitch, fin pitch, and fin configuration. One of the important design parameter is the sequence in which the tubes are connected in order to exchange heat efficiently with the air [3]. There are n! possible connections for n number of tubes and is, therefore, not possible to evaluate each design manually. Hence, more sophisticated optimization techniques are required to numerically investigate the optimized refrigerant circuitry. One such method is using a genetic algorithm (GA) that has been employed in Intelligent System for Heat Exchanger Design (ISHED) [4], which produces an optimal refrigerant circuitry for evaporators and condensers based on the applied operating conditions. This tool is commercially available in a simulation package EVAP-COND [5] developed by NIST 2003. It has the capability to cope with different parameters and the non-uniform airflow distribution which results in significant heat exchanger performance improvement [6]. Furthermore, optimal refrigerant circuitry also reduces the size, cost and amount of refrigerant [7-8]. Liu et al also reported an optimization technique to improve heat transfer capacity with minimized cost [9]. Peng et al applied particle swarm algorithm technique on multi-stream plate-fin heat exchangers to investigate the effect of non-uniform inlet flow distribution along with pass arrangement [10].

Most of the heat exchangers up to the date had been designed based on peak load operating conditions that occur for about 1%-2.5% of the year. Such heat exchangers performs well under rated load condition but are generally oversized and ineffective in conditions other than peak load [11]. Therefore, the focus has been shifted on maximizing part-load efficiencies of the heat exchangers which required peak load operating point

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alongside the load profile [12]. Outdoor unit heat exchangers are exposed to environmental conditions that keep changing throughout the year. A refrigerant circuitry is sensitive to non-uniform airflow distribution along with other geometric and operating conditions. In many cases, the heat exchanger optimization process focuses on identifying refrigerant circuitry that provides the maximum heat transfer rate for given environmental constraints [13,14]. Thus, optimized refrigerant circuitry based on one set of technical and operating conditions may not perform well in other conditions. Therefore, the current study emphasizes developing an optimized refrigerant circuitry based on partial load operating conditions with the highest weighting factor that leads to significant heat exchanger seasonal performance improvement.

2. Physical description

Finned tube heat exchangers have been used as an outdoor unit with refrigerant R410A as a working fluid, inside the tubes circulating throughout the closed cycle. It consists of three depth rows with 30 tubes in each row. Louver wide fins have been used in the current study. Figure 1 is a basic model showing refrigerant flow path, with red circles indicating the inlets and blue circles indicating the outlets. The lower graph of air flow velocity distribution serve as an input to the EVAP_COND [5] in order to solve heat exchanger along with other geometric details i.e. tube length, tube diameter, fin pitch, etc. as given in Table 1.

Table 1. Geometric details of outdoor unit heat exchanger

Items	Outdoor unit
No. of tube rows	3
No. of tubes per row	30
Tube length	1720 mm
Tube inner diameter	6.3 mm
Tube outer diameter	7 mm
Fin pitch	1.7 mm
Fin thickness	0.092 mm
Fin type	Φ7 Louver(Wide)
Airflow	Non-uniform

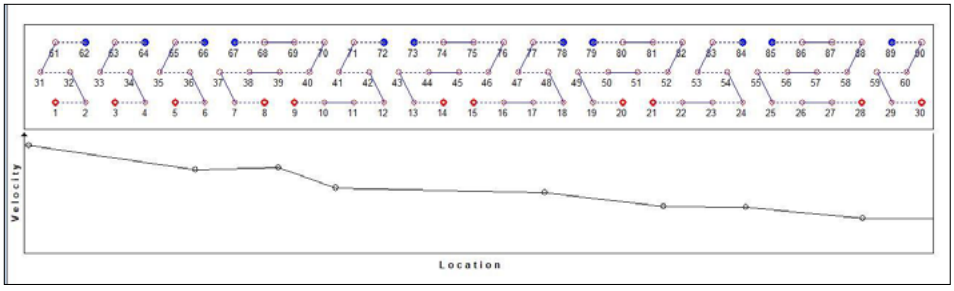


Fig. 1. Refrigerant path of outdoor unit heat exchanger used as a basic model

3. Operating conditions

Outdoor unit of a heat pump systems are exposed to environmental conditions and non-uniformly distributed air flows interact with the tubes. The magnitude and distribution of the flow velocity varies for each seasonal outdoor condition [15]. Heat pump and air conditioning systems operate in different outdoor seasonal conditions throughout the year. The actual weighting factor of the partial load operating condition is difficult to determine as it depends on the application and environment conditions of the AC and heat pump system. Alternatively, few standards are used nowadays, for better determination of the reflecting time of each load condition. One such method adopted in the current study is to utilize the weighting factors of the European standard EN14825 to calculate the average heat exchanger performance as given in Eqs. (1) and (2) for cooling and heating standards.

$$SEER = 0.03A + 0.33B + 0.41C + 0.23D \quad (1)$$

$$SCOP = 0.03A + 0.24B + 0.23C + 0.5D \quad (2)$$

where A, B, C, and D represent EER under 100%, 75%, 50%, and 25% load conditions for cooling mode and 100%, 84%, 51%, and 33% load conditions for heating mode respectively. Detailed partial load operating conditions in cooling and heating modes are summarized in Table 2 and Table 3.

Table 2. Partial load operating conditions in a cooling mode

S No	Part load ratio	Dry bulb Temperature (°C)	Wet bulb Temperature (°C)	Weighting coefficients	Airflow rate (m^3/min)	Average airflow velocity (m/s)	Refrigerant mass flow rate (kg/h)
Case 1	100	35	30	3%	182	3.18	430
Case 2	75	30	26	33%	135	2.36	323
Case 3	50	25	22	41%	112	1.96	209
Case 4	25	20	18	23%	65	1.14	112

Table 3. Partial load operating conditions in a heating mode

S No	Part load ratio	Dry bulb Temperature (°C)	Wet bulb Temperature (°C)	Weighting coefficients	Airflow rate (m^3/min)	Average airflow velocity (m/s)	Refrigerant mass flow rate (kg/h)
Case 1	100	-7	-8	3%	149	2.61	176
Case 2	84	2	1	24%	135	2.36	159
Case 3	51	7	6	23%	86	1.50	95
Case 4	33	12	11	50%	65	1.14	65

4. Numerical methodology

The current study has been carried out using commercially available software EVAP-COND version 4. This program uses the tube by tube method, where outlets from the first tube are considered as inputs to the second tube, and the outlets of the second tube are considered as an inlet to the third one and so on. Parametric tube details; inlet conditions, properties, and mass flow rates for air and refrigerant are known parameters and provide input for the first tube. Airside heat transfer rate can be expressed as Eq. (3)

$$Q_a = \dot{m}_a C_{p_a} \left(DBT - T_{ri}\right)\left(1 - \exp\left(-\frac{U_{an} A_i}{\dot{m}_a C_{p_a}}\right)\right) \quad (3)$$

where $\dot{m}_a$ is the air mass flow rate, C_{p_a} is the specific heat capacity of air-side, DBT is the air dry-bulb temperature, T_{ri} is the refrigerant inlet temperature, where U_{an} is the overall heat transfer coefficient, A_i is the surface area exposed to air. Based on heat transfer, outlet parameters are evaluated, with subsequent pressure drop calculations across each circuit.

Refrigerant side heat transfer and pressure drop correlations are based on flow quality i.e. annular flow ($0< x \leq 0.8$), dispersed flow ($0.8< x \leq 1$) or superheated flow ($x>1$). Overall heat transfer coefficient can be calculated using Eq. (4).

$$1/U_{an} = R_t = R_i + R_t + R_o + R_f = \sum R \quad (4)$$

where U_{an} is the overall heat transfer coefficient, R_t is the total thermal resistance, R_i is the inside tube resistance, R_t is the tube thermal contact resistance, R_o is the outside tube resistance and R_f is the fin to tube contact resistance. These variables are defined as follows,

$$R_i = \frac{A_i}{A_{i_ht} h_{an}}, R_t = \frac{A_i t}{A_{pm} k_t}, R_o = \frac{A_i}{A_{o_ht} h_{co}}, R_f = \frac{1}{h_t (1 - \frac{A_{fs}}{A_i})(1 - \phi)}$$

where A_{fs} is fin surface area, A_{i_ht} is tube inside surface area, A_{pm} is mean surface area, A_{o_ht} is tube outside surface area, t is tube thickness, k_t is tube thermal conductivity, h_{an} is inside tube heat transfer coefficient, h_{co} is air side heat transfer coefficient, ϕ is the fin efficiency. Two phase heat transfer coefficient can be found using Gungor Winterton [16] correlation as given by Eq. (5).

$$h_{an} = Eh_l + Sh_{pool}, \tag{5}$$

where E is the enhancement factor, h_l is the liquid heat transfer coefficient, S is the suppression factor, and h_{pool} is pool boiling heat transfer coefficient. Similarly air side heat transfer coefficient for louvered fins can be calculated using Eq. 6.

$$h_{co} = \frac{jGC_{pa}}{\Pr^{\frac{2}{3}}}, \tag{6}$$

Where j-factor for louvered fins can be found using J. Dong [17] air-side heat transfer coefficient correlation. G is refrigerant mass flux, Pr is the Prandtl number and C_{pa} is the specific heat capacity of air-side. Finned tube heat exchangers have several numbers of refrigerant circuits, with different lengths. Hence each circuit will have different refrigerant mass flowing through it. Fraction of refrigerant through each circuit can be expressed as Eq. (7)

$$F_i = \frac{m_i}{m_{tot}} = \frac{1}{\sum_{j=first}^{last} \left(R_i / R_j \right)^{0.571}}, \tag{7}$$

where $R_i = \Delta P_i / \Delta G_i^{1.75}$ is the total refrigerant flowing through the i-th circuit, $R_j = \Delta P_j / \Delta G_j^{1.75}$ is the total flow resistance for all branches splitting from that point. The first iteration loop estimates R_i. Resistance in each tube is assumed to be the same regardless of flow quality. Therefore, initial Ri depends on the number of refrigerant circuits and their paths. Subsequent iterations update R_i and F_i values. Iteration continues until the pressure drop across each circuit becomes uniform.

5. Results and discussion

ISHED is an optimization tool that provides optimized refrigerant path with maximum heat exchanger capacity as an objective function. Figure 2(a) shows optimized designs for different outdoor cooling conditions according to the European standard EN14825. Each optimized design differs from the other based on non-uniform airflow velocity distribution, outdoor air temperature, and refrigerant mass flow rate. In each case, different flow velocity profile interacts with the tubes which leads to different optimized design. Figure 2(b) shows a significant increase of 4.64%, 6.97%, 4.42%, and 10.36% in heat exchanger capacity for optimized designs as compared to the basic model under cooling conditions A, B, C, and D respectively.

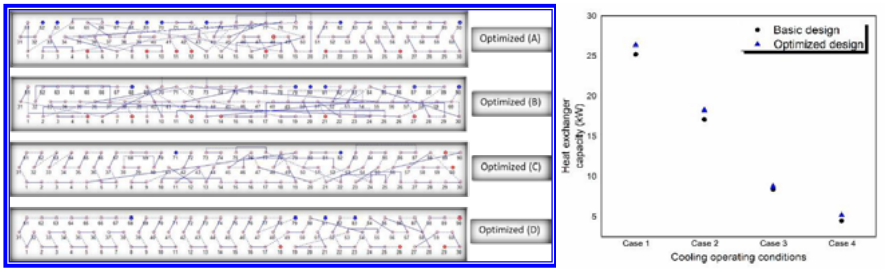


Fig. 2. Cooling mode (a) Optimized refrigerant paths, (b) Heat exchanger capacities for each optimized design

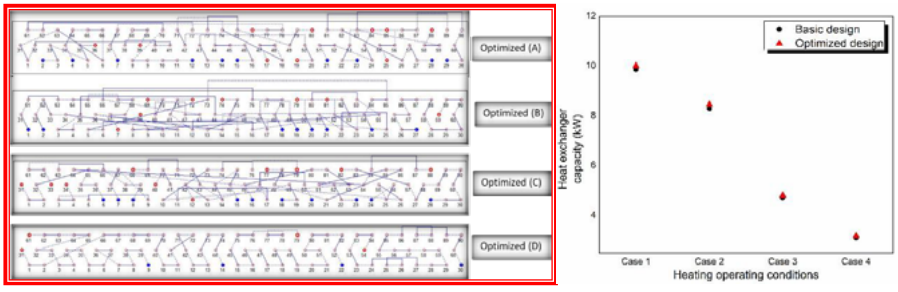


Fig. 3. Heating mode (a) Optimized refrigerant paths, (b) Heat exchanger capacities for each optimized design

Figure 3(a) shows optimized designs for different heating conditions according to the European standard EN14825. The optimized designs differ from the cooling mode as the outdoor operating conditions are changed. Figure 3(b) shows a significant increase of 1.52%, 2.05%, 2.96%, and 2.25% in heat exchanger capacity for optimized designs as compared to the basic model under heating conditions A, B, C, and D respectively.

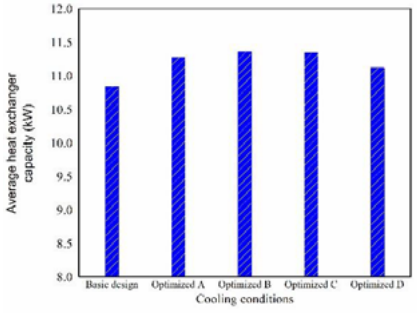


Fig. 4. Average heat exchanger capacities for each optimized design in cooling mode

Heat exchangers worked under different load conditions and are usually designed based on rated load conditions that worked for about 1%-2.5% of the time. In order to choose an optimal refrigerant path, the best approach is to determine an optimized refrigerant circuitry that worked well under all conditions. One similar approach is used here by finding the optimal refrigerant paths for all partial load operating conditions. Once all optimized paths have been achieved, the same paths are used for each condition and the average cooling capacities based on their weighting factors are calculated. Although it is difficult to accurately reflect the operating time under each load condition as it varies with respect to the application and environmental conditions. Alternatively, European standard EN14825 has been utilized to emphasize on the importance of refrigerant circuit optimization based on the most weighted load factor.

In the current study, the same outdoor unit has been optimized for all operating conditions and based on their weighting factor the average heat exchanger capacities have been calculated using Eq. (1). Figure 4 reveals that refrigerant circuitry optimized for the partial load operating condition with the highest weighting factor can significantly increase up to 4.70% in the average heat exchanger cooling capacity. Heat exchanger

capacities based on weighted load factor does not increase linearly with optimized designs. A maximum heat exchanger capacity has been found for using optimized design B in all conditions. Similarly, optimized design C is also better than optimized design A and D because the performance under part-load conditions of B and C were more highly weighted than the others. Therefore, it is recommended to use an optimized refrigerant circuitry based on the highest weighting factor throughout the year instead of the peak load.

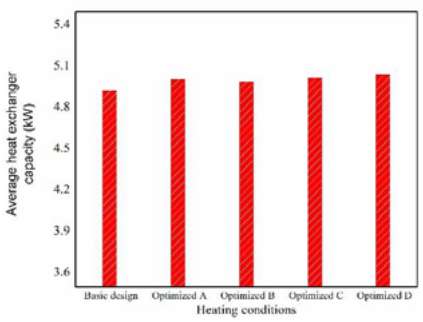


Fig. 5. Average heat exchanger capacities for each optimized design in heating mode

Similarly, Fig. 5 shows the average heat exchanger capacities using optimized designs A, B, C, and D. Each optimized design is tested for all four partial load operating heating conditions and the average heat exchanger capacity in each case has been calculated using the weighting factors of Eq. (2). The results reveal that the average heat exchanger capacity based on an optimized design for the highest weighted factor can improve heat transfer performance up to 2.4%. Sometimes the optimized design based on rated load condition shows more improvement but that is not realistic as the same design performs poorly in other more weighted partial load conditions.

6. Conclusion

The current study has been carried out to determine an optimized refrigerant path of an outdoor unit heat exchanger under rated as well as partial load operating conditions. One basic outdoor unit heat exchanger model has been selected with non-uniform airflow distribution. Heat exchanger capacity has been determined using a commercially available software EVAP-COND. The built-in refrigerant circuitry optimization tool ISHED was utilized to find the optimal refrigerant paths for all partial load operating conditions. Each partial load operating condition is applied to the obtained optimized designs. Average heat exchanger capacities have been calculated using weighting factors of the European standard EN14825. The findings of the present study have been listed below.

- A significant increase of 4.64%, 6.97%, 4.42%, and 10.36% in heat exchanger capacity have been achieved for optimized designs as compared to the basic model under cooling conditions A, B, C, and D respectively.
- A significant increase of 1.52%, 2.05%, 2.96%, and 2.25% in heat exchanger capacity have been achieved for optimized designs as compared to the basic model under heating conditions A, B, C, and D respectively.
- A maximum percentage increase of 4.70% has been achieved in average heat exchanger capacity using optimized refrigerant circuitry based on the highest weighted factor among the partial cooling loads.
- A maximum percentage increase of 2.40% has been achieved in average heat exchanger capacity using optimized refrigerant circuitry based on the highest weighted factor among the partial heating loads.

The current study is limited to the optimization of refrigerant circuitry using ISHED that optimized the refrigerant path for a single condition at a time. Further studies will extend the matrix by optimizing different geometric parameters based on the partial load operating conditions rather than a single peak load. Designing heat exchangers based on rated as well as partial load conditions can reduce huge energy consumption. A similar idea can also be extended to other types of heat exchangers used in industrial applications.

References

[1] G. Blanco, R. Gerlagh and S. Suh (eds). Drivers, Trends and Mitigation. In: Climate Change. 2014;351–412.
[2] B. Metz, O.R. Davidson, P.R. Bosch, R. Dave LAM (eds). Climate Change. Mitigation. 2007.
[3] P. A. Domanski, D. Yashar, 2007. Optimization of finned-tube condensers using an intelligent system. International Journal of Refrigeration 30, p.482-488.
[4] P. A. Domanski, D. Yashar, M. Kim, 2005. Performance of a finned-tube evaporator optimized for different refrigerants and its effect on system efficiency. International Journal of Refrigeration 28, p.820-827.
[5] P.A. Domanski, D.A. Yashar, EVAP-COND Version 3-Simulation Models for Finned Tube Heat Exchangers with Circuitry Optimization, National Institute of Standards and Technology, Gaithersburg, MD, 2010. http://www.nist.gov/el/building_environment/evapcond_software.cfm.
[6] D.A. Yashar, S. Lee, P.A. Domanski, 2015. Rooftop air-conditioning unit performance improvement using refrigerant circuitry optimization. Applied Thermal Engineering 83, p.81-87.
[7] Z. Wu, G. Ding, K. Wang, M. Fukaya, 2008. A general steady state mathematical model for fin-and-tube heat exchanger based on graph theory. International Journal of Thermal sceince 47, p.985-997.
[8] Z. Wu, G. Ding, K. Wang, M. Fukaya, 2008. Knowledge-based evolution method for optimizing refrigerant circuitry of fin-and-tube heat exchangers. HVAC&R Research. 14, p. 435-451.
[9] J. Liu, W.J. Wei, G.L. Ding, C.L. Zhang, M. Fukaya, K.J. Wang, T. Inagaki, 2004. A general steady state mathematical model for fin-and-tube heat exchanger based on graph theory. International Journal of Refrigeration 27, p. 965-973.
[10] X. Peng, Z. Liu, C. Qiu, J. Tan. 2016. Effect of inlet flow maldistribution on the passage arrangement design of multi-stream plate fin heat exchanger. Applied Thermal Engineering 103, p.67–76.
[11] Whole Building Design Guide. High-Performance HVAC. http://www.wbdg.org/resources/hvac.php (accessed June 9, 2010).
[12] United States Department of Energy Office of Energy Efficiency and Renewable Energy. How to Buy an Energy-Efficient Commercial Boiler.
http://www1.eere.energy.gov/femp/technologies/eep_boilers.html (accessed June 11, 2010).
[13] P. A. Domanski, D Yashar, K. A. Kaufman, R. S. Michalski, 2004. An optimized design of finned-tube evaporators using the learnable evolution model. Ventilating, Air-Conditioning and Refrigerating Research 10, p. 201–211.
[14] M.-H. Kim, S.Y. Lee, S.S. Mehendale, R.L. Webb, 2003. Microchannel Heat Exchanger Design for Evaporator and Condenser Applications. Advances in Heat Transfer 37, p. 297-429.
[15] F.A. Tereda , N. Srihari , B. Sunden, S. K. Das, 2007. Experimental Investigation on Port-to-Channel Flow Maldistribution in Plate Heat Exchangers. Heat Transfer Engineering 28, p.435-443.
[16] K.E. Gungor and R.H.S. Winterton, 1985. A general correlation for flow boiling in tubes and annuli. International Journal of Heat and Mass Transfer 29, p.351-358.
[17] J. Dong, J. Chen, Z. Chen, W. Zhang, Y. Zhou, 2007. Heat transfer and pressure drop correlations for the multi-louvered fin compact heat exchangers. Energy Conversion and Management 48, p.1506-1515.