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Operation parameters variation of air source heat pump according to the outdoor temperature

Hava kaynaklı ısı pompasının çalışma parametrelerinin dış ortam sıcaklığına göre değişimi

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Operation Parameters Variation of Air Source Heat Pump According to the Outdoor Temperature

Highlights

- *Outdoor air temperature highly impacts the ASHPs COP value.*
- *As the outdoor air temperature rises, the evaporator temperature and pressure increase.*
- *Determining the change in evaporator pressure allows charge control with manometer.*

Graphical Abstract

In this study, the changes in the COP, condenser heating load, refrigerant evaporation and condensation temperatures and pressures of an air source heat pump with a nominal heating capacity of 3.4 kW according to the outdoor temperature were determined.

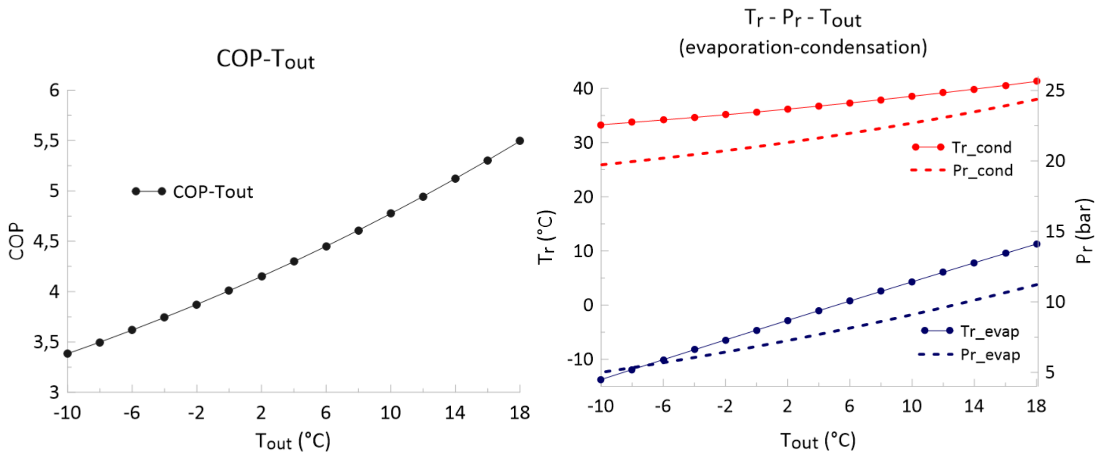

Figure. Change of COP, refrigerants evaporation and condensation temperature and pressure according to outdoor air temperature

Aim

It is aimed to determine to what extent the outdoor temperature changes the operating parameters of an air source heat pump.

Design & Methodology

A calculation method, based on the previously developed model related to logarithmic temperature difference changes in evaporator and condense is presented to determine the operation parameter changes.

Originality

To the best of our knowledge, there seems to be a lack of study that presents how the operating parameters of an ASHP change, with multiple data, depending on the outdoor temperature.

Findings

A raise from -10°C to 18°C in outdoor temperature, increases the COP value from 3.38 to 5.49, and evaporation pressure from 5 bar to 11.2 bar at selected ASHP.

Conclusion

The increase in outdoor temperature significantly improves ASHP performance. Determining the change in evaporation pressure can provide accurate charge control with the manometer.

Declaration of Ethical Standards

The author of this article declares that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Operation Parameters Variation of Air Source Heat Pump According to the Outdoor Temperature

Research Article

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ABSTRACT

Air source heat pumps (ASHP), known as energy efficient systems, emerge as environmentally friendly and economical solutions for building heating. Unfortunately, the operating parameters of ASHPs and accordingly their performance are directly affected by the daily and seasonal temperature changes of the outside air. In this study, change in operation parameters of a selected ASHP with 3.4kW nominal heating capacity is investigated with a calculation method based on the logarithmic mean temperature difference, between -10°C to 18°C outdoor temperature range with 2°C steps, at constant compressor power. The study is verified with published COP data of the manufacturer. Results are shared with graphs that give the variation of refrigerant evaporation and condensation temperature and pressure, COP, and condenser heating load according to outdoor air temperature. A raise from -10°C to 18°C in outdoor temperature, increases the COP value from 3.38 to 5.49. Detection of the increase in evaporation pressure in parallel with the outdoor temperature may allow easy control of the refrigerant charge level with a manometer. This study can be a useful guide for researchers who aim to determine the outdoor temperature dependent operating parameters of an ASHP and for technical personnel who need this information in fields.

Keywords: Air source heat pump, outdoor temperature, operation parameter, logarithmic mean temperature difference.

Hava Kaynaklı Isı Pompasının Çalışma Parametrelerinin Dış Ortam Sıcaklığına Göre Değişimi

ÖZ

Enerji etkin sistemler olarak bilinen hava kaynaklı ısı pompaları (HKIP) bina ısıtmalarında çevre dostu ve ekonomik çözümler olarak karşımıza çıkmaktadır. Ne yazık ki, HKIP'larının çalışma parametreleri ve buna bağlı olarak performansları, dış havanın günlük ve mevsimsel sıcaklık değişimlerinden doğrudan etkilenmektedirler. Bu çalışmada, -10°C ile 18°C dış sıcaklık aralığında, 2°C'lik aralıklarla, logaritmik ortalama sıcaklık farkına dayalı bir hesaplama yöntemi ile 3,4 kW nominal ısıtma gücüne sahip bir HKIP'nın, sabit kompresör gücünde, çalışma parametrelerindeki değişim incelenmiştir. Çalışma, üreticinin yayınlanmış COP verileriyle doğrulanmıştır. Sonuçlar, soğutucu akışkan buharlaşma ve yoğuşma sıcaklığının ve basıncının, COP'un ve yoğuşturucu ısıtma yükünün dış hava sıcaklığına göre değişimini veren grafiklerle paylaşılmıştır. Dış hava sıcaklığının -10°C'den 18°C'ye yükselmesi, COP değerini 3,38'ten 5,49'e yükseltmektedir. Ortam sıcaklığına paralel olarak gerçekleşen buharlaşma basıncındaki artışın tespiti, sistemin soğutucu akışkan dolum seviyesinin manometre ile kolayca kontrolüne imkan verebilir. Bu çalışma, bir HKIP'nın dış ortam sıcaklığına bağlı çalışma parametrelerini belirlemeyi amaçlayan araştırmacılar ve saha çalışmalarında bu bilgilere ihtiyaç duyan teknik personeller için faydalı bir rehber olabilir.

Anahtar Kelimeler: Hava kaynaklı ısı pompası, dış ortam sıcaklığı, çalışma parametreleri, logaritmik ortalama sıcaklık farkı.

1. INTRODUCTION

The increase in the heating and cooling demands of buildings, which comes with the increase in urbanization, also brings with it economic and environmental problems related to fossil based energy consumption [1]. According to the EIA's 2021 report [2], residential and commercial usages are liable for a substantial 24% of the world's energy consumption, and heating accounts for half of it [3]. Unfortunately, heating provided by burning fossil fuels with high carbon emissions is still the most widely used method in many countries and regions, nonetheless heat pumps consuming electricity, which are known as energy efficient heating systems, are replacing these traditional systems day by day [4]. Moreover, heat pumps, which can provide cooling as well as heating, stand out as a general air conditioning solution in most hot climate regions, as they offer a single system that can meet the summer and winter demands of a building [5, 6]. Heat pumps also provides commercial benefits by using it for drying agricultural products [7].

Heat pumps are systems that can meet the heating demand with low electricity consumption in the ratio of performance of coefficient (COP). Furthermore, electricity can be obtained from renewable sources instead of fossil fuels, and that can make these systems more environmentally friendly [8]. For this reason, it is necessary to make use of heat pumps as much as possible

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for more environmentally friendly, economical and sustainable heating in buildings [9].

Air is the most chosen heat source for heat pumps due to its ease of use and accessibility from anywhere. However, the fact that the temperature of the air is much more fluctuating during the season and day than the heat sources such as ground and water, causes the ASHP operating parameters to be variable [10]. This variability primarily affects the end user with the change in COP and heating capacity. In addition, knowing the temperature and pressure values in the evaporator and condenser can ensure that the controls such as the refrigerant charge level of the system can be made correctly.

In the literature, studies examining the variation of ASHP performance according to outdoor conditions were mostly examined together with an innovation for ASHP. Therefore, while focusing on the seasonal benefit of innovation, the effect of instantaneous temperature changes has not been adequately studied. Lee, S. H., et al. [11] has developed a model that simulates ASHP to determine the optimum operating parameters of ASHP such as frequency of compressor, evaporator and condenser air flow rates, maximum load, and compressor size for the highest seasonal performance. Kinab, E., et al. [12] has extensively modeled ASHP with detailed sub models of its components. In case of a change in equipment design, the seasonal performance variations were investigated by taking into account outdoor temperature and verified with experimental data. Underwood, C.P., et al. [13] developed an ASHP model, which includes a new efficiency-based compressor sub-model, parameterizing with the manufacturer's field or experimental data, and examined the performance variation at low ambient temperatures when the compressor is running at full load. Xu, Z., et al. [14] developed a partial theoretical model for evaluating the energy efficiency of ASHP systems and determined the stop-start and defrost losses for fixed speed ASHPs. Wang, W., et al. [15] defined a new performance indicator for non-inverter ASHPs in order to evaluate the deviation from the nominal operating condition due to the effects of ambient air temperature and defrosting.

The change in outdoor temperature will affect all the operating parameters of ASHP. In addition to the performance change, the determination of all operating parameters of the system depending on the outdoor temperature can clarify and facilitate the control of the system by the service personnel. For example, refrigerant charge control, which has an important place in service operations in ASHPs, can be achieved by controlling the evaporator pressure that changes with the outdoor temperature instead of the complicated subcooling method. The subcooling method provides an estimation of the refrigerant charge amount in modern heat pumps with thermal expansion valves by measuring the superheat level set for the evaporator outlet and the subcooling degree at the condenser outlet together. Therefore, it is necessary to measure the temperature from four points [16, 17].

The performance of ASHPs is largely related to the evaporation and condensation temperature of the refrigerant. These temperature values are directly related to the inlet temperatures of air flowing through the evaporator and condenser, which are fan added finned-tube heat exchangers. It can be assumed that the condenser air entrance temperature is constant after the indoor conditions reach equilibrium. On the other hand, outdoor air temperatures constantly change during the day and throughout the year. In consequence, the varying evaporator inlet air temperature effects the refrigerant evaporation temperature and hence the COP value, depending on the logarithmic temperature difference.

In this study, variation of operation parameters of a selected ASHP according to outdoor air temperature is investigated with the presented calculation method, based on the previously developed model related to logarithmic mean temperature difference (LMTD) changes in evaporator and condenser [18]. The examined ASHP is a conventional air conditioner with nominal heating capacity of 3.4 kW and COP of 4.53 under EN1451 test conditions, where the indoor temperature and the outdoor temperature are kept constant at 20°C and 7°C, respectively. The presented calculation method is applied to selected ASHP by utilizing the MS-EXCEL “COOLPROP” and “SOLVER” add-ons, in the outdoor temperature range of -10°C to 18°C with 2°C intervals, while the compressor power and fan speeds are kept constant at nominal value. The calculation method is validated by comparing calculated COP data with manufactures published experimental COP data. The variation of COP, condenser heating load and evaporator and condenser temperature and pressure values according to outdoor air temperature are discussed by sharing results graphically. This study will guide researchers who intends to identify the operation parameter changes of ASHP based on outdoor air temperature, and technical staff who need this information in field studies.

2. MATERIAL and METHOD

2.1. Theoretical Model of ASHP

The theoretical modeling of the system was developed based on the previous study. The performance of ASHP is mainly related to the evaporation and condensation temperature of the refrigerant. The refrigerant temperatures in the evaporator and condenser, which are counter-flow heat exchangers in fact, can be determined by finding the heat transfer rate $\dot{Q}_{he}$ and logarithmic mean temperature difference ΔT_m.

Heat exchangers heat transfer rate can be expressed as,

$$\dot{Q}_{he} = UA\Delta T_m \tag{1}$$

where UA is heat exchangers total thermal conductance, and can be assumed to remain constant as long as the fluid velocity doesn’t change.

NTU value at products datasheet can be used to identify the total thermal conductance as follow,

$$UA = C_{min}.NTU \tag{2}$$

The logarithmic mean temperature difference of the heat exchanger can be expressed as,

$$\Delta T_m = \frac{(T_{1i}-T_{2o})-(T_{1o}-T_{2i})}{\ln\frac{(T_{1i}-T_{2o})}{(T_{1o}-T_{2i})}} \tag{3}$$

where T_{1i} and T_{1o} are the inlet and outlet temperature of first fluid and T_{2i} and T_{2o} are the inlet and outlet temperature of second fluid.

For the fluid pair in the evaporator, this equation can be written as,

$$\Delta T_{me} = \frac{(T_{aei}-T_{re})-(T_{aeo}-T_{re})}{\ln\frac{(T_{aei}-T_{re})}{(T_{aeo}-T_{re})}} \tag{4}$$

where T_{aei} and T_{aeo} are the inlet and outlet temperature of air and T_{re} is the refrigerant evaporation temperature.

For the condenser, this equation can be expressed as,

$$\Delta T_{mc} = \frac{(T_{rc}-T_{aci})-\left(T_{rc}-T_{aci'}\right)}{\ln\frac{(T_{rc}-T_{aci})}{\left(T_{rc}-T_{aci'}\right)}} \tag{5}$$

where T_{aci} is the airs entrance temperature and T_{rc} is the refrigerants condensation temperature at condenser. $T_{aci'}$ is the modified entrance air temperature after heated by refrigerant until its condensation point, and can be determined by,

$$T_{aci'} = T_{aci} + \frac{\dot{m}_r(h_{rci}-h_{rc})}{\dot{m}_{ac}.c_{pa}} \tag{6}$$

where h_{rci} is condenser inlet refrigerant enthalpy, and h_{rc} is refrigerant condensation enthalpy at condenser.

ASHPs refrigeration cycle has been built on the basis of the ideal vapor compression cycle with the following assumptions.

- The refrigerant pressure losses along the pipe in the evaporator and condenser are neglected.
- Isentropic efficiency of compressor is accepted as fixed.
- The pressure drop in the expansion valve is isenthalpic.
- Superheat and subcooling degree is accepted as 5°C.
- Heat loss to ambient from components are neglected.

According to above mentioned assumptions, refrigerants temperature, pressure and enthalpy values can be determined with sufficient accuracy through the compressor, expansion valve, evaporator and condenser by using the COOLPROP add-on in MS-EXCEL. Heating loads of evaporator and condenser and compressor work output can be determined with following equations shown in Figure 1.

The COP value of ASHP, is defined by the equation below,

$$COP = \frac{Condenser\ heating\ capacity}{Sum\ of\ input\ powers\ of compressor\ and\ fans} = \frac{\dot{Q}_C}{\dot{W}_{Ci}+\dot{W}_F} \tag{7}$$

Considering the electrical efficiency, the power drawn by the compressor is defined as follows.

$$\dot{W}_{Ci} = \frac{\dot{W}_{Co}}{\eta_{ce}} \tag{8}$$

2.2. Solving Procedure

The logarithmic mean temperature differences (LMTD) in the evaporator and condenser of examined ASHP at given particular indoor and outdoor air temperature are determined with equations specified in Section 2.1 by using the datasheet values given in Table 1.

Table 1. The datasheet values of examined ASHP [19, 20]

ASHPs operating parameters	Reference ASHP FTXS35G2V1B RXS35G2V1B
Outdoor air temperature T_{aci}	7 °C
Indoor air temperature	20 °C
Refrigerant type	R410A
Condenser heating load	3400W
Air flow rate at evaporator	30.2 m^3/s
Air flow rate at condenser	12 m^3/s
Fans electricity consumptions	50 W + 23 W
COP in given conditions	4.53
NTU (Evaporator)	2
Compr. electrical efficiency η_{ce}	0.8
Compr. isentropic efficiency η_{cs}	0.85

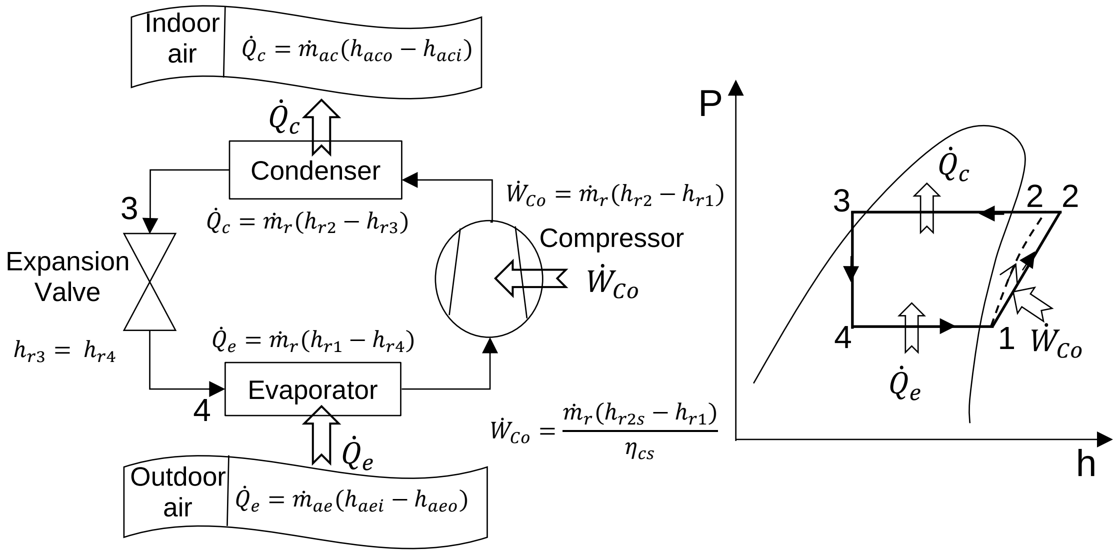

Figure 1. Energy balance equations of air and refrigerant throughout the cycle

The new LMTDs that will occur in the evaporator and condenser in case of a change in the outdoor temperature can be determined with the following equations, taking into account the change in heating capacities. Calculation of LMTD change in condenser is simplified compared to the previous work, by ignoring the change in the areas of the heat transfer zones inside the condenser.

It should be noted that these equations are valid if the fan speed parameters do not change.

$$\frac{\Delta T_{me}|_n}{\Delta T_{me}|_0} = \frac{\dot{Q}_e|_n}{\dot{Q}_e|_0} \tag{9}$$

$$\frac{\Delta T_{mc}|_n}{\Delta T_{mc}|_0} = \frac{\dot{Q}_c|_n}{\dot{Q}_c|_0} \tag{10}$$

Here, the sub character (o) signifies the known reference conditions and the sub character (n) signifies the changing outdoor temperature conditions.

The operating parameters at a different outdoor temperature were determined by estimation and iteration of the difference between airs entrance temperature T_{aci} and the refrigerants condensation temperature T_{rc} at condenser, and the difference between airs entrance temperature T_{aei} and the refrigerants evaporation temperature T_{re} at evaporator. SOLVER add-on in MS-EXCEL is used to solve this iteration, and COOLPROP plugin is used to define the states of air and refrigerant at each condition in every equipment. The matrix showing the solving procedure is illustrated in Figure 2.

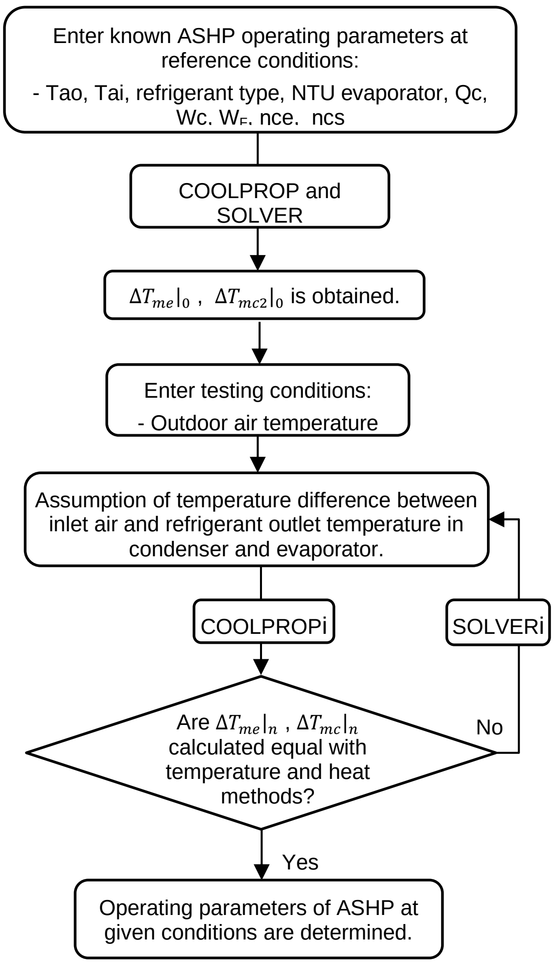

Figure 2. The solving procedure matrix

2.3. Validation

The validation of model was made by comparing the experimental COP data of ASHP published by the manufacturer, obtained at different outdoor temperatures and heating capacities, with the COP values calculated by the presented solution method. Calculated COP values and experimentally measured COP values are in agreement with absolute percentage error less than 3.2%, as shown in Table 2.

Table 2. Comparison of calculated COP values with experimental data

Tout	Condenser Heating load	COP (calc.)	COP (exp.)	Absolute percent error
-10°C	2.17 kW	3.42	3.34	2.4%
-5°C	2.56 kW	3.76	3.76	0.0%
0°C	2.94 kW	4,11	4.14	0.1%
7°C	3.40 kW	4.53	4.53	0.0%
10°C	3.71 kW	4.89	4.74	3.2%

3. RESULTS AND DISCUSSION

The operating parameters of an ASHP are determined at 2°C intervals in the outdoor temperature range of -10°C to 18°C under conditions where the compressor power and fan speeds remain constant at their nominal values. Under these conditions, a decrease in outside temperature from 7°C to -10°C causes a 25% decrease in COP value from 4.53 to 3.38. On the other hand, while the outside temperature rises to 18°C, the COP value increases by 21% to 5.49, as shown in Figure 3.

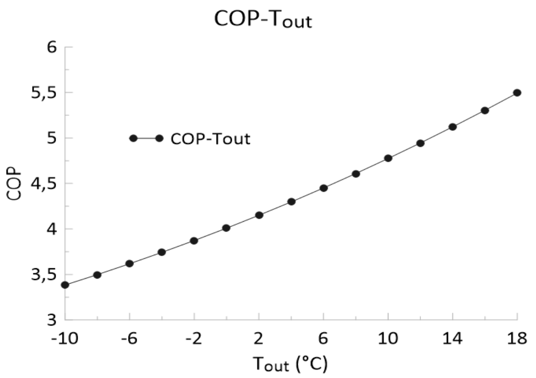

Figure 3. Variation of COP with respect to outside temperature.

As shown in Figure 4, the evaporation temperature and pressure of the refrigerant in the evaporator changed from -13.7°C, 5 bar to 11.3°C, 11.2 bar with the raise in the outdoor temperature from -10°C to 18°C. On the other hand, the pressure and temperature increase in the condensation is relatively less but still noticeable, from 33.3°C, 19.7 bar to 41.3°C, 24.4 bar.

Evaporator pressure can be measured by a manometer connected to the service valve in the low pressure line of ASHPs, where the refrigerant charge is also made. Unfortunately, some service personnel with insufficient technical knowledge use this pressure measurement alone as a shortcut to control the refrigerant charge level. Results of this study showing the variation of evaporator pressure with outdoor temperature also reveals how wrong it is to use the pressure value alone in refrigerant charge control. For this reason, in the service manuals of ASHP systems using thermal expansion valves, it is requested to control the charge with the subcooling method, which includes simultaneous monitoring of subcooling and superheat values. On the other hand, the developed model, which can determine the evaporator pressure values of ASHP depending on the outdoor air temperature, provides simple, fast and accurate charge control with the pressure measurement method, beside subcooling method. Also, the results show that the pressure variation is more pronounced in the evaporator, so the service valve in the low pressure line would be the best choice for the pressure method in charge control. In addition, it should be noted that the model considers the case of constant indoor temperature, compressor power and fan speeds, however it can still be applied to inverter ASHPs by running the compressor at rated power.

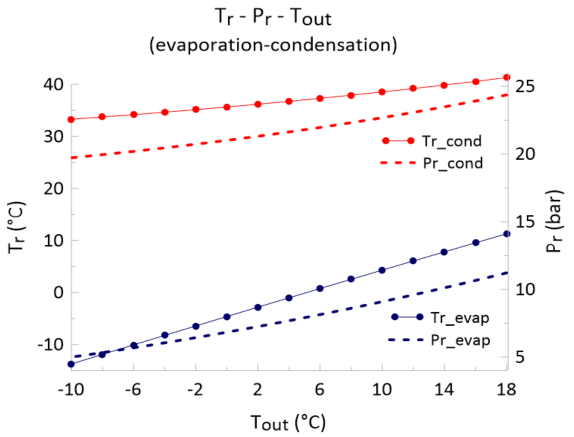

Figure 4. Change of refrigerants evaporation and condensation temperature and pressure according to outdoor air temperature.

The rise in evaporator pressure, which comes with increase in the outdoor temperature, elevates the condenser pressure as a result of constant compressor power. Rising condensing pressure and temperature also increase the condenser heating load. As shown in Figure 5, the condenser heating load increases from 2540 W to 4124 W, while the outdoor temperature rises from -10°C to 18°C.

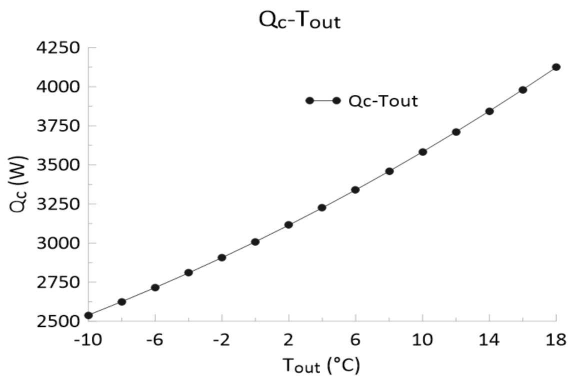

Figure 5. Change of condenser heating load according to outdoor air temperature.

4. CONCLUSION

In this paper, variation of operation parameters of a selected ASHP at constant compressor power according to the outdoor temperature was investigated with a calculation method based on the change at LMTD differences in the evaporator and condenser. The calculation method was validated by comparing COP results with manufacturer's experimental COP data. It is assumed that the fan speeds of evaporator and condenser do not change with conditions. The leading conclusions of the research are given below.

(1) The raise in outdoor temperature escalates the evaporation temperature and pressure of the refrigerant in the evaporator, thus increases the performance of the ASHP. When the outdoor temperature rises from -10°C to 18°C, the refrigerants evaporation temperature rises from -13.7°C to 11.3°C and the COP jumps from 3.38 to 5.49.

(2) A raise in the outdoor air temperature from -10°C to 18°C elevates the evaporation pressure from 5 bar to 11.2 bar. Determining the evaporation pressure change according to the outside air temperature with the developed model can lead to a simple and fast refrigerant charge control with only pressure check.

(3) The raise in outdoor temperature also increases the condenser heating load by raising the refrigerant condensing temperature. A rise in the outdoor air temperature from -10°C to 18°C, increases the refrigerants condensing temperature and pressure from 33.3°C, 19.7 bar to 41.3°C, 24.4 bar, and that results a enhance at condenser heating load from 2540W to 4124W.

In future studies, seasonal or daily performance variations of ASHPs can be determine, and refrigerant charge control charts with respect to outdoor air conditions can be presented. ASHPs using different refrigerants can be examined and compared. The outcomes of this paper can be a beneficial sample for researchers and field workers examining the operation parameter changes at ASHP.

DECLARATION OF ETHICAL STANDARDS

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

AUTHORS' CONTRIBUTIONS

Kutbay SEZEN: Developed the model, analysed the results, wrote the manuscript.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

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