ISSN:1306-3111
e-Journal of New World Sciences Academy
2010, Volume: 5, Number: 2, Article Number: 1A0091

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ENGINEERING SCIENCES
Received: September 2009
Accepted: March 2010
Series : 1A
ISSN : 1308-7231

THERMO-ECONOMIC OPTİMİZATION OF A VAPOUR COMPRESSED REFRIGERATION SYSTEM USING ALTERNATIVE REFRIGERANTS

ABSTRACT

This study deals with the thermo-economic optimization of superheating and sub-cooling heat exchangers in vapor compressed refrigeration system. Using for R22 and its alternative refrigerants (R410A and R407C), the temperature of condenser at the interval of (35°C – 55°C) and temperature of evaporator at the interval of (-10°C – 10°C) have been obtained from the calculation process. The second law analysis of refrigeration system is carried out and then the whole system is optimized by thermo-economically. As a result of calculations, optimum superheating and sub-cooling temperatures of heat exchangers areas corresponding to these temperatures are obtained.

Keywords: Thermo-Economy, Optimization, Refrigeration, Super Heating, Alternative Refrigerants (R410A and R407C)

BUHAR SIKIŞTIRMALI SOĞUTMA SİSTEMİNDE ALTERNATİF SOĞUTUCU AKIŞKAN KULLANILARAK TERMOEKONOMİK OPTİMİZASYON

ÖZET

Bu çalışmada, buhar sıkıştırmalı bir soğutma sisteminde aşırı soğutma ve aşırı kızdırma ısı değiştiricilerinin termo-ekonomik optimizasyonu yapılmıştır. Soğutucu akışkan R22 ve onun alternatifleri olan R410A ve R407C için kondenser sıcaklık aralığı (35°C – 55°C) ve evaporatör sıcaklık aralığı (-10°C – 10°C) alınarak hesaplamalar yapılmıştır. Soğutma sisteminin II. Kanun analizi yapılarak tüm sistem termo-ekonomik olarak optimize edilmiştir. Sonuç olarak, verilen sıcaklık aralıklarına göre optimum aşırı kızdırma ve aşırı soğutma ısı değiştiricileri alanları hesaplanmıştır.

Anahtar Kelimeler: Termo-Ekonomi, Optimizasyon, Soğutma, Aşırı Kızdırma, Alternatif Soğutucular (R410A ve R407C)

1. INTRODUCTION (GİRİŞ)

Thermo-economic optimization is the process of optimizing through analyzing the refrigeration system both thermodynamically and economically [1, 2, 3, 4, and 5]. To achieve optimum design and analysis of thermal systems, thermo-economic techniques are often used. Plenty of studies have been carried out on optimizing the operating parameters for better performance, minimum investment and operating costs [6, 7, and 8]. To further boost the system efficiency, superheating process in evaporator and sub-cooling process in condenser is performed [13]. New alternative refrigerants are searched for, which fit to the requirements in a heat pump, and refrigerant mixtures which are composed of HFC (hydro fluorocarbon) refrigerants having zero ODP (ozone depletion potential) are now being suggested as drop in or mid-term replacement [14].

The heat exchanger operation has been optimized through thermo-economic optimization of the heat pump by means of the CSB (Coefficient of Structural Bond) method by Koçoğlu [2]. Ammonia was employed in the heat pump and the evaporator temperature was taken between 0°C and 10°C, and the condenser temperature between 32°C and 44°C. Dingeç [3] has effected the thermo-economic optimization of a simple refrigerating system. R-12 was employed as refrigerant. It was presumed that no sub-cooling and superheating was present during the optimization. For optimum condenser temperatures of 47°C, 48°C and 49°C, optimum condenser areas of 1.41 m^2, 1.22 m^2 and 0.96 m^2 have been identified respectively.

Abou-Ziyan et al [4] have compared the widely used conventional heat pump with the solar assisted heat pump for various refrigerants (R22, R134A, R404A). Performance of the heat pump (COP) has augmented by 50% with R134A, leading to a saving of 25% in collector area. D'Accadia and F. De Rossi [5] have minimized all the operating and depreciation costs in a particular refrigerating system. Condenser and evaporator are equipped with a water-cooling mechanism where water chilling the condenser is cooled down by the cooling tower. R-22 has been employed as the refrigerant. In the article; compressor, condenser (pump, cooling tower, electric motor) and evaporator have been optimized. Upon calculations, optimum compressor isentropic efficiency has been determined as 0.83; optimum evaporator efficiency as 0.681; optimum condenser efficiency as 0.78 and optimum C.O.P as 4.5. Furthermore, after the optimization, a saving of 1.8% in investment and operating costs and 10.8% in electrical power consumption has been achieved.

The thermo-economic optimization of an irreversible Stirling heat pump cycle with a parametric study for the finite heat capacity of external reservoirs is presented by Tyagi et al [6]. He found that the effect of regenerative effectiveness and the economic parameter are more pronounced than that of the other parameters. The design optimization of a condenser is discussed, using a thermo-economic approach by d'Accadia [10]. A cost function to be minimized is the amortization cost of the condenser, related to the heat exchange area for R22, R134A and R410A. Dincer's [11] study deals with the thermo-economic optimization of vapor compression refrigeration systems along with the cost terms. Cost of the system is determined by using the annuity factor, the cost per mass flow rate of water and efficiency for condenser and evaporator. The variation of component cost versus component efficiency for evaporator temperature -10°C, -15°C and -20°C. Cabello et al. [14] studied how operating variables affect the energetic performance of a refrigeration plant from an experimental point of view when using R134A, R407C and R22 as working fluids.

2. RESEARCG SIGNIFICANCE (ÇALIŞMANIN ÖNEMİ)

In this study, the impact of superheating in the evaporator and subcooling in the condenser on the system's performance and cost has been surveyed, and optimum superheating and subcooling temperatures as well as the heat exchanger areas corresponding to these temperatures have been identified for alternative refrigerants (R407C and R410A).

3. EXERGY ANALYSIS (EKSERJI ANALİZİ)

Energy-based efficiencies sometimes fall inadequate and are misleading. For instance all energies in energy efficiency are taken as equal. On the other hand, energy may be of a different quality in practice. For example work is more precious than heat and likewise heat at a higher temperature is more precious than heat that at a lower temperature, because the whole work is transformed into heat, but on the contrary whole heat is not transformed into work [1, 2, and 3].

Irreversibility is defined as the difference between the reversible work and the useful work during a state change. Irreversibility implies the unconvertible energy that can be transformed into work. Exergy is the process of obtaining maximum work from a fluid having a particular energy through reducing it into the environmental conditions [2 and 3]. Accordingly, the exergy of the refrigerant flowing through our refrigeration system can be defined as follows with specific values:

$$e_x = (h - T_0 * s) + \frac{v^2}{2} + g * z - (h_0 - T_0 * s_0) \tag{1}$$

neglecting the potential and kinetic energies in equation (1),

$$e_x = (h - T_0 * s) - (h_0 - T_0 * s_0) \tag{2}$$

is obtained.

$$E_x = \dot{m}_R * e_x \tag{3}$$

To calculate the irreversibility of all system components in our refrigeration system, amounts of all input and output energy for each component should be calculated with the following equation.

$$\dot{W} = \sum (E_Q) + \sum (\dot{m}_R * e_x)_{input} - \sum (\dot{m}_R * e_x)_{output} - T_0 * s_{gen} \tag{4}$$

- **Internal Irreversibility of the Refrigeration System (Soğutma Sisteminin Toplam Tersinmezliği):**

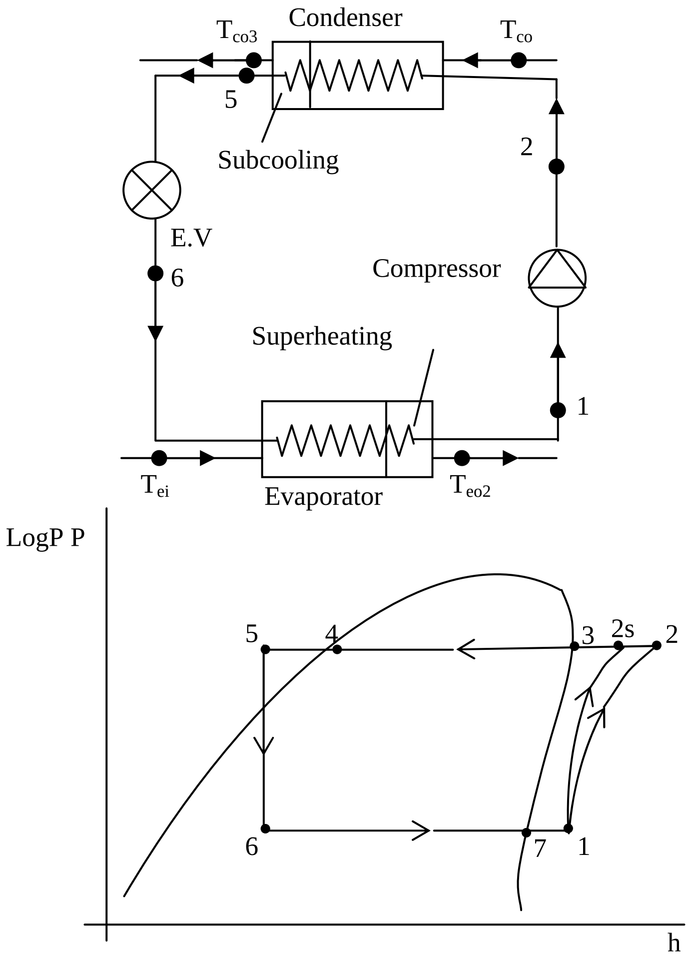

Figure 1. The vapor-compressed refrigeration system. Components and LogP-h fig
(Şekil 1. Buhar Sıkıştırmalı Soğutma Sistemi ve Log P-h Diyagramı)

Figure 1 shows the vapor-compressed refrigeration system which is components and LogP-h. The sub-cooling area (in condenser) and superheating area (in evaporator) are seen in this figure. Internal irreversibility of the refrigeration system is the total of the irreversibility value of each system component, namely $I_{i,i} = \sum I_{Systemcomponents}$ [2]. Using Eq. 2-4 is obtained Eq. 5-7 for condenser, Eq. 8, 9 for evaporator and Eq. 10 for compressor as fallows.

$$I_{C_I} = T_0 * \left[\dot{m}_c * (s_{co_1} - s_{ci}) - \dot{m}_R * (s_2 - s_3) \right] \tag{5}$$

$$I_{C_{II}} = T_0 * \left[\dot{m}_c * (s_{co_2} - s_{co_1}) - \dot{m}_R * (s_3 - s_4) \right] \tag{6}$$

$$I_{SC} = T_0 * \left[\dot{m}_c * (s_{co_3} - s_{co_2}) - \dot{m}_R * (s_4 - s_5) \right] \tag{7}$$

$$I_E = T_0 * \left[\dot{m}_e * (s_{eo_1} - s_{ei}) - \dot{m}_R * (s_6 - s_7) \right] \quad (8)$$

$$I_{SH} = T_0 * \left[\dot{m}_e * (s_{eo_2} - s_{eo_1}) - \dot{m}_R * (s_7 - s_1) \right] \quad (9)$$

$$I_c = \dot{m}_R * T_0 * (s_2 - s_1) \quad (10)$$

$$I_{E.V} = \dot{m}_R * T_0 * (s_6 - s_5) \quad (11)$$

$$I_{i,i} = I_{C_I} + I_{C_{II}} + I_{SC} + I_E + I_{SH} + I_c + I_{E.V} \quad (12)$$

The following equation is obtained by writing Eq. 5-11 into Eq. 12.

$$I_{i,i} = T_0 * \left[\dot{m}_c * (s_{co_3} - s_{ci}) + \dot{m}_e * (s_{eo_2} - s_{ei}) \right] \quad (13)$$

$$s_o - s_i = c_p * Ln \frac{T_o}{T_i} \quad (14)$$

$$\dot{m}_C = \frac{U_{C_I} * A_{C_I} * \Delta T_{m_{C_I}}}{c_{p_{cc}} * (T_{co_1} - T_{ci})} = \frac{U_{C_{II}} * A_{C_{II}} * \Delta T_{m_{C_{II}}}}{c_{p_{cc}} * (T_{co_2} - T_{co_1})} = \frac{U_{SC} * A_{SC} * \Delta T_{m_{SC}}}{c_{p_{cc}} * (T_{co_3} - T_{co_2})} \quad (15)$$

$$\dot{m}_E = \frac{U_E * A_E * \Delta T_{m_E}}{c_{p_{ce}} * (T_{ei} - T_{eo_1})} = \frac{U_{SH} * A_{SH} * \Delta T_{m_{SH}}}{c_{p_{ce}} * (T_{eo_1} - T_{eo_2})} \quad (16)$$

Eq. 17 is obtained by substituting Eq.14, 15 and 16 with Eq.13. Eq.17 is the final equation built up and yields the internal irreversibility of the system.

$$\frac{I_{i,i}}{T_0} = \frac{U_{C_I} * A_{C_I} * \Delta T_{m_{C_I}}}{T_{co_1} - T_{ci}} * Ln \frac{T_{co_1}}{T_{ci}} + \frac{U_{C_{II}} * A_{C_{II}} * \Delta T_{m_{C_{II}}}}{T_{co_2} - T_{co_1}} * Ln \frac{T_{co_2}}{T_{co_1}}$$

$$+ \dot{m}_C * c_{p_{cc}} * Ln \left(1 + \frac{U_{SC} * A_{SC} * \Delta T_{m_{SC}}}{\dot{m}_c * c_{p_{cc}} * T_{co_2}} \right) + \frac{U_E * A_E * \Delta T_{m_E}}{T_{ei} - T_{eo_1}} * Ln \frac{T_{eo_1}}{T_{ei}}$$

$$+ \dot{m}_E * c_{p_{ce}} * Ln \left(1 - \frac{U_{SH} * A_{SH} * \Delta T_{m_{SH}}}{\dot{m}_e * c_{p_{ce}} * T_{eo_1}} \right) \quad (17)$$

In vapor-compressed refrigeration system, optimization parameters and assumptions are provided below:

- Refrigerating capacity of the system, $\dot{Q}_E$ = 2 kW and constant.
- 2nd area of condenser inlet temperature, T_3 ranges between 35°C … 55°C.
- First area of evaporator output temperature, T_7 ranges between -10°C …10°C.
- For condenser and evaporator cooling water c_{pcc}= c_{pce} =4.181 kJ/kgK and constant.

- Input temperature of the condenser cooling water, T_{ci}= 20°C and constant.
- Input temperature for the evaporator cooling water, T_{ei}= 20°C and constant.
- Output temperature for the condenser cooling water (input to the sub-cooling exchanger), T_{co2}= 25°C and constant.
- Output temperature for the evaporator heating water (input to the superheating exchanger), T_{eo1}= 15°C and constant.
- All heat exchangers in the system (evaporator, condenser etc) are of telescopic tubular design (parallel flow).
- All calculations have separately been effected for the refrigerant R22, R410A and R407C.
- The values of thermodynamic properties were obtained using equations for R22 in Ref [4] and its alternative refrigerants (R407C and R410A) in Ref [12].

4. THERMO-ECONOMIC OPTIMIZATION (TERMOEKONOMİK OPTİMİZASYON)

Thermo-economy is the combination of the exergy analysis and the economical analysis. Thermo-economy strives to ensure the balance at minimum cost for any component in the system between its operating cost and exergy cost. It is difficult to perform thermo-economic implementations in complex systems. Therefore, simplified systems are considered for optimization. Employing the exergy analysis in thermo-economic implementations is quite useful. For instance, different components of the system may be self-optimized. The component given and the whole system can be recognized by referring to the local exergy input cost or exergy loss, and also irreversibility highly contributes to process perfection and identification of optimum system parameters [1, 6, and 8].

4.1. Coefficient Structural Bond (C.S.B) (Yapısal Bağ Katsayısı)

Coefficient structural bond is defined as follows [2]:

$$\sigma_{k,i} = \frac{\left(\dfrac{\partial I_T}{\partial x_i}\right)}{\left(\dfrac{\partial I_k}{\partial x_i}\right)} \tag{18}$$

Here, it is presumed that the system is in equilibrium and input energy is of constant quality. The impact of change in x_i alters the input exergy amount when output exergy is constant. In this study, this method has been employed.

4.2. Equivalence of Optimization (Optimizasyon Eşitliği)

In this optimization technique, electrical energy is supplied to the system by a compressor at exergy input. Operating cost is calculated through electrical energy consumption. Total annual operating costs have been specified below for optimization [2]:

$$M_T(x_i) = t_{op} * M_{input} * E_{input}(x_i) + a^C * \sum_{i=1}^{n} M_1^M(x_i) + b^M \tag{19}$$

$$\frac{\partial E_{input}}{\partial x_i} = \frac{\partial I_T}{\partial x_i} \tag{20}$$

$$\frac{\partial M_T}{\partial x_i} = t_{op} * M_{input} * \frac{\partial I_T}{\partial x_i} + a^C * \sum_{i=1}^{n} \frac{\partial M_1^M}{\partial x_i} \tag{21}$$

In equation (21), rebuilding the second term on the right side of the equivalence,

$$a^C * \sum_{i=1}^{n} \frac{\partial M_1^M}{\partial x_i} = \frac{\partial I_k}{\partial x_i} * \sum_{i'=1}^{n} \left[\frac{\partial M_{1'}^M}{\partial I_k} \right] \tag{22}$$

Applying equation (20) and (21) to the equation (19), then the equation is finalized as follows.

$$\frac{\partial M_T}{\partial x_i} = t_{op} * M_{k,i}^I * \frac{\partial I_k}{\partial x_i} + a^C * \frac{\partial M_k^M}{\partial x_i} \tag{23}$$

The x_i value that makes the result of the so-obtained equation (23) zero is the optimum value. In other words, as seen in equation (23), the x_i value that makes the derivative of the operating cost zero is the thermo-economically optimized value.

$$M_{k,i}^I = M_{input} * \sigma_{k,i} + \frac{a^C}{t_{op}} * \beta_{k,i} \tag{24}$$

If equation (23) is applied to equation (24) and the equation is equaled to zero, then the equation for optimization is obtained.

$$\left(\frac{\partial I_k}{\partial x_i} \right)_{Opt} = \left(- \frac{a^C}{t_{op} * M_{k,i}^I} \right) * \left(\frac{\partial M_k^M}{\partial x_i} \right) \tag{25}$$

The so-obtained equation (25) here yields the thermo-economical optimization according to the x_i parameter of the k^{th} component in the system.

4.3. Thermo-Economic Optimization Equivalence of the System Components (Sistem Elemanlarının Termoekonomik Optimizasyon Eşitliği)

4.3.1. Condenser (Kondenser)

a) Area I (superheating area) (Aşırı Kızdırma Alanı)

$$\left(\frac{\partial I_{C_I}}{\partial A_{C_I}} \right)_{Opt} = \left(- \frac{a^C}{t_{op} * M_{C_I,A_I}^I} \right) * \left(\frac{\partial M_{C_I}^M}{\partial A_{C_I}} \right) \tag{26}$$

where

$$M_{C_I,A_I}^I = M_{input} * \sigma_{C_I,A_I} + \frac{a^C}{t_{op}} * \beta_{C_I,A_I} \tag{27}$$

is calculated.

$$\sigma_{C_I,A_I} = \frac{\left(\frac{\partial I_T}{\partial A_{C_I}} \right)}{\left(\frac{\partial I_{C_I}}{\partial A_{C_I}} \right)} \tag{28}$$

$$\frac{\partial I_T}{\partial A_{C_I}} = \frac{\partial I_{C_I}}{\partial A_{C_I}} \tag{29}$$

In Eq.28, σ value is accepted as 1 at all areas of evaporator and condenser due to Eq.29.

$$\beta_{C_I,A_I} = \frac{\partial M_E^M}{\partial I_{C_I}} + \frac{\partial M_{SH}^M}{\partial I_{C_I}} + \frac{\partial M_{C_{II}}^M}{\partial I_{C_I}} + \frac{\partial M_{SC}^M}{\partial I_{C_I}} \quad (30)$$

The following equation is provided by calculations required at Eq.30.

$$\beta_{C_I,A_I} = \left[\frac{\frac{\partial M_E^M}{\partial A_E}}{\frac{\partial I_T}{\partial A_E}} + \frac{\frac{\partial M_{C_{II}}^M}{\partial A_{C_{II}}}}{\frac{\partial I_T}{\partial A_{C_{II}}}} + \frac{\frac{\partial M_{SC}^M}{\partial A_{SC}}}{\frac{\partial I_T}{\partial A_{SC}}} + \frac{\frac{\partial M_{SH}^M}{\partial A_{SH}}}{\frac{\partial I_T}{\partial A_{SH}}} \right] \quad (31)$$

The following equation is obtained by using Eq.26, 27 and 31.

$$\left(\frac{\partial I_{C_I}}{\partial A_{C_I}}\right)_{Opt} = \left(- \frac{\left(\frac{\partial M_{C_I}^M}{\partial A_{C_I}}\right)}{\frac{t_{op} * M_{input}}{a^C} + \left[\frac{\frac{\partial M_E^M}{\partial A_E}}{\frac{\partial I_T}{\partial A_E}} + \frac{\frac{\partial M_{C_{II}}^M}{\partial A_{C_{II}}}}{\frac{\partial I_T}{\partial A_{C_{II}}}} + \frac{\frac{\partial M_{SC}^M}{\partial A_{SC}}}{\frac{\partial I_T}{\partial A_{SC}}} + \frac{\frac{\partial M_{SH}^M}{\partial A_{SH}}}{\frac{\partial I_T}{\partial A_{SH}}} \right]} \right) \quad (32)$$

Equation (32) given here is the thermo-economic optimization equivalence based on the heat exchanger zone of Area I of the condenser.

b) Area II (Condensation area) (Yoğuşma Alanı)

$$\left(\frac{\partial I_{C_{II}}}{\partial A_{C_{II}}}\right)_{Opt} = \left(-\frac{a^C}{t_{op} * M_{C_{II},A_{II}}^I}\right) * \left(\frac{\partial M_{C_{II}}^M}{\partial A_{C_{II}}}\right) \quad (33)$$

where

$$M_{C_{II},A_{II}}^I = M_{input} * \sigma_{C_{II},A_{II}} + \frac{a^C}{t_{op}} * \beta_{C_{II},A_{II}} \quad (34)$$

is calculated.

$$\beta_{C_{II},A_{II}} = \frac{\partial M_E^M}{\partial I_{C_{II}}} + \frac{\partial M_{SH}^M}{\partial I_{C_{II}}} + \frac{\partial M_{C_I}^M}{\partial I_{C_{II}}} + \frac{\partial M_{SC}^M}{\partial I_{C_{II}}} \quad (35)$$

The following equation is provided by calculations required at Eq.35.

$$\beta_{C_{II},A_{II}} = \left[\frac{\frac{\partial M_E^M}{\partial A_E}}{\frac{\partial I_T}{\partial A_E}} + \frac{\frac{\partial M_{C_I}^M}{\partial A_{C_I}}}{\frac{\partial I_T}{\partial A_{C_I}}} + \frac{\frac{\partial M_{SC}^M}{\partial A_{SC}}}{\frac{\partial I_T}{\partial A_{SC}}} + \frac{\frac{\partial M_{SH}^M}{\partial A_{SH}}}{\frac{\partial I_T}{\partial A_{SH}}} \right] \quad (36)$$

The following equation is obtained by using Eq.33, 34 and 36.

$$\left(\frac{\partial I_{C_{II}}}{\partial A_{C_{II}}}\right)_{Opt} = \left(- \frac{\left(\frac{\partial M^{M}_{C_{II}}}{\partial A_{C_{II}}}\right)}{\frac{t_{op} * M_{input}}{a^{C}} + \left[\frac{\frac{\partial M^{M}_{E}}{\partial A_{E}}}{\frac{\partial I_{T}}{\partial A_{E}}} + \frac{\frac{\partial M^{M}_{C_{I}}}{\partial A_{C_{I}}}}{\frac{\partial I_{T}}{\partial A_{C_{I}}}} + \frac{\frac{\partial M^{M}_{SC}}{\partial A_{SC}}}{\frac{\partial I_{T}}{\partial A_{SC}}} + \frac{\frac{\partial M^{M}_{SH}}{\partial A_{SH}}}{\frac{\partial I_{T}}{\partial A_{SH}}} \right]} \right) \quad (37)$$

Equation (37) given here is the thermo-economic optimization equivalence based on the heat exchanger zone of Area II of the condenser.

c) Area III (Sub-cooling area) (Aşırı Soğutma Alanı)

$$\left(\frac{\partial I_{SC}}{\partial A_{SC}}\right)_{Opt} = \left(- \frac{a^{C}}{t_{op} * M^{I}_{SC,A_{SC}}} \right) * \left(\frac{\partial M^{M}_{SC}}{\partial A_{SC}} \right) \quad (38)$$

where

$$M^{I}_{SC,A_{SC}} = M_{input} * \sigma_{SC,A_{SC}} + \frac{a^{C}}{t_{op}} * \beta_{SC,A_{SC}} \quad (39)$$

is calculated.

$$\beta_{SC,A_{SC}} = \frac{\partial M^{M}_{E}}{\partial I_{SC}} + \frac{\partial M^{M}_{C_{I}}}{\partial I_{SC}} + \frac{\partial M^{M}_{C_{II}}}{\partial I_{SC}} + \frac{\partial M^{M}_{SH}}{\partial I_{SC}} \quad (40)$$

The following equation is provided by calculations required at Eq.40.

$$\beta_{SC,A_{SC}} = \left[\frac{\frac{\partial M^{M}_{E}}{\partial A_{E}}}{\frac{\partial I_{T}}{\partial A_{E}}} + \frac{\frac{\partial M^{M}_{C_{I}}}{\partial A_{C_{I}}}}{\frac{\partial I_{T}}{\partial A_{C_{I}}}} + \frac{\frac{\partial M^{M}_{C_{II}}}{\partial A_{C_{II}}}}{\frac{\partial I_{T}}{\partial A_{C_{II}}}} + \frac{\frac{\partial M^{M}_{SH}}{\partial A_{SH}}}{\frac{\partial I_{T}}{\partial A_{SH}}} \right] \quad (41)$$

The following equation is obtained by using Eq.38, 39 and 41.

$$\left(\frac{\partial I_{SC}}{\partial A_{SC}}\right)_{Opt} = \left(- \frac{\left(\frac{\partial M^{M}_{SC}}{\partial A_{SC}}\right)}{\frac{t_{op} * M_{input}}{a^{C}} + \left[\frac{\frac{\partial M^{M}_{E}}{\partial A_{E}}}{\frac{\partial I_{T}}{\partial A_{E}}} + \frac{\frac{\partial M^{M}_{C_{I}}}{\partial A_{C_{I}}}}{\frac{\partial I_{T}}{\partial A_{C_{I}}}} + \frac{\frac{\partial M^{M}_{C_{II}}}{\partial A_{C_{II}}}}{\frac{\partial I_{T}}{\partial A_{C_{II}}}} + \frac{\frac{\partial M^{M}_{SH}}{\partial A_{SH}}}{\frac{\partial I_{T}}{\partial A_{SH}}} \right]} \right) \quad (42)$$

Equation (42) given here is the thermo-economic optimization equivalence based on the sub-cooling exchanger zone of Area III of the condenser.

3.3.2. Evaporator (Evaporatör)

a) Area I (Evaporating area) (Buharlaştırma Alanı)

$$\left(\frac{\partial I_E}{\partial A_E}\right)_{Opt}=\left(-\frac{a^C}{t_{op}*M^I_{E,A_E}}\right)*\left(\frac{\partial M^M_E}{\partial A_E}\right) \tag{43}$$

where

$$M^I_{E,A_E}=M_{input}*\sigma_{E,A_E}+\frac{a^C}{t_{op}}*\beta_{E,A_E} \tag{44}$$

is calculated.

$$\beta_{E,A_E}=\frac{\partial M^M_{SC}}{\partial I_E}+\frac{\partial M^M_{C_I}}{\partial I_E}+\frac{\partial M^M_{C_{II}}}{\partial I_E}+\frac{\partial M^M_{SH}}{\partial I_E} \tag{45}$$

The following equation is provided by calculations required at Eq.45.

$$\beta_{E,A_E}=\left[\frac{\frac{\partial M^M_{SC}}{\partial A_{SC}}}{\frac{\partial I_T}{\partial A_{SC}}}+\frac{\frac{\partial M^M_{C_I}}{\partial A_{C_I}}}{\frac{\partial I_T}{\partial A_{C_I}}}+\frac{\frac{\partial M^M_{C_{II}}}{\partial A_{C_{II}}}}{\frac{\partial I_T}{\partial A_{C_{II}}}}+\frac{\frac{\partial M^M_{SH}}{\partial A_{SH}}}{\frac{\partial I_T}{\partial A_{SH}}}\right] \tag{46}$$

The following equation is obtained by using Eq.43, 44 and 46.

$$\left(\frac{\partial I_E}{\partial A_E}\right)_{Opt}=\left(-\frac{\left(\frac{\partial M^M_E}{\partial A_E}\right)}{\frac{t_{op}*M_{input}}{a^C}+\left[\frac{\frac{\partial M^M_{SC}}{\partial A_{SC}}}{\frac{\partial I_T}{\partial A_{SC}}}+\frac{\frac{\partial M^M_{C_I}}{\partial A_{C_I}}}{\frac{\partial I_T}{\partial A_{C_I}}}+\frac{\frac{\partial M^M_{C_{II}}}{\partial A_{C_{II}}}}{\frac{\partial I_T}{\partial A_{C_{II}}}}+\frac{\frac{\partial M^M_{SH}}{\partial A_{SH}}}{\frac{\partial I_T}{\partial A_{SH}}}\right]}\right) \tag{47}$$

Equation (47) given here is the thermo-economic optimization equivalence based on the heat exchanger zone of Area I of the evaporator.

b) Area II (Superheating area) (Aşırı Kızdırma Alanı)

$$\left(\frac{\partial I_{SH}}{\partial A_{SH}}\right)_{Opt}=\left(-\frac{a^C}{t_{op}*M^I_{SH,A_{SH}}}\right)*\left(\frac{\partial M^M_{SH}}{\partial A_{SH}}\right) \tag{48}$$

where

$$M^I_{SH,A_{SH}}=M_{input}*\sigma_{SH,A_{SH}}+\frac{a^C}{t_{op}}*\beta_{SH,A_{SH}} \tag{49}$$

is calculated.

$$\beta_{SH,A_{SH}}=\frac{\partial M^M_{SC}}{\partial I_{SH}}+\frac{\partial M^M_{C_I}}{\partial I_{SH}}+\frac{\partial M^M_{C_{II}}}{\partial I_{SH}}+\frac{\partial M^M_E}{\partial I_{SH}} \tag{50}$$

The following equation is provided by calculations required at Eq.50.

$$\beta_{SH,A_{SH}} = \left[\frac{\dfrac{\partial M_{SC}^{M}}{\partial A_{SC}}}{\dfrac{\partial I_T}{\partial A_{SC}}} + \frac{\dfrac{\partial M_{C_I}^{M}}{\partial A_{C_I}}}{\dfrac{\partial I_T}{\partial A_{C_I}}} + \frac{\dfrac{\partial M_{C_{II}}^{M}}{\partial A_{C_{II}}}}{\dfrac{\partial I_T}{\partial A_{C_{II}}}} + \frac{\dfrac{\partial M_{E}^{M}}{\partial A_{E}}}{\dfrac{\partial I_T}{\partial A_{E}}} \right] \quad (51)$$

The following equation is obtained by using Eq.48, 49 and 51.

$$\left(\frac{\partial I_{SH}}{\partial A_{SH}}\right)_{Opt} = \left(- \frac{\left(\dfrac{\partial M_{SH}^{M}}{\partial A_{SH}}\right)}{\dfrac{t_{op} * M_{input}}{a^{C}} + \left[\frac{\dfrac{\partial M_{SC}^{M}}{\partial A_{SC}}}{\dfrac{\partial I_T}{\partial A_{SC}}} + \frac{\dfrac{\partial M_{C_I}^{M}}{\partial A_{C_I}}}{\dfrac{\partial I_T}{\partial A_{C_I}}} + \frac{\dfrac{\partial M_{C_{II}}^{M}}{\partial A_{C_{II}}}}{\dfrac{\partial I_T}{\partial A_{C_{II}}}} + \frac{\dfrac{\partial M_{E}^{M}}{\partial A_{E}}}{\dfrac{\partial I_T}{\partial A_{E}}} \right]} \right) \quad (52)$$

Equation (52) given here is the thermo-economic optimization equivalence based on the superheating heat exchanger zone of Area II of the evaporator. Capital recovery factor is calculated with the following equation [1, 9]:

$$a^{C} = \frac{f_y * (1+f_y)^{N}}{(1+f_y)^{N} - 1} \quad (53)$$

where
f_y = 10% annual interest rate (%)
t_{op} = 800 hours/year,
M_{input} = 0.08 $/kW-h
a^C = 0.163 (calculated for an annual dollar interest rate of 10% with equation (53))
N = 10 years (period for which the system pays itself off)

$M_E^M = M_{SH}^M = 323.3 * A_E + 83.07$ $ [15] (54)

$M_{C_I}^M = M_{C_{II}}^M = M_{SC}^M = 525.96 * A_K + 530.18$ $ [15] (55)

5. RESULTS AND DISCUSSION (SONUÇLAR VE TARTIŞMA)

The variations of internal irreversibility (I_T, kW) by condenser and evaporator temperatures are showed in Fig. 3-6. Figure 2 shows the variation of coefficient of performance (COP) by condenser temperatures for each refrigerant. When condenser temperature increases, COP decreases. Figure 3, Figure 4 and Figure 5 show the variation of internal irreversibility by condenser and evaporator temperatures for each refrigerant. When condenser temperature increases, internal irreversibility also increases whereas when evaporator temperature increases, internal irreversibility decreases. Figure 6 shows the variation of irreversibility for all refrigerants. The descending order of the refrigerants for their irreversibility values is R22, R07C and R410A.

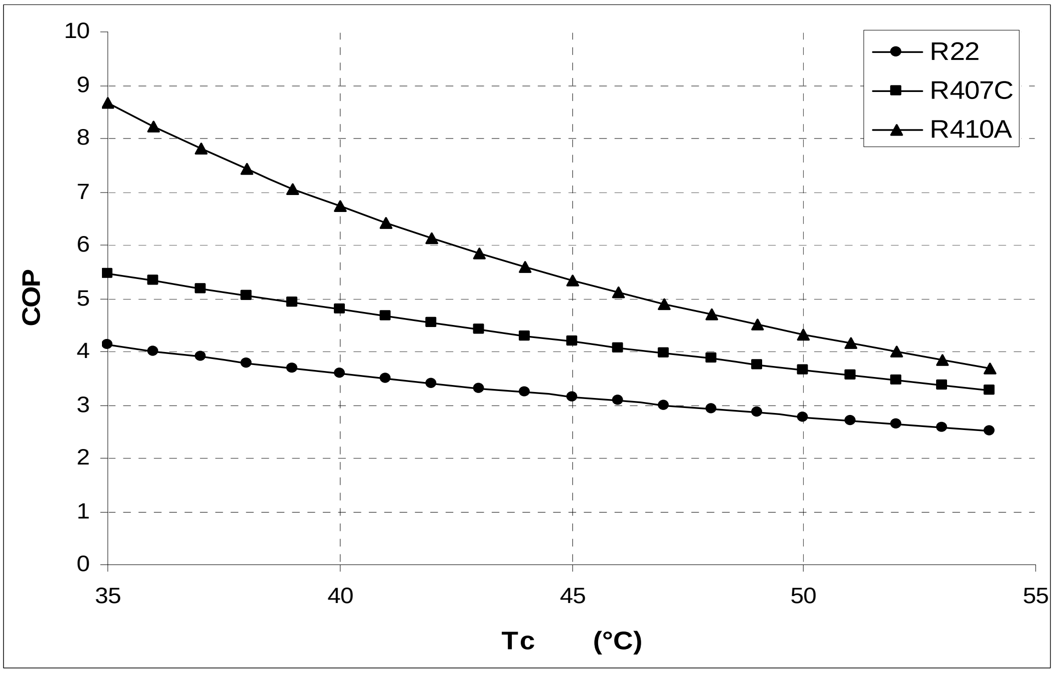

Figure 2. Variation of coefficient of performance (COP) by condenser temperatures (ΔT_{SH}=5°C, ΔT_{SC}=5°C)
(Şekil 2. Soğutma Tesir Katsayısının Kondenser Sıcaklığına Göre Değişimi)

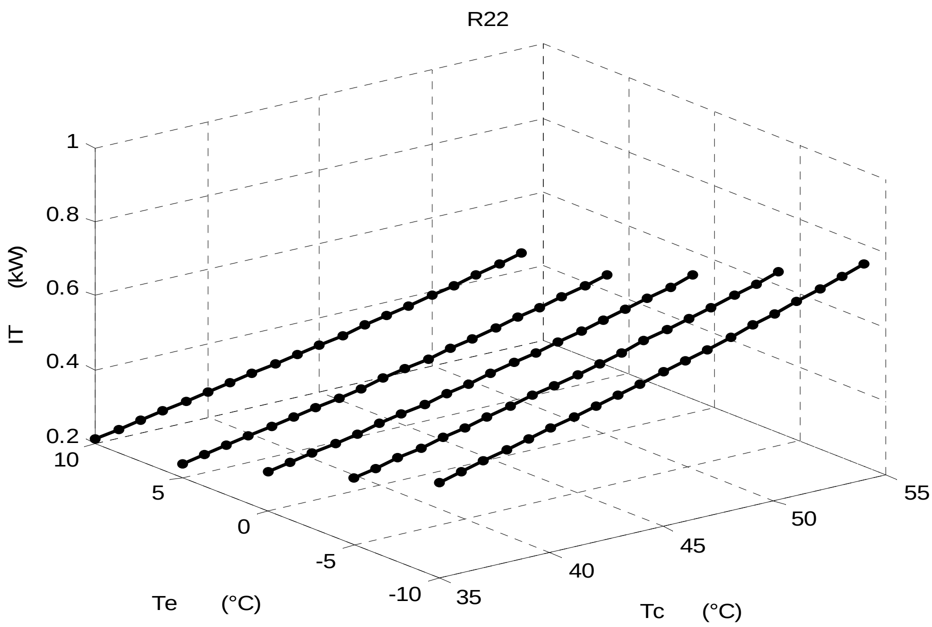

Figure 3. Variation of internal irreversibility (I_T, kW) by condenser and evaporator temperatures (R22)(ΔT_{SH}=5°C, ΔT_{SC}=5°C)
(Şekil 3. Toplam Tersinmezliğin (I_T, kW) Kondenser ve Evaporatör Sıcaklığına Göre Değişimi) (R22)

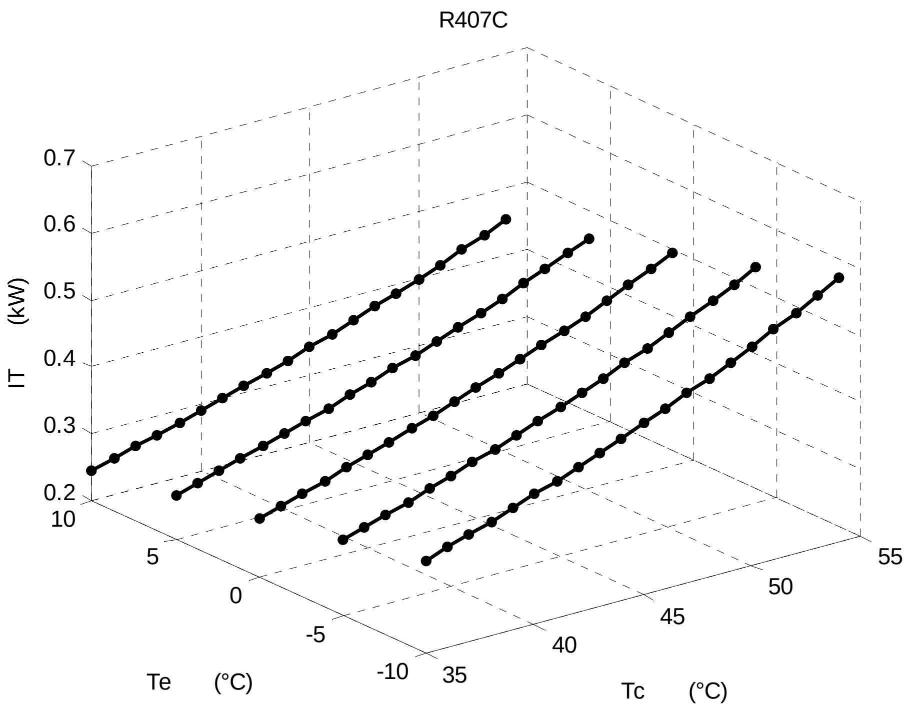

Figure 4. Variation of internal irreversibility (I_T, kW) by condenser and evaporator temperatures (R407C) (ΔT_{SH}=5°C, ΔT_{SC}=5°C)
(Şekil 4. Toplam Tersinmezliğin (I_T, kW) Kondenser ve Evaporatör Sıcaklığına Göre Değişimi) (R407C)

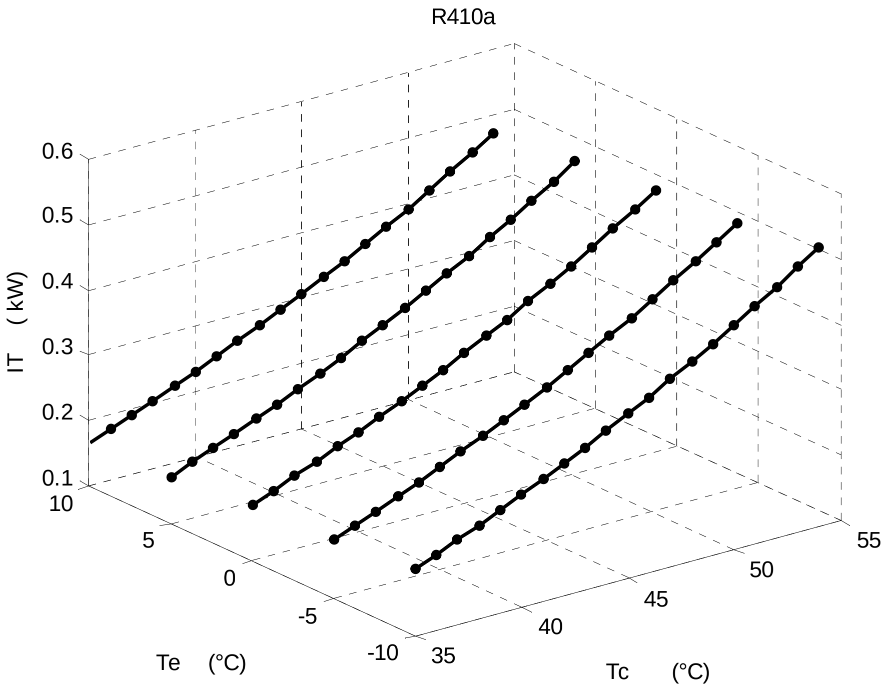

Figure 5. Variation of internal irreversibility (I_T, kW) by condenser and evaporator temperatures (R410A) (ΔT_{SH}=5°C, ΔT_{SC}=5°C)
(Şekil 5. Toplam Tersinmezliğin (I_T, kW) Kondenser ve Evaporatör Sıcaklığına Göre Değişimi) (R410A)

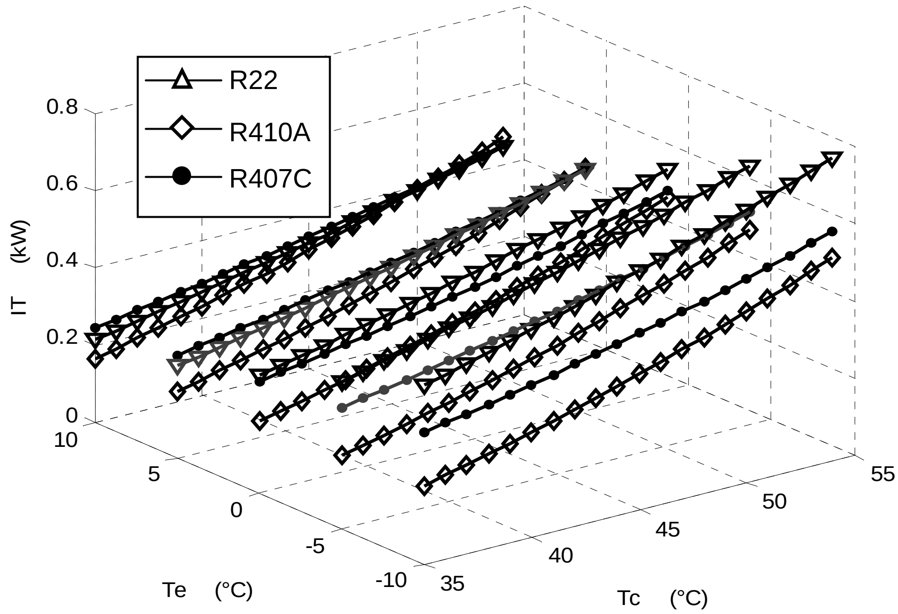

Figure 6. Variation of internal irreversibility (I_T, kW) by all refrigerants and condenser and evaporator temperatures (ΔT_{SH}=5°C, ΔT_{SC}=5°C)
(Şekil 6. Toplam Tersinmezliğin (I_T, kW) Tüm Soğutucu Akışkanlar İçin Kondenser ve Evaporatör Sıcaklığına Göre Değişimi)

This study has been employed for identifying the thermodynamic (irreversibility) and economical (investment and operating costs) optimization of the refrigeration system. Thermo-economic optimization results of condenser's 1st, 2nd and 3rd areas together with the evaporator's 1st, 2nd areas have been obtained through utilizing the design parameters given in equation (26), (27), (28), (29) and (30) and effecting computer-aided iteration. As a result of the optimization, best heat exchanger areas and the corresponding best temperature values for alternative refrigerants (R410A and R407C) have been identified. The results of the thermo-economical optimization of the system are shown in Table 1, Table 2 and Table 3. For instance, for an evaporator temperature of 7°C in a system where R22 refrigerant is utilized, thermo-economically optimized condenser temperature is 43°C, sub-cooling temperature is 5.1°C and superheating temperature is 5.0 °C. In other example, for an evaporator temperature of 7°C in a system where R407C refrigerant is utilized, thermo-economically optimized condenser temperature is 48°C, sub-cooling temperature is 5°C and superheating temperature is 5 °C. In another example, for an evaporator temperature of 7°C in a system where R410A refrigerant is utilized, thermo-economically optimized condenser temperature is 37°C, sub-cooling temperature is 7°C and superheating temperature is 7°C.

In this study, thermo-economic optimization is conducted in both sub-cooling heat exchanger and super heating heat exchanger. Variation of internal irreversibility to exchanger area has been found and cost analysis has made for this area. When a comparison is made according to thermo-economic optimization; A_{SH}=0.1086 m^2 of super heating area and A_{SC}=0.0397 m^2 of sub-cooling exchanger area for T_E= 7°C at R22. A_{SH}=0.2024 m^2 of super heating area and A_{SC}=0.0398 m^2 of sub-cooling exchanger area for T_E= 7°C at R407C. A_{SH}=0.1604 m^2 of super heating area and A_{SC}=0.033 m^2 of sub-cooling exchanger area for T_E= 7°C at R410A.

Thus, the lowest exchanger areas for the same evaporator temperature are R22, R410A and R407C respectively for super heating exchanger. At the same conditions, the order is R410A, R22 and R407C for sub-cooling exchanger.

Table 1. Thermo-economically optimized condenser, sub-cooling and superheating temperatures and exchanger areas for various evaporator temperatures (R22)
(Tablo 1. Çeşitli evaporatör sıcaklıkları için termoekonomik olarak optimize edilmiş kondenser, aşırı soğutma, aşırı kızdırma sıcaklıkları ve eşanjör alanlarının değişimi (R22)

	T_E=7°C		T_E=8°C		T_E=9°C	
T_C (°C)	43	61	40	49	36	39
ΔT_{sc} (°C)	5.1	3.7	3.2	2.6	3.2	2.5
ΔT_{sh} (°C)	5	2	4	2	3.2	2.5
A_C (m^2)	0.18643	0.104568	0.20942	0.14463	0.254547	0.21369
A_{sc} (m^2)	0.03974	0.01733	0.02788	0.0153	0.037716	0.022733
A_E (m^2)	0.323672	0.323672	0.35933	0.35933	0.40409	0.40409
A_{sh} (m^2)	0.108683	0.037836	0.09292	0.0397	0.080992	0.059652

Table 2. Thermo-economically optimized condenser, sub-cooling and superheating temperatures and exchanger areas for various evaporator temperatures (R407C)
(Tablo 2. Çeşitli evaporatör sıcaklıkları için termoekonomik olarak optimize edilmiş kondenser, aşırı soğutma, aşırı kızdırma sıcaklıkları ve eşanjör alanlarının değişimi (R407C)

	T_E=5°C		T_E=7°C		T_E=9°C	
T_C (°C)	39	47	37	48	37	41
ΔT_{sc} (°C)	7	5	8	5	4	3
ΔT_{sh} (°C)	9	8	7	5	4	3
A_C (m^2)	0.2282	0.1748	0.2542	0.1722	0.2544	0.2125
A_{sc} (m^2)	0.0891	0.0417	0.1322	0.03985	0.0648	0.0349
A_E (m^2)	0.27031	0.27031	0.3237	0.3237	0.40409	0.40409
A_{sh} (m^2)	0.298	0.221	0.319	0.2024	0.1888	0.197

Table 3. Thermo-economically optimized condenser, sub-cooling and superheating temperatures and exchanger areas for various evaporator temperatures (R410A)
(Tablo 3. Çeşitli evaporatör sıcaklıkları için termoekonomik olarak optimize edilmiş kondenser, aşırı soğutma, aşırı kızdırma sıcaklıkları ve eşanjör alanlarının değişimi (R410A)

	T_E=5°C		T_E=7°C		T_E=9°C	
T_C (°C)	42	51	37	52	38	42
ΔT_{sc} (°C)	9	8	7	5	4	3
ΔT_{sh} (°C)	9	8	7	5	4	3
A_C (m^2)	1.2473	1.017	1.4938	1.111	1.47	1.3121
A_{sc} (m^2)	0.078	0.06570	0.1238	0.033	0.0105	0.0306
A_E (m^2)	0.27031	0.27031	0.3237	0.3237	0.4041	0.4041
A_{sh} (m^2)	0.186	0.155	0.2016	0.1604	0.1974	0.1683

6. CONCLUSION (SONUÇ)

In vapor-compressed refrigeration systems, performance is improved with increase in superheating and sub-cooling. However, as phase transformation of the refrigerant ends and it turns into superheating steam form in the section that provides superheating in the evaporator. Heat transfer rapidly reduces and need to additional surface increases. Results have been obtained for alternative refrigerants (R410A and R407C). Descending order of refrigerants by the amount of irreversibility is R22, R407C and R410A. After the optimization, best heat exchanger areas and the corresponding best temperature values for all refrigerants have been identified. The lowest exchanger areas for the same evaporator temperature are R22, R410A and R407C respectively for super heating exchanger; the order is R410A, R22 and R407C for sub-cooling exchanger. Analyzes in the study will contribute to implementations to be performed in the future for new refrigerants in the thermo-economic optimization of vapor-compressed refrigeration systems.

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NOMENCLATURE (KISALTMALAR)

a^C	:	Capital recovery factor
b^M	:	Annual cost of non-effected from optimization ($)
c_p	:	Specific heat capacity (kJ kg^{-1} K^{-1})
E	:	Exergy (kW)
E.V	:	Expansion valve
g	:	Gravitational acceleration (m s^{-2})
h	:	Enthalpy (kJ kg^{-1})
I	:	Irreversibility (kW)
$\dot{m}$	:	Mass flow rate (kg s^{-1})
K	:	Overall heat transfer coefficient (kW m^{-2}K^{-1})
P	:	Pressure (bar)
Q	:	Heat transfer rate (kW)
S_{gen}	:	Entropy generation (kW K^{-1})
s	:	Specific entropy (kJ kg^{-1}K^{-1})
T	:	Temperature (°C)
U	:	Overall heat transfer coefficient (kWm^{-2}K^{-1})
V	:	Velocity (m s^{-1})
W	:	Work (kJ kg^{-1})
x_i	:	System parameter
z	:	Local high (m)
$M^I_{k.i}$	:	Local unit cost of irreversibility ($ W^{-1})
M^C	:	Capital cost ($)
M_T	:	Total operating cost of year ($)
M^M_1	:	Investment cost of system components (1…n) ($)
M^M_E	:	Investment cost of evaporator ($)
M^M_{SH}	:	Investment cost of super heating exchanger ($)
M^M_{CI}	:	Investment cost of condenser (area I) ($)
M^M_{CII}	:	Investment cost of condenser (area II) ($)
M^M_{SC}	:	Investment cost of sub-cooling exchanger ($)

Greek letters

$\sigma_{k.i}$	:	Coefficient of structural bonds of element k with respect to parameter i.
$\beta_{k.i}$	:	Capital cost coefficients of element k wrt. parameter i
ΔT_m	:	Logarithmic mean temperature difference (LMTD)
ΔT_{SC}	:	Sub-cooling temperature (°C)
ΔT_{SH}	:	Super heating temperature (°C)
Σ	:	Summation function

Subscripts

C : Condenser
C_I : Area I (Condenser)
C_{II} : Area II (Condenser)
c : Compressor
cc : Condenser cooling
ce : Evaporator cooling
co_1 : Condenser (Area I) output
co_2 : Condenser (Area II) output
co_3 : Subcooling output
E : Evaporator
E.V. : Expansion valve
İ : Inside
eo_1 : Evaporator output
eo_2 : Superheating output
i,i : Internal irreversibility
k : k^{th} component
o : Outside
op : Operate
opt : Optimum
R : Refrigerant
SC : Subcooling
SH : Superheating
T : Total
y : Year
0 : Reference condition

Superscript

I : Irreversibility