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Research Article

ANALYSIS OF THE HEAT PUMP PERFORMANCE OF DIFFERENT REFRIGERANTS IN A BUILDING WITH DETERMINED HEATING LOAD ACCORDING TO TS 825

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In this study, the performance of a geothermal heat pump used in heating mode for a house whose heating energy need is determined according to the Turkish thermal insulation standard TS 825, with different refrigerants has been theoretically analyzed. Accordingly, the annual heating energy need was calculated by taking the building components of the house into consideration, and it was determined that the heat pump condenser to meet this need should have a capacity of 15 kW. In addition, 7 different refrigerants with zero ozone depletion potential (ODP) in the heat pump system were annalized at variable evaporator temperatures and the performance of the geothermal heat pump was investigated. In the study, thermodynamic analysis was applied to the system by using Hydrochlorofluorocarbons (HFC) refrigerant types R 134a, R 143a, R 404A, R 407A, R 407C, R 410A and R 507A refrigerants for the determined operating conditions of the geothermal heat pump. Solway Solkane 8.0 calculation program was used for analysis. As a result of the analysis, it has been determined that the lowest operating loads and the highest heating effect coefficient values for the same operating conditions will be obtained from the system using R 134a, R407A, R 407C refrigerants, respectively. Moreover it was determined that the refrigerant with the highest pressure ratio value was R 134a.

Key words*: Refrigerant, Heat pump, Performance analysis, Heating effect, TS 825*

1. Introduction

The negative climate changes brought by the developing technology and industrialization, the use of air conditioning and cooling systems have become indispensable. Correspondingly, these systems constitute approximately 9% of the world energy consumption. Due to the increase of environmental problems such as global warming, the refrigerants used in these systems are inevitable to be environmentally friendly. Therefore, the search for alternative fluids has accelerated in place of fluids that are harmful to the environment with the increasing concern about global warming. Fluids that will be considered as alternative fluids will be fluids with low global warming potential (GWP) and low ozone depletion potential (ODP). CFC (Chlorofluorocarbons) and HCFC (Hydrochlorofluorocarbons), which have high GWP and ODP values, have become more and more limited to use since they damage the ozone layer. Accordingly, HFC (Hydrofluorocarbons) refrigerants have been developed as an alternative to these refrigerants [1-2].

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Heat pumps have a wide range of uses, from home refrigerators to cold storage. Heat pumps that are used for heating or cooling purposes take heat from the medium that low temperature that is and give it to the medium that is high temperature [3]. They usually do this with the vapor compression refrigeration cycle. It is possible to come across various studies in the literature for heat pumps and using refrigerants. Some of these studies, Çamdalı et al. The system parameters were calculated by applying numerical model techniques for the ground source heat pump in the study conducted by. The solution codes of the equations obtained from the model developed for the city of Bolu are written in the Matlab program. In the curve fitting method, ASHRAE table values and state equations of R 12, R 22 and R 502 are used to obtain the properties of R 134a, R 404A and R 410A and as a result, COP value of R 134a is 3.33 in Matlab program and in Solkane Refrigerant Software program, It was found to be 3.28 [4]. In the study conducted by D'Accadia and Vanoli, R 22, R 134a and R 410A were used as refrigerant in a vapor compression refrigeration system and the outdoor unit element was optimized using numerical analysis. An optimum heat exchanger area has been calculated with constant thermal capacity and variable inner tube diameter [5]. Boran et al. Experimentally investigated the performance of R 134a /R 152a refrigerant mixtures in the heat pump. Performance values were calculated using equations defined by thermodynamics 1st law. In the results obtained, they showed that the performance coefficients improved between 5% and 23% in heating and 6% and 28% in cooling due to the increase in R 152a refrigerant ratio in the mixtures [6]. Xu et al. In their study, they compared the performance of R 410A and R 32 refrigerants used in heat pump systems. They said that the use of the R 32 coolant, which has a lower global warming potential, is 10% and 9% better in terms of capacity and COP, respectively, than the use of the R 410A coolant [7]. In their study, Alkan et al. Conducted a thermodynamic study of alternative refrigerant for the ground source heat pump for space heating in Isparta province conditions. In their studies, they selected R 22, R 404A, R 410A, R 407C, R 134a and R 600 refrigerants as refrigerants and calculated performance parameters such as COP, exergy efficiency and total irreversibility of the system for certain conditions [8]. Çomaklı et al. Conducted research on cooling fluids in the project related to the investigation of energy and exergy efficiencies in heat pumps using zeotropic gas mixtures for Erzurum province. They investigated the use of alternative fluids (R 134a, R 407C, R 404A) instead of the restricted R 22 refrigerant and modeled the performance of these fluids in heat pump systems [9]. In a study by Yang et al., R 12, R 134a, R 410A and R 717 have applied thermodynamic analysis to various condensation and evaporation processes in the vapor compression refrigeration system using cooling fluids [10].

In this study, the heating load of a residence with a gross area of 100 m^2, located in the 2nd region and considered to be in the city center of Tekirdağ province, according to the Turkish thermal insulation standard TS 825, was calculated by considering the building's structural components. A theoretical analysis of the heat pump performance was carried out for seven different refrigerants at three different evaporator temperatures at a fixed condenser capacity selected according to the obtained heating load. In this respect, this study differs from other studies in the literature. The study is expected to be effective in the design of heat pump devices and refrigerants selection.

2. Material and Method

The study is divided into two parts. The annual heating energy need of the house examined in the first part was calculated considering the building components according to TS 825 [11] and the condenser capacity was determined accordingly. In the second part, heat pump performance analysis of

7 different refrigerants was made for varying evaporator temperatures according to the determined condenser capacity.

2.1. Calculation of annual heating energy needs of investigation building

It has been accepted that the building whose heat pump performance will be investigated is located in the city center of Tekirdağ, which is located in the 2nd Region according to TS 825. In Figure 1, details of the structural condition of the examined building and the examined part of the heat pump are given. In order to calculate the heating energy need of a building according to TS 825, the components of the building and the location of the building must be taken into account. In addition, the insulation made on the outer shells will gain great importance in increasing the efficiency of the energy spent in the air conditioning of the buildings. If insulation materials are used in correct and sufficient quantities, material, energy and costs will be saved [1-3].

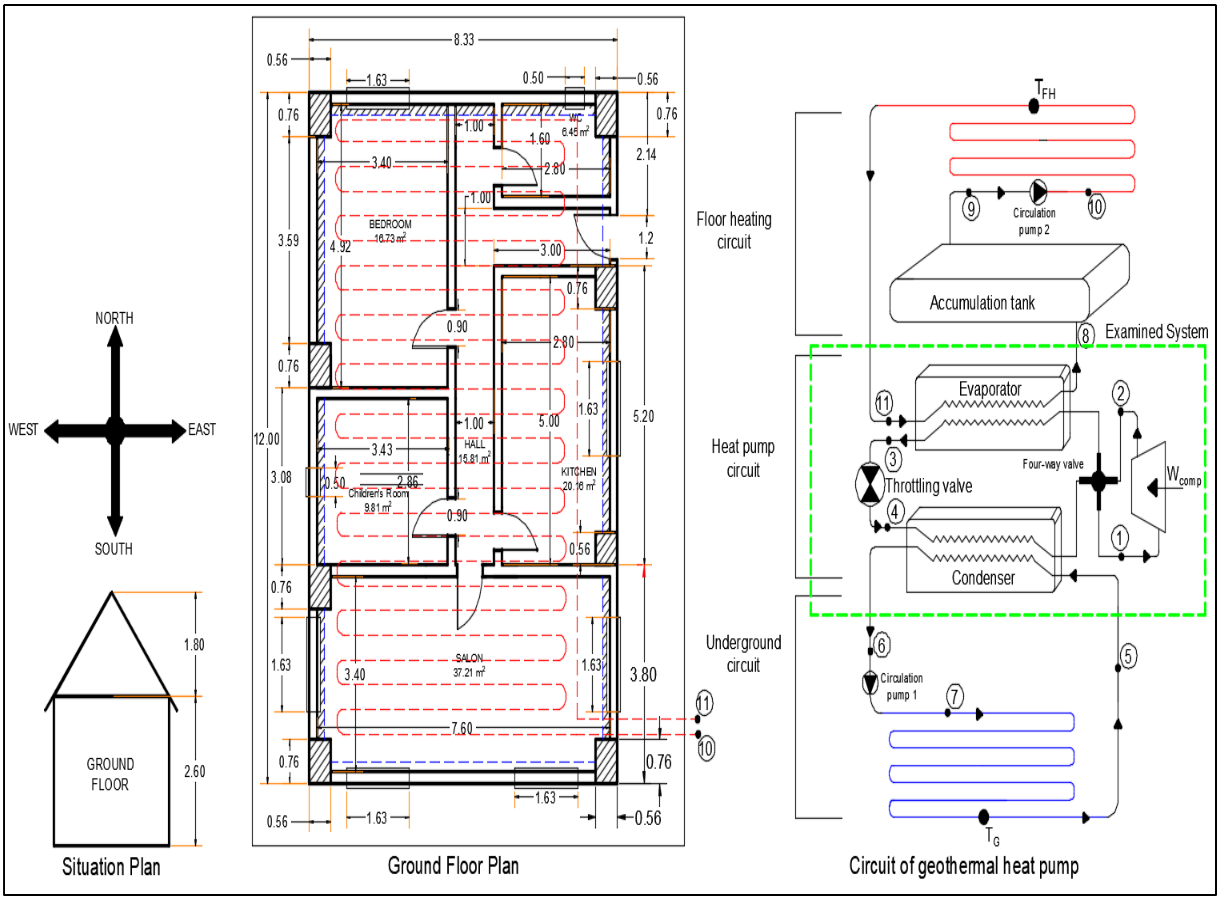

Figure 1. Floor plan of the examined building with circuit of geothermal heat pump.

When the floor plan of the building taken as an example in Figure 1 is examined, it is seen that it has a usage area of 90.36 m^2 and a height of 2.6 m. The surface area of reinforced (column and beam) walls open to outside air is 10.4 m^2, and the surface area of unreinforced (brick) walls is 82.24 m^2. In addition, there are 7 plastic joinery double glazed windows with a total surface area of 11.98 m^2 and non-heat insulated metal exterior doors with a surface area of 2.4 m^2. The surface area of the ground contact base and the unused roof (hipped roof) is 100 m^2. For the thermal insulation of the building components, 6 cm thick rock wool was used between the roof and 4 cm thick extruded Polystyrene foam

(EPS) was used in other building components. In the calculations, the building components that make up the building were examined separately and these are presented in Table 1.

Table 1. Types and components of building elements

Constructional component	Constructional element component (Inside-Out)	Thickness (m)	Heat conduction coefficient (W/mK)
Reinforced outer wall (A=10.4 m^2)	No aggregate, only gypsum used interior plaster	0.02	0.51
	Reinforcement	0.5	2.5
	Extruded polystyrene foam (TS 11989 – EN 13164)	0.04	0.03
	Lime-cement mortar exterior plaster	0.03	1.0
Unreinforced outer wall (A=82.24 m^2)	Non-aggregate exterior plaster with only gypsum used	0.02	0.51
	Vertical perforated brick (TS EN 771-1)	0.19	0.5
	Extruded polystyrene foam (TS 11989 – EN 13164)	0.04	0.03
	Lime-cement mortar exterior plaster	0.03	1.0
Unused roof space (hipped roof) (A=100 m^2)	No aggregate, only gypsum used interior plaster	0.02	0.51
	3 rows hollow concrete block	0.23	1.3
	Reinforcement	0.07	2.5
	Mineral and vegetable thermal insulation material (rock wool)	0.06	0.035
	Vertical air layer	0.25	0.139
	Plywood (TS 1047)	0.02	0.13
	Tile	0.03	0.81
Earth contact base (foundation) (A=100 m^2)	Crystalline igneous and metamorphic stones (marble)	0.03	2.3
	Cement mortar screed	0.04	1.4
	Polymer bituminous waterproofing blanket	0.06	0.19
	Extruded polystyrene foam (TS 11989 – EN 13164)	0.03	0.03
	Unreinforced concrete	0.1	1.65
	Sand-gravel ground cover	0.1	2.0
	Clay, alluvium	0.2	1.5

In this case, the equations used to determine the annual heating energy need of the building are presented below [11].

The annual heat energy need of the space can be calculated with the help of equation numbered 1.

$$Qyear=\Sigma Q_{month}\ ,(kJ) \qquad (1)$$

$$\Sigma Q_{month}=[H(\Theta i-\Theta e)-\eta_{monthly}\ (\Phi i,_{monthly}+\Phi s,_{monthly})]\ \ x\ t \qquad (2)$$

In Equation 2, *H* is the specific heat loss of the building, *i* and *e* monthly average internal and external temperature value, $\Phi i,_{monnthly}$ is the average monthly internal gain value and $\Phi s,_{monnthly}$ is the monthly average solar heat gain value, $\eta_{monthly}$ is the monthly average usage factor for earnings, and t is the time in seconds.

The specific heat loss *(H)* of the building is calculated by the sum of heat loss (H_T) through conduction and convection and heat loss (H_V) through ventilation.

$$H=H_T+H_V \tag{3}$$

$$H_T=\sum AU+IU_I \tag{4}$$

U in Equation 4; is the total heat transfer coefficient (W/m^{2}K). I; thermal bridge length, *UI*; is the linear permeability (W/mK) of the thermal bridge.

$$\sum AU=U_D.A_D + U_p.A_p + U_k.A_k + 0.8.U_T.A_T + 0.5\ U_t.A_t + U_d.A_d + 0.5.U_{ds}.A_{ds} \tag{5}$$

Sub-indices in the formula given in Equation 5 are *D*; outer walls, p; windows, *k;* outer door, *T;* ceiling, *t;* floor in contact with the ground, *d*; earth contact wall, *ds;* It refers to the building component in contact with the wall. Heat loss through natural ventilation can be calculated using Equation 6.

$$H_V=\rho.c.V' = \rho.c.n_h.V_h = 0,33.n_h.V_h \tag{6}$$

In Equation 6, n_h; air exchange rate, V_h; refers to the ventilated volume.

2.2. Calculation of heat pump performance

The flow chart of the evaluated geothermal heat pump system is given in Figure 1. The schematic diagram for the heating mode of the system considered in Figure 1 is shown. The cycle forming the system works as follows. (5-6) heat source, (1-4) refrigerant (Heat pump) and (7-8) heated space. Heat pump cycle; It consists of compressor, condenser, evaporator and expansion valve. The fluid selected as the refrigerant enters the compressor and compresses to the desired pressure and temperature, and becomes an superheated vapor, then enters the condenser. In the condenser, the refrigerant condenses due to the heat drawn.

First law analysis of the geothermal heat pump cycle shown in Figure 1. According to the first law of thermodynamics, the amount of heat transferred from the condenser in the geothermal heat pump system can be determined as follows;

$$\dot{Q}_{Condenser} = \dot{m}_{ref}(h_2 - h_3) \tag{7}$$

The heat load in the evaporator is expressed as follows;

$$\dot{Q}_{Evaporator} = \dot{m}_{ref}(h_1 - h_4) \tag{8}$$

The refrigerant flow rate circulating in the system is expressed as follows;

$$\dot{m}_{ref} = \frac{\dot{Q}_{Evaporator}}{(h_1 - h_4)} \tag{9}$$

We can express the working load of the compressor of the heat pump with the following equation;

$$\dot{W}_{compressor} = \dot{W}_{net} = \dot{m}_{ref} \frac{(h_2 - h_1)}{\eta} \tag{10}$$

Here, η value refers to isentropic efficiency. In this study, isentropic efficiency was determined to be 0.850. The pressure ratio of the system is expressed in Equation 11.

$$p_r = \frac{p_{high}}{p_{low}} \tag{11}$$

System performance coefficient (ITK_{HP}) is expressed as the ratio of condenser load to compressor power. The performance of the heat pump system is determined by the COP_{HP} value. Calculated with COP_{HP} Equation 12 [13].

$$COP_{HP} = \frac{\dot{Q}_{Condenser}}{\dot{W}_{compressor}} = \frac{\dot{Q}_{Condenser}}{\dot{W}_{net}} \tag{12}$$

The refrigerants used as an alternative in the system were selected from the Solakane 8.0 program, and the thermophysical and thermodynamic properties of the refrigerants are given in Table 2.

Table 2. Thermophysical properties of refrigerants used in analysis

Refrigerant	R 134a	R 143a	R 404A	R 407A	R 407C	R 410A	R 507A
Chemical Formula	CH_2FCF_3	CH_3CF_3	CHF_2CF_3/ CH_3CF_3/ CH_2FCF_3	CHF_2CF_3/ CH_2F_2/ CH_2FCF_3	CHF_2CF_3/ CH_2F_2/ CH_2FCF_3	CH_2F_2/ CHF_2CF_3	CHF_2CF_3/ CH_3CF_3
Critical Pressure (bar)	40.6	37.8	37.4	45.15	46.3	47.7	37.1
Critical temperature (ºC)	101.1	72.9	72.1	82.25	86.0	70.2	70.7
Boiling point (ºC)	-26.1	-47.2	-46.6	-45.0	-43.8	-51.6	-47.1
Ozone Depletion Potential	0	0	0	0	0	0	0
Global Warming Potential	1300	3800	3260	2100	1530	1730	3300
Molecular weight (kg/kmol)	102.0	84.0	97.6	90.11	86.2	72.6	98.9

3. Findings and Discussion

3.1. Analysis of annual heating energy needs of investigation building

The floor plan of the building taken as an example for the purpose of calculating the annual heating energy need is given in Figure 1, and the technical features of the building elements that make up this structure are given in Table 1. By using the formulas presented in Section 2.1 according to TS 825, the specific heat loss values to be realized in the building elements were calculated and these values are presented in Table 3.

Tablo 3. Specific heat loss of building elements

Building element	Thermal conductivity resistance R (m^2K/W)	Thermal permeability coefficient U (W/m^2K)	Heat losing surface area A (m^2)	Specific heat loss A x U (W/K)
Reinforced wall	1.773	0.577	10.4	6.00
Unreinforced wall	1.953	1.615	82.24	132.81
Unused attic	4.493	0.253	100	25.32
Earth contact base	2.103	0.649	100	64.97
Windows	-	3.4	11.98	40.75
Exterior door	-	5.5	2.4	13.2

As can be seen in Table 3, the total specific heat loss occurring in the building elements is 283.08 W/K. With the help of Equation 6, the heat loss realized through the ventilation in the structure examined with the help of equation 6 was calculated as 549.12 W/K. In this case, the total specific heat loss (H) in the building was found to be 832.2 W/K. The heating energy requirement of the building on a monthly basis during the heating season is presented in Table 4.

Tablo 4. Heating energy need of the building by months

Months	Heat loss			Heat gain			Gain usage rate	Gain utilization factor	Heating energy requirement
	Specific heat loss	Temp. Diff.	Heat losses	Internal heat gain	Solar gain	Total			
	$H = H_T + H_V$ (W/K)	$\Theta i - \Theta e$ (K, °C)	$H(\Theta i - \Theta e)$ (W)	Φi, (W)	Φs (W)	$\phi_T = \phi_i + \Phi s$ (W)	γ	η_{ay}	Qmonth (kJ)
Jan	832.2	16.1	133,98.42	4160	581	4741	0.26	0.98	22,685,806.08
Feb		14.6	12,150.12		730	4890	0.32	0.96	19,325,226.24
March		11.7	9,736.74		881	5041	0.44	0.90	13,477,985.28
Apr		6.2	5,159.64		998	5158	0.89	0.67	4,416,197.76
May		1.0	832.2		1155	5315	5.97	0.00	0
Jun		0.0	0		1215	5375	0.00	0.00	0
July		0.0	0		1183	5343	0.00	0.00	0
Aug		0.0	0		1104	5264	0.00	0.00	0
Sept		0.0	0		924	5084	0.00	0.00	0
Oct		4.9	4,077.78		741	4901	0.96	0.65	2,312,400.96
Nov		10.5	8,738.1		555	4715	0.39	0.92	11,405,577.60
Dec		15.2	12,649.44		507	4667	0.26	0.98	20,932,421.76

As can be seen from Table 4, the annual heating energy need of the examined building was determined as 94,555,615.68 kJ (26,286.46 kWh) during the 7-month heating season. Assuming that the heat pump system will operate 8 hours a day during the heating season, it has been determined that the condenser capacity of the heat pump should be 15.64 kW and a 15 kW condenser with a tolerance of ± 4-5% would be appropriate.

3.2. Analysis of heat pump performance

Table 5. Operating parameters used in cycle analysis of the geothermal heat pump.

Condenser load (kW)	15
Evaporator temperature range (°C)	5 – 15
Compressor isentropic efficiency	0.85
Condenser temperature (°C)	40
Suction line pressure loss (bar)	0.001
Discharge line pressure loss (bar)	0.001
Alternative refrigerants	R 134a, R 143a, R 404A, R 407A, R 407C, R 410A, R 507

The required heating load for the inspected building has been found and after selecting the condenser capacity according to this load, the operating parameters of the heat pump used in the study are presented in Table 5.

Performance analyzes were made in the geothermal heat pump cycle of different refrigerants at variable evaporator temperatures. The results obtained for different refrigerants as a result of the analyzes made are given in Table 6.

Table 6. Results obtained from the cycle analysis of the geothermal heat pump.

Determined Parameters for Geothermal Heat Pump		Condenser Capacity and Temperature (15 kW and 40 ºC) — Variable Evaporator Temperature		
Refrigerant	**Value**	**5 ºC**	**10 ºC**	**15 ºC**
R 134a	Pressure Ratio	2.91	2.45	2.08
	Mass Flow (g/s)	83.96	84.93	85.26
	W_{net} (kW)	2.27	1.93	1.58
	COP_{HP}	5.58	6.78	8.46
	Evaporator Capacity (kW)	12.7	13.1	13.4
R 143a	Pressure Ratio	2.54	2.19	1.90
	Mass Flow (g/s)	94.29	94.68	95.85
	W_{net} (kW)	2.44	2.06	1.69
	COP_{HP}	5.16	6.28	7.85
	Evaporator Capacity (kW)	12.6	12.9	13.3
R 404A	Pressure Ratio	2.59	2.22	1.92
	Mass Flow (g/s)	107.66	108.74	109.88
	W_{net} (kW)	2.46	2.08	1.71
	COP_{HP}	5.08	6.21	7.78
	Evaporator Capacity (kW)	12.5	12.9	13.3
R 407A	Pressure Ratio	2.79	2.37	2.02
	Mass Flow (g/s)	81.39	82.03	83.33
	W_{net} (kW)	2.33	1.96	1.62
	COP_{HP}	5.46	6.63	8.26
	Evaporator Capacity (kW)	12.7	13.0	13.4
R 407C	Pressure Ratio	2.82	2.39	2.04
	Mass Flow (g/s)	76.32	76.99	78.27
	W_{net} (kW)	2.34	1.98	1.63
	COP_{HP}	5.42	6.58	8.20
	Evaporator Capacity (kW)	12.7	13.0	13.4
R 410A	Pressure Ratio	2.59	2.23	1.93
	Mass Flow (g/s)	76.77	77.88	79.66
	W_{net} (kW)	2.44	2.06	1.71
	COP_{HP}	5.16	6.25	7.79
	Evaporator Capacity (kW)	12.6	12.9	13.3
R 507A	Pressure Ratio	2.56	2.21	1.91
	Mass Flow (g/s)	111.93	109.48	114.09
	W_{net} (kW)	2.49	2.04	1.73
	COP_{HP}	5.03	6.14	7.70
	Evaporator Capacity (kW)	12.5	12.5	13.3

As a result of the analyzes made for the cycle of the geothermal heat pump with the Solkane 8.0 program, the pressure ratio values obtained depending on the variable evaporator temperatures of the different refrigerants are given in Figure 2.

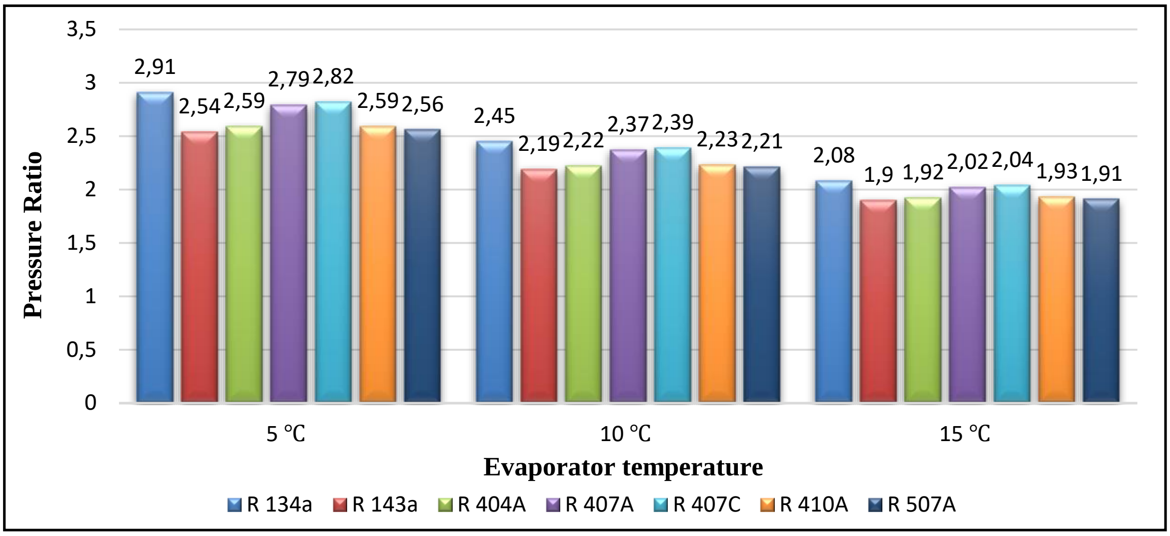

Figure 2. Pressure ratio change of geothermal heat pump for different refrigerants at variable evaporator temperatures.

The ratio of compressor outlet pressure to compressor inlet pressure gives the compressor pressure ratio. For this reason, there is a linear proportion between the increase in the pressure ratio of the compressor and the power consumed by the compressor. In this case, as the pressure ratio of the compressor increases, the energy consumed by the compressor will increase. Figure 2 shows the pressure ratio change in the compressor when 7 different refrigerants are used according to three different evaporator temperatures. It is seen that the pressure ratio values decrease for all refrigerants with the increase of the evaporator temperature. This situation can be explained by the increase in the compressor inlet pressure as the evaporator temperature increases, and this causes the pressure difference in the compressor to decrease. Since the compressor outlet pressure will be constant, the compressor inlet pressure that increases with the temperature increase will decrease the compressor pressure ratio.

Among the refrigerants examined, it was determined that the refrigerant with the highest pressure ratio was R 134a, and the refrigerant with the lowest pressure ratio was R 143a at all evaporator temperatures. Mass flow values obtained depending on variable evaporator temperatures in the analyzes performed are given in Figure 3.

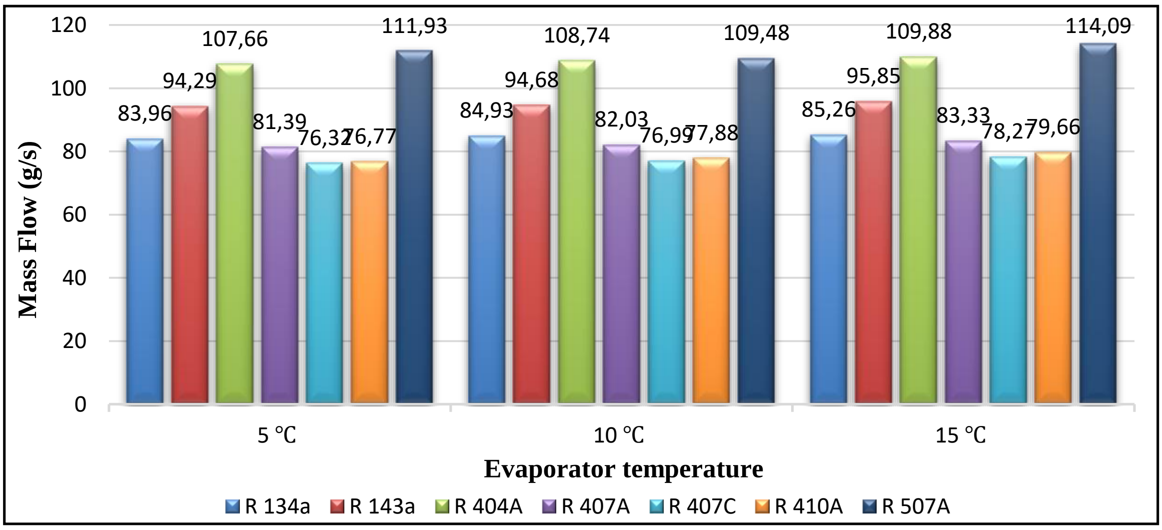

Figure 3. Mass flow change of geothermal heat pump for different refrigerants at variable evaporator temperatures.

When Figure 3 is examined, it can be seen that the increase in the evaporator temperature under the same operating conditions causes a slight increase in the mass flow rate of the refrigerant circulating in the heat pump system. The increase in evaporator temperature will cause the temperature of the refrigerant at the compressor inlet to increase. The temperature of the refrigerant compressed in the compressor will increase more depending on the inlet temperature. In this case, there will be a slight increase in the mass flow of the refrigerant whose specific volume decreases, that is, its density increases. Among the refrigerants examined, it was determined that the refrigerant with the highest mass flow was R 507A, and the refrigerant with the lowest mass flow rate was R 407C. Also, compressor working load values ($W_{compresor} = W_{net}$) obtained for three different evaporator temperatures are given in Figure 4.

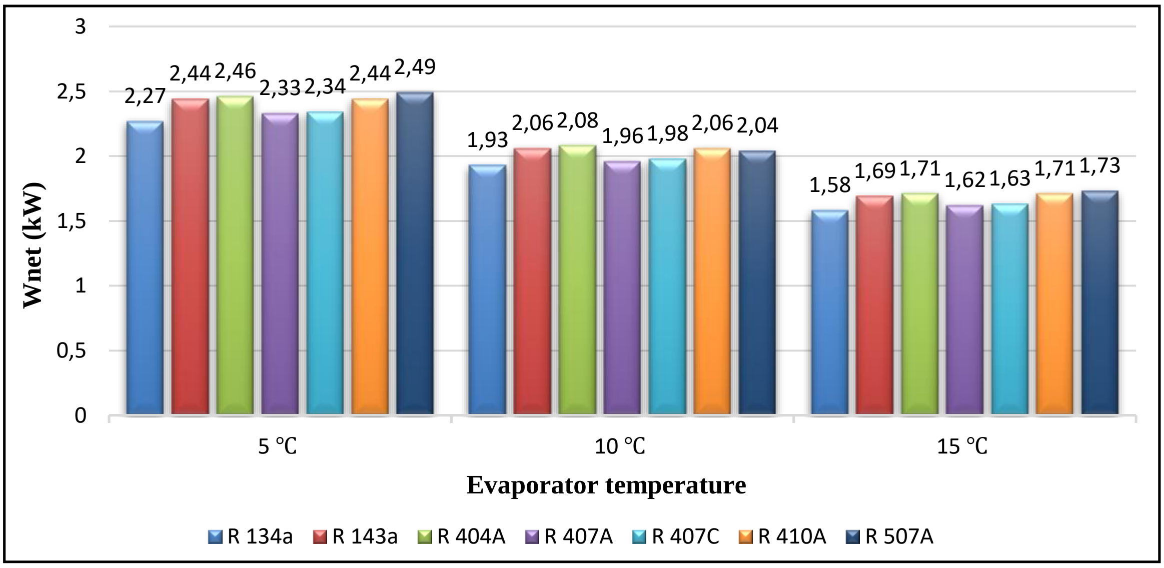

Figure 4. Highest working load change of geothermal heat pump for different refrigerants at variable evaporator temperatures.

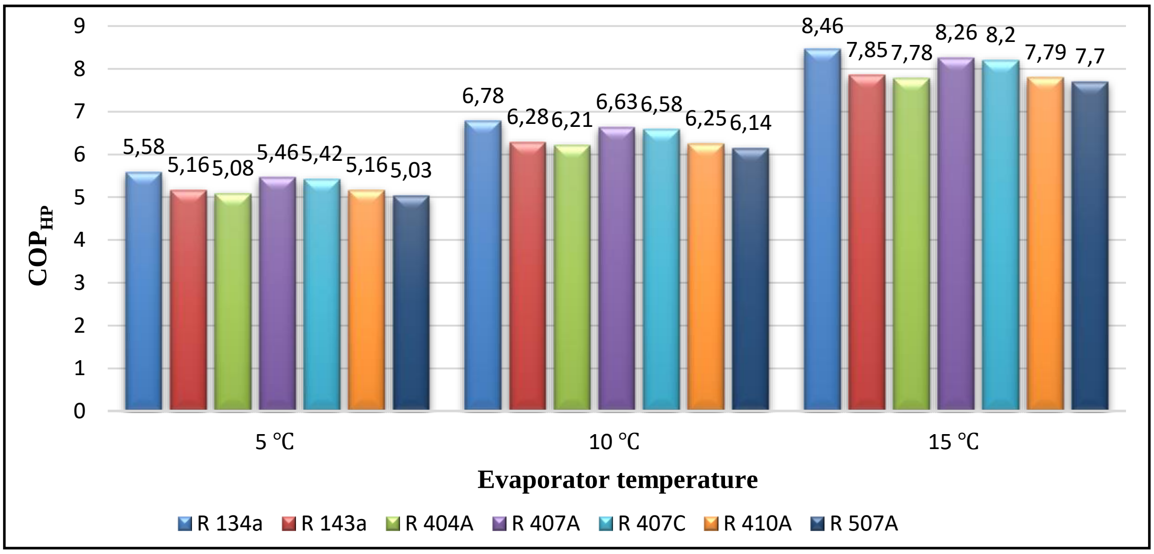

Figure 5. Cooling effect coefficient of geothermal heat pump for different refrigerants at variable evaporator temperatures.

Figure 4 shows the graph of W_{net} values obtained from the refrigerants examined at three different evaporator temperatures. It can be seen from the figure that the W_{net} values decrease with the increase of the evaporator temperature. The increase in evaporator temperature causes some increase in the mass flow rates of the refrigerants. Nevertheless, the increase in evaporator temperature causes an increase in compressor inlet enthalpy values and a decrease in compressor outlet enthalpy values, causing a decrease in compressor work. Among the refrigerants examined, the highest W_{net} value belongs to R 507A under all operating conditions, while the smallest W_{net} value belongs to R 134a refrigerant. In addition, for the heat pump system consisting of a 15 kW capacity condenser and a compressor with 85% isentropic efficiency, when the eaporator temperature is increased by 5 ºC, it has been determined that the W_{net} values decrease by approximately 15-30% for all refrigerants. The heating effect coefficient (COP_{HP}) values obtained for variable evaporator temperatures are given in Figure 5.

The term COP, which is the performance indicator of the heat pump system, can be evaluated as a function of the heat absorbed by the indoor unit and the power drawn by the compressor. In this case, COP will increase with the increase of the evaporator temperature and decrease with the increase of the condenser temperature. When Figure 5 is examined, the best heating effect coefficient values for the same operating conditions were obtained with R134a, R 407A, R 407C refrigerants. The lowest heating effect coefficient values are determined for R507 and R 404A refrigerants. The condenser capacities obtained for all cooling gases in the study are very close to each other and are presented in Figure 6.

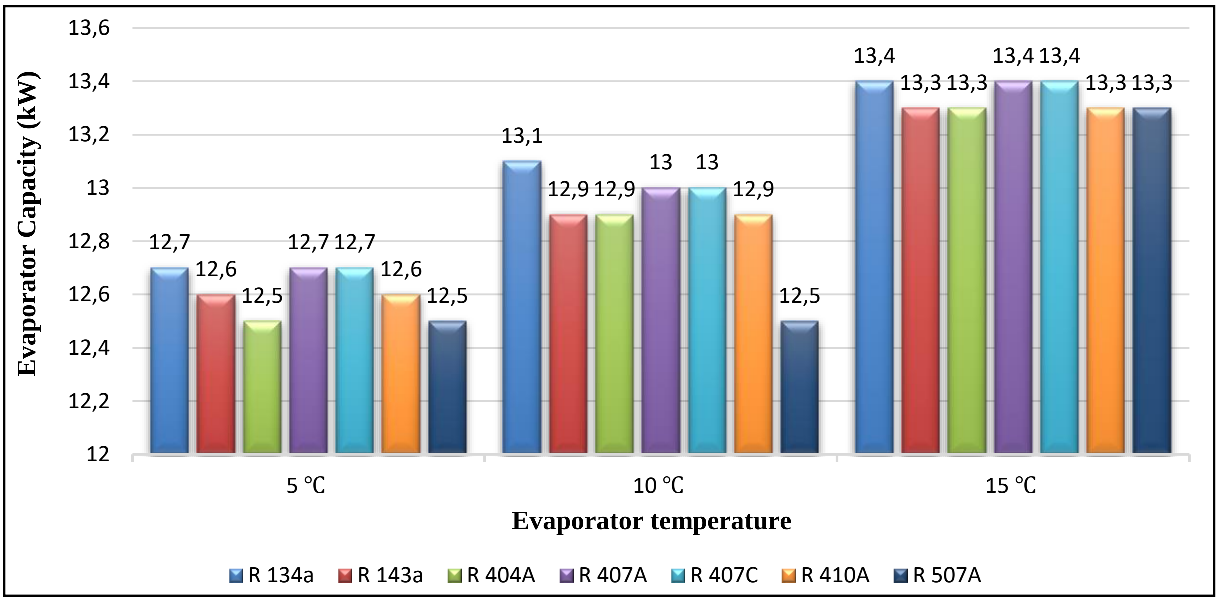

Figure 6. Evaporator capacity variation for different fluids of geothermal heat pump.

When Figure 6 is examined, it is seen that the evaporator capacity changes at three different evaporator temperatures. It can be seen that the evaporator capacity increases for all refrigerants with the increase of the evaporator temperature. Evaporator capacities for refrigerants evaluated under the same operating conditions are very close to each other.

4. Conclusions

In the upcoming periods, besides the increase in energy demand in the world, environmental factors that are expected to cause difficulties in terms of efficient use of energy and energy supply will be an important factor in the use and sustainability of energy. Therefore, the main goal should be to use energy in the most efficient way for the selection of air conditioning systems. In this context, determining the optimum system parameters to be used has become a necessity. In the selection of alternative fluids that can be used in geothermal heat pump systems, the working load of the system and the heating effect coefficient should be considered.

In this study, refrigerants with zero ozone depletion potential were investigated. In this respect, the lowest operating loads and the highest heating effect coefficient values were obtained when R 134a, R 407A, R 407C fluids were used, respectively. In addition, SES 36 refrigerant with zero ozone depletion potential was examined under the specified conditions, but it was determined that it would have a two-phase structure at the compressor outlet under these conditions, and it was concluded that it would not be suitable for use. In addition, the pressure ratio and mass flow values that the examined fluids will have in the operation are important factors in system design.

Nomenclature

GWP : Global warming potential
ODP : Ozone depleting potential
CFC : Chlorofluorocarbons
HCFC : Hydrochlorofluorocarbons
HFC : Hydrofluorocarbons
Ref : Refrigerant
COP : Coefficient of performance
$\dot{m}$:Mass flow(gs^{-1})
h :Enthalpy ($kjkg^{-1}$)
p :Pressure(bar)
comp :Compressor
p_r :Pressure ratio
HP :Heat pump
$\dot{Q}$:Heat flux(kW)

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