The performance evaluation of R744 transcritical ejector and R290/R744 cascade refrigeration systems for Turkey

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THE PERFORMANCE EVALUATION OF R744 TRANSCRITICAL EJECTOR AND R290/R744 CASCADE REFRIGERATION SYSTEMS FOR TURKEY

by

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In this study, performance of two different environmentally friendly systems with natural refrigerant solutions, R744 transcritical booster system with ejector and a R290/R744 cascade system are examined theoretically by using engineering equation solver software. Operating conditions are determined to represent different climatic regions in Turkey using summer dry bulb temperatures of various cities. The transcritical and the cascade system are assumed to operate at two different evaporation temperatures of –10 °C and –32 °C. The overall energy efficiency ratio values for each system with respect to the same ambient and evaporation conditions are compared and evaluated. Finally, performance of both systems has been compared and the appropriate solution for each city has been suggested. For cold and mild climate regions of Turkey, the performance of transcritical alternative is found better than the proposed cascade system. Moreover, the performance of transcritical system is observed slightly lower than those of the cascade system in hot climate regions of Turkey such as Aegean, Mediterranean, and South-East Anatolian regions. It is also found that the performance of the transcritical system is better in regions having lower ambient conditions such as near the Black Sea and eastern regions of Turkey. Therefore, for the mild and cold regions of Turkey, the transcritical ejector option is the better alternative due to having higher performance compared to the cascade system.

Key words: *natural refrigerants, transcritical ejector system, cascade system, energy efficiency ratio*

Introduction

Environmentally friendly refrigeration systems are readily available and using numerous regions on the world, especially in Europe. Refrigeration industry faces strict regulations regarding the use of the most fluorinated refrigerants. To control emissions from fluorinated GHG (F-gases), including hydrofluorocarbons (HFC), the EU has adopted the *F-gas Regulation* which covers applications in which F-gases are used. The EU Member States are responsible for implementing Regulation (EU No 517/2014). Also, in October 2016 international controls on the use of HFC were put in place via the Montreal Protocol, these will lead to a global phase-down in HFC use. In the case of commercial refrigeration, evidently the strong options are hydrocarbon propane (R290) and CO_2 (R744). The R744 is already widely used in

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supermarket refrigeration systems and is now available for a range of industrial refrigeration applications. In large supermarkets refrigeration R744 cascade systems are an alternative to common HFC systems in cold and moderate climates. Hydrocarbons also have proven to be highly efficient alternatives in most applications under high-ambient temperatures, except for larger condensing units. Tendency of the commercial refrigeration industry is on to R744 refrigerant, for its unique thermophysical, favourable environmental and safety properties and also efficiency levels for certain ambient conditions. Also it is inexpensive refrigerants compared with HFC. Besides its advantages, there are several disadvantages coming along with using R744 refrigerant, especially high operating pressures and low efficiency levels at high ambient temperatures [1-3]. There are two types of systems with the use of R744, subcritical and transcritical. Subcritical system applications are very broad, since their characteristics allow operation of high, medium and low evaporation. The R744 systems operate subcritically when the condensing temperature is below 31 °C. Transcritical system applications are concerned with high and medium evaporation temperature in a direct compression system. The R744 systems operate transcritically when the gas cooler exit temperature is above 31 °C and the evaporating temperature is below 87.8 °C. Wide range of the R744's operation allows researchers to implement the most efficient configuration for different types of requirements.

Recent works show that even in hot climates transcritical solutions with ejector is more efficient with respect to standard refrigeration cycle for refrigeration, air-conditioning and hot water supply applications. As known, an ejector used as an expansion device in R744 transcritical systems, has improved the COP [4, 5]. Several theoretical and experimental investigations have been performed for different configurations using ejector [6-9]. Design of the ejector is revised to work at refrigeration systems with R744. There are also proposed too many solutions for integrating the ejector into the system. They are studied extensively in literature to determine how addition of this device affects the COP of the refrigeration systems [10, 11]. In most cases, ejector is not enough to make the system efficient and needs some addition improvements such as internal heat exchanger, flooded evaporation, mechanical sub-cooling, and parallel compression [12-15]. Studies have shown that the COP of system increase up to 7% using these improvements. But, it has been noted that the performance of refrigeration systems using ejector are very dependent on the operating conditions [16, 17].

The aim of this study is to investigate and evaluate the effect of climatic conditions of Turkey's cities on the performance of two different refrigeration system alternatives, namely a cascade with R790/R744 and a R744 transcritical system with ejector. Both alternatives have been examined theoretically using thermodynamic models developed by engineering equation solver (EES) software. The proposed models for two systems with respect to various climate conditions of Turkey could be used to develop data for the future experimental refrigeration applications.

System description

In this study, two system configurations are selected for theoretical performance comparison, namely a R744 transcritical gas ejector system and a R290/R744 cascade system. Both systems are examined for the same ambient and evaporation conditions. Ambient conditions are considered in terms of maximum summer dry bulb temperatures that are selected based on the weather data of different cities in Turkey [18]. Evaporation temperatures are chosen to be –10 °C as medium temperature (MT) level and –32 °C as low temperature (LT) level. The temperature difference between conditioned space and ambient air is assumed to be 10 K [14].

Refrigeration capacities of MT and LT are selected to be 50 kW and 25 kW, respectively. These values are utilized for the entire performance analysis of both systems.

System configurations

The configurations of studied systems R290/R744 cascade system and R744 transcritical gas ejector system have been presented in this section.

As the first option, the schematic diagram of R290/R744 cascade system is shown in fig. 1. The cascade system consists of two cycles, namely LTC and HTC, using different refrigerants. Two cycles are connected to each other through a cascade heat exchanger. R744 in the LTC and R290 in the HTC have been utilized as the refrigerant pairs of the system. While R744 cycle controls the temperature of the refrigerated space and the LT evaporator pressure, R290 cycle controls the condenser pressure and condensation temperature of the R744. As shown in fig. 1, there are four temperature levels such as T_{E}, T_{C}, $T_{\mathrm{CAS,E,}}$ and $T_{\mathrm{CAS,C}}$ in the cascade systems. In the present cascade system, MT level evaporator is fed by a CO_2 pump.

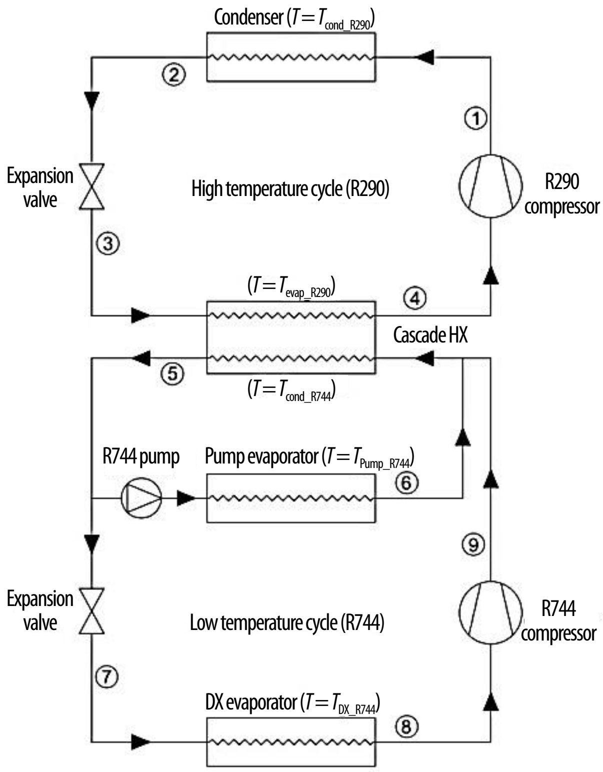

Figure 1. The R290/R744 cascade system

The second alternative is the R744 transcritical gas ejector system whose schematic diagram with its P-h diagram is shown in figs. 2(a) and 2(b). The system consists of three compressors, a gas cooler condenser, two evaporators (MT and LT evaporators), 2 – separator tanks (MT and LT separator tank), 3 – expansion valves and an ejector. The working principle of the cycle can be described briefly as: the vapor leaving the compressor (state 2) enters the gas cooler where it rejects heat to the environment. The refrigerant leaves the gas cooler in vapor phase (state 3), then it enters the ejector (motive flow). The fluid leaving the ejector (state 7) enters the HP separator tank – 1. The saturated liquid leaving from HP separator tank-1 (state 8) is split into two streams which are directed to both LT and MT evaporators. In order to obtain two different cooling level, one stream is throttled to the MT evaporator pressure through expansion valve 1 (state 9) while the other is throttled to the LT evaporator pressure through expansion valve 2 (state 11). The vapor from LT evaporator (state 12) is compressed to MP separator tank-2 (state 13). At the same time, the vapor from MT evaporator (state 10) enters the MP separator tank-2. Then two vapor streams are mixed there. From MP separator-2, the vapor phase of the refrigerant is divided into two streams. One stream flows (state 14) to the ejector to be used as suction flow. The second stream is pressurized by a compressor. Thus, the entire cycle is completed.

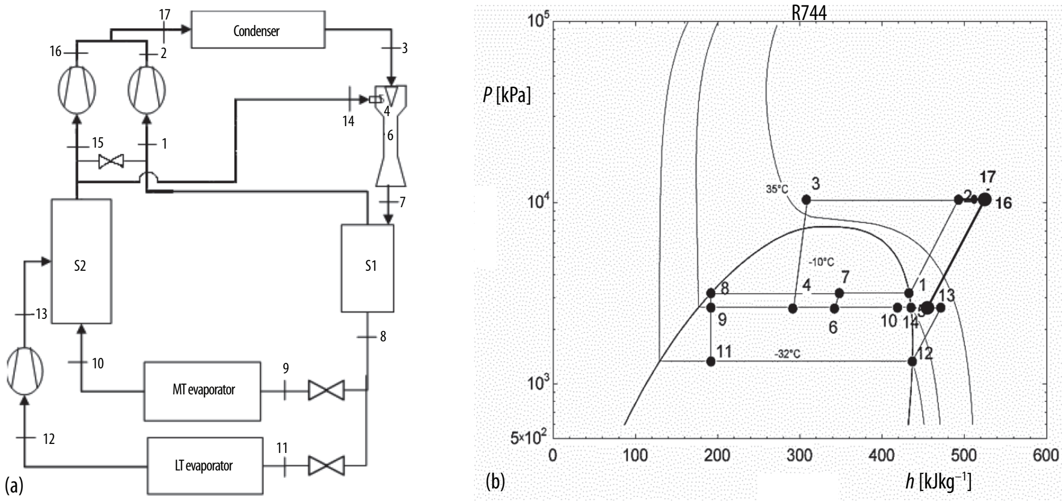

Figure 2. (a) The R744 transcritical gas ejector system, (b) P-h diagram of the second alternative

Mathematical models of both systems

The mathematical models of two studied systems are developed using EES software. For each system component, mathematical relations representing compression, expansion, evaporation, condensation and heat exchanging processes are derived based on the conservation laws of thermodynamics.

In the analysis, the R290/R744 cascade system is modeled according to following assumptions:

– The heat transfer with the surrounding is neglected.
– Expansion processes at expansion valves are isenthalpic.
– Pressure drops in system pipes and heat exchangers are neglected.
– Potential and kinetic energy changes are neglected.
– The effectiveness of cascade heat exchanger is assumed to be 0.8.
– The power of the pump used in the R744 cycle is 3 kW.
– The isentropic compressor efficiency of R290 cycle is calculated using eq. (1) and the isentropic compressor efficiency of R744 cycle is calculated by eq. (2) [2].

$$\eta_{\text{comp\_R290}} = -0.0069\left(\frac{P_{HP}}{P_{MP}}\right)^2 + 0.0930\left(\frac{P_{HP}}{P_{MP}}\right) + 0.4056 \tag{1}$$

$$\eta_{\text{comp\_R744}} = +0.0111\left(\frac{P_{MP}}{P_{LP}}\right)^2 - 0.0793\left(\frac{P_{MP}}{P_{LP}}\right) + 0.8030 \tag{2}$$

The developed mass and energy conservation equations of the R290/R744 cascade system components are given in tab. 1. The corresponding points in equations are shown in fig. 1.

The COP is defined as the ratio of cooling energy output [W] to electrical energy input [W]. On the other hand, the energy efficiency ratio (EER) is defined as the ratio of cooling energy output [BTU] to electrical energy [Wh]. In applications, an air conditioner has an energy efficiency rating that lists how many BTU's per hour are used for each Watt of power it draws.

Table 1. The mathematical model equations for R290/R744 cascade system

	Component	Mass balance	Energy balance
R290 (HTC)	Compressor power of HTC	$\dot{m}_1 = \dot{m}_2 = \dot{m}_{\text{HTC}}$	$\dot{W}_{\text{HTC}} = \dot{m}_{\text{HTC}}\ (h_1 - h_4)$
	Condenser capacity	$\dot{m}_3 = \dot{m}_4 = \dot{m}_{\text{HTC}}$	$\dot{Q}_{\text{cond}} = \dot{m}_{\text{HTC}}\ (h_2 - h_1)$
	Expansion valve HTC		$h_2 = h_3$
	Cascade's evaporation capacity		$\dot{Q}_{\text{CAS,evap}} = \dot{m}_{\text{HTC}}\ (h_4 - h_3)$
R744 (LTC)	Cascade's condensation capacity	$\dot{m}_5 = \dot{m}_6 + \dot{m}_7$	$\dot{Q}_{\text{CAS,cond}} = \dot{Q}_{\text{CAS,evap}}$
	Evaporator capacity	$\dot{m}_7 = \dot{m}_8 = \dot{m}_9$	$\dot{Q}_{\text{evap}} = \dot{m}_8 (h_8 - h_7)$
	Pump evaporator capacity		$\dot{Q}_{\text{pump}} = \dot{m}_6 (h_6 - h_5) / RC$
	Compressor power of LTC		$\dot{W}_{\text{LTC}} = \dot{m}_8 (h_9 - h_8)$
	Expansion valve of LTC		$h_5 = h_7$
	Condenser capacity of LTC		$\dot{Q}_{\text{CAS,cond}} = \left(1 - \dfrac{1}{RC}\right)\dot{m}_6 (h_6 - h_5) + \dot{m}_9 (h_9 - h_5)$
	COP	$COP = \dfrac{\dot{Q}_{\text{pump\_LTC}} + \dot{Q}_{\text{LTC}}}{\dot{W}_{\text{comp\_HTC}} + \dot{W}_{\text{comp\_LTC}} + \dot{W}_{\text{pump\_LTC}}}$	
	EER	$EER = 3.412 \dfrac{\dot{Q}_{\text{pump\_LTC}} + \dot{Q}_{\text{LTC}}}{\dot{W}_{\text{comp\_HTC}} + \dot{W}_{\text{comp\_LTC}} + \dot{W}_{\text{pump\_LTC}}}$	

Therefore, EER values indicating the performance of a refrigeration system are preferred to present the results in this study. The relations between COP and EER are given:

$$EER = 3.412\, x\, COP \tag{3}$$

The mathematical model of the R744 transcritical gas ejector system has been developed based on the following assumptions:

– The pressure drops in gas cooler and evaporators and connecting pipes are neglected.
– System components are adiabatic,
– The flow inside the ejector is a one-dimensional, homogeneous equilibrium flow,
– The expansion efficiencies of the motive stream and suction stream are as assumed to be 0.80. The diffuser of the ejector has also the same efficiency of 0.80 [19].
– Suction pressure ratio ($P_{\text{Diff,out}} / P_{\text{evap,out}}$) is chosen to be 1.2.
– The velocities of the fluid entering the leaving the ejector are neglected
– The efficiencies of compressors are calculated by means of correlation proposed by Robinson and Groll, given in eq. (4) [19].

$$\eta_{\text{comp}} = 0.815 + 0.022\left(\frac{P_{\text{c}}}{P_{\text{dc}}}\right) - 0.0041\left(\frac{P_{\text{c}}}{P_{\text{dc}}}\right)^2 + 0.0001\left(\frac{P_{\text{c}}}{P_{\text{dc}}}\right)^3 \tag{4}$$

The developed mass and energy conservation equations of the R744 transcritical gas ejector system are summarized in tab. 2.

Table 2. The mathematical model equations for R744 transcritical gas ejector system

Component	Mass balance	Energy balance
Power input of MT compressor	$\dot{m}_1 = \dot{m}_2$	$\dot{W}_{\text{MT\_comp}} = \dot{m}_1(h_2 - h_1)$
Gas cooler capacity	$\dot{m}_2 = \dot{m}_3 = \dot{m}_{\text{motive}}$	$\dot{Q}_{\text{Gascooler}} = \dot{m}_2(h_2 - h_3)$
Ejector expansion with isentropic efficiency		$h_4 = h_2 - \eta_{\text{isen\_exp}}(h_2 - h_{4s})$ $u_4 = \sqrt{2(h_3 - h_4)}$
Adiabatic mixing in ejector	$\dot{m}_3 = \dot{m}_4$	
Diffusion of R744 in the ejector with $\eta_{\text{isen\_diff}}$	$\dot{m}_6 = \dot{m}_4 + \dot{m}_5$ $\dot{m}_7 = \dot{m}_8 + \dot{m}_1$	$h_4 = \dfrac{\dot{m}_{\text{motive}}\left(h_4 + \frac{1}{2}u_4^2\right) + \dot{m}_{\text{suction}}\left(h_5 + \frac{1}{2}u_4^2\right)}{(\dot{m}_{\text{motive}} + \dot{m}_{\text{suction}})}$
Throttling in the expansion valve at MT level	$\dot{m}_{14} = \dot{m}_{10} + \dot{m}_{13} = \dot{m}_{\text{suction}}$	$h_8 = h_9$
MT evaporator capacity		$\dot{Q}_{\text{MT\_evap}} = \dot{m}_9(h_{10} - h_9)$
Throttling in the expansion valve at LT level		$h_8 = h_{11}$
LT evaporator capacity		$\dot{Q}_{\text{LT\_evap}} = \dot{m}_{11}(h_{12} - h_{11})$
Power input of LT compressor		$\dot{W}_{\text{LT\_comp}} = \dot{m}_{12}(h_{13} - h_{12})$
Suction line expansion inside the ejector	$\dot{m}_5 = \dot{m}_{14}$	$h_5 = h_{14} - \eta_{\text{isen\_exp}}(h_{14} - h_{5s})$ $u_5 = \sqrt{2(h_{14} - h_5)}$
Power input of parallel compressor	$\dot{m}_{15} = \dot{m}_{16}$	$\dot{W}_{\text{comp\_1}} = \dot{m}_{15}(h_{16} - h_{15})$
System's COP	$COP = \dfrac{\dot{Q}_{\text{MT\_evap}} + \dot{Q}_{\text{LT\_evap}}}{\dot{W}_{\text{LT\_comp}} + \dot{W}_{\text{comp\_1}} + \dot{W}_{\text{MT\_comp}}}$	
System's EER	$EER = 3.412\dfrac{\dot{Q}_{\text{MT\_evap}} + \dot{Q}_{\text{LT\_evap}}}{\dot{W}_{\text{LT\_comp}} + \dot{W}_{\text{comp\_1}} + \dot{W}_{\text{MT\_comp}}}$	

Results and discussion

The theoretical modeling results are presented in terms of the effect of both the ambient conditions varying in different cities and the evaporation temperature conditions. The performance of two studied systems has been compared in terms of EER values.

The effect of evaporation temperature on EER of the systems

The proposed systems contain two evaporation temperature levels. One level corresponds to relatively higher temperature at –10 °C, namely MT evaporator, and the lower temperature level, LT evaporator, corresponds to –32 °C. In cascade option which operates under subcritical conditions, these levels are maintained by addition of a pumped circulation loop in which R744 is circulated in the mixture region at –10 °C. Recirculation ratio (RC) controls the capacity of MT evaporator. On the other hand, in the second alternative, the ejector coupled tanscritical R744 system; these operating temperatures are obtained by means of splitting the liquid part working fluid in two branches and expanding to two different pressure levels. The capacities of 50 kW and 25 kW for MT and LT levels, respectively, are chosen to be the same in both systems. The effect of the evaporation temperature has been investigated only for the lower temperature level in the range of –40 °C to –20 °C. As seen in fig. 3, the performance of both systems in terms of EER values increases with increasing the evaporation temperature. It is observed that the increase in the performance is larger at higher evaporation temperatures for trancritical system.

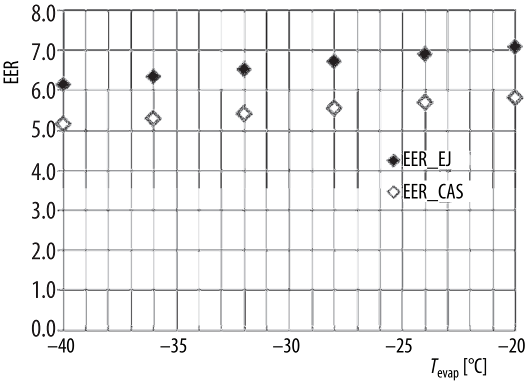

Figure 3. Effect of T_{evap} on EER values of different systems

The effect of ambient temperatures has been also studied for both alternatives. The comparison of performances is plotted in fig. 4. The EER values are computed for the temperature range varies between 25 °C to 45 °C. In this analysis, the evaporator temperatures are assumed to be –10 °C and –32 °C for MT and LT evaporators. It is seen that the transcritical system results in better performances at lower ambient conditions. However, by increasing the ambient temperature the cascade system performance gets closer even shows slightly higher values at temperature levels higher than 35 °C. Thus, it can be concluded that at higher ambient conditions cascade system may be more suitable in terms of performance.

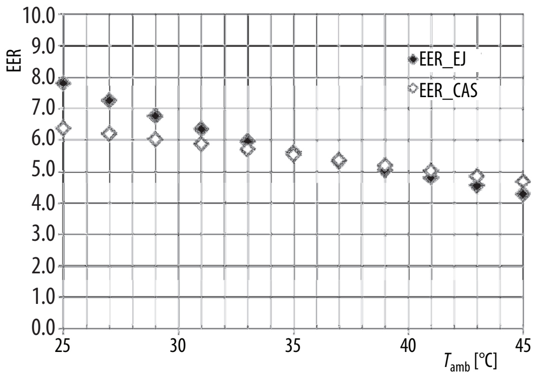

Figure 4. Effect of T_{amb} on EER values of different systems

Comparison of the system performances at different cities of Turkey

A comparative performance analysis between two different refrigeration solutions has been performed. Evaluation is based on the summer ambient air temperature data for all cities of Turkey [18]. Therefore, measured dry bulb (DB) temperatures are selected as the design conditions for cooling. The outdoor DB temperatures of 15 cities represent all cities of Turkey. The performance comparison of two systems for 15 cities of Turkey is given in tab. 3. The differences between the alternative system solutions of the cities are also calculated. It is obvious that the positive difference implies an improvement in performance. In the present study, it is assumed that the DB temperatures of the hot climate regions range from 36 °C to 43 °C and DB temperatures of the mild and cold regions change from 28 °C to 35 °C. It is found that EER of transcrtical system are lower than those of the cascade system between 1.19% and 6.90% in

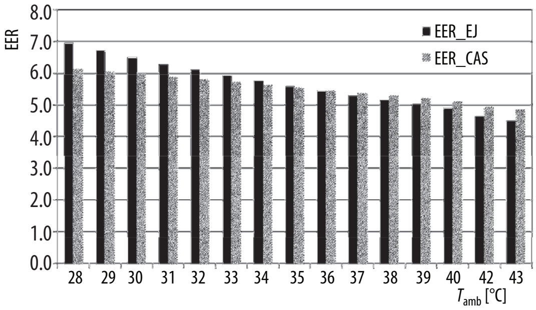

Figure 5. The EER values of two systems based on DB temperatures in cities of Turkey

hot climate regions of Turkey. On the other hand, for cold and mild climate regions of Turkey, the performance difference between the transcritical and cascade systems are observed to be 13.61% and 1.11%.

The comparison of the performance of two systems for all cities in Turkey are shown in a bar plot in fig. 5. It is revealed that the EER values of transcritical system are better than the cascade system for the cities having DB temperatures between 28 °C and 35 °C. On the other hand, it is found that the EER values of cascade system are higher than transcritical system for the cities having DB temperatures between 37 °C and 43 °C. It is also observed that the transition between two system options takes place at 36 °C.

Table 3. The comparison of EER values of two system alternatives for cities of Turkey

City	Max. summer DB temperature [°C]	EER of R290/R744 cascade system	EER of R744 transcritical system	Difference [%]
Nevsehir	28	6.12	6.95	13.61
Giresun	29	6.04	6.72	11.23
Sinop, Ardahan	30	5.95	6.50	9.27
Trabzon	31	5.87	6.30	7.28
Samsun	32	5.78	6.10	5.61
Istanbul	33	5.70	5.92	3.93
Canakkale	34	5.61	5.75	2.57
Ankara	35	5.53	5.59	1.11
Burdur	36	5.44	5.44	-0.02
Izmir	37	5.36	5.30	-1.19
Adana	38	5.27	5.16	-2.15
Antalya	39	5.19	5.03	-3.12
Manisa	40	5.10	4.90	-3.88
Diyarbakir	42	4.93	4.65	-5.73
Sanliurfa	43	4.85	4.52	-6.90

In figs. 6 and 7 the EER of the studied systems are displayed for different climate regions on Turkey's map. The lowest DB temperatures are measured in Giresun, Sinop and Ardahan cities which are located central and eastern Black Sea region of Turkey whereas the highest DB temperatures are measured in Diyarbakır, Sanliurfa and Batman cities which are located south-eastern side of Turkey [18]. In addition, the transcritical ejector system has the highest EER (6.50) for relatively cold regions such as central and eastern Black Sea cities. On the other hand, the cascade system has the lower EER (5.95) in the same regions of Turkey. In addition, it is also revealed that the cascade system's EER (4.85) is better than the transcritical system's EER (4.52) in south-eastern cities of Turkey.

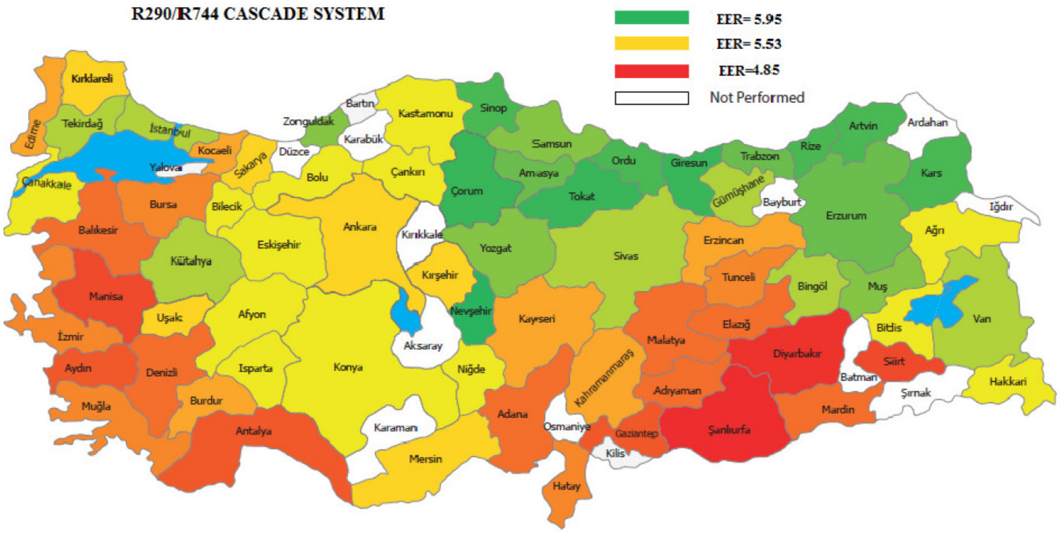

Figure 6. The EER values of R290/R744 cascade system on Turkey map

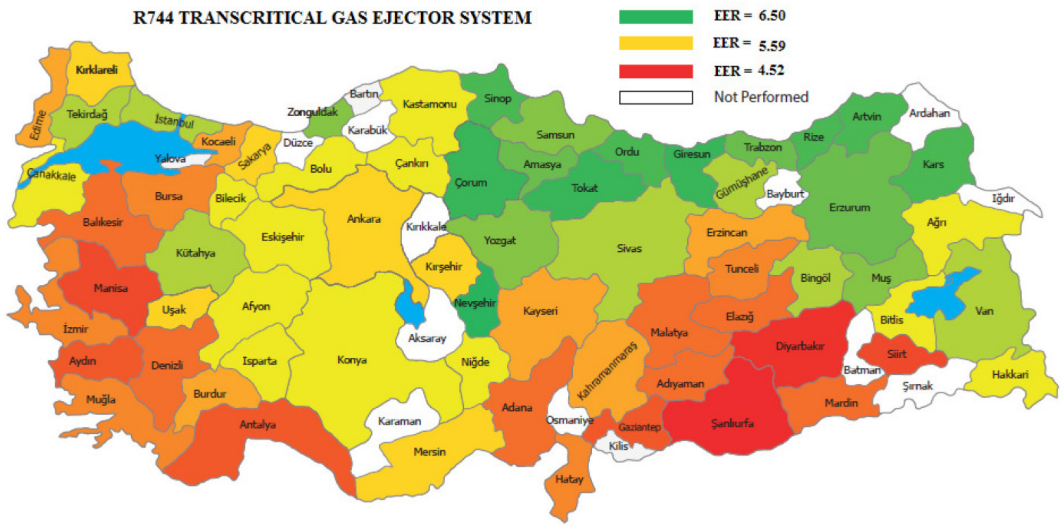

Figure 7. The EER values of R744 transcritical ejector system on Turkey map

As a result, it may be proposed the construction of the cascade systems in mild and hot climate regions such as Aegean, Mediterranean, East Anatolian, and South-East Anatolian of Turkey. Additionally, transcritical ejector system may be suggested to mild and cold climate regions such as Marmara, central Anatolian and eastern Black sea regions. Particularly, transcritical system is suitable option with respect to performance for İstanbul which is located in Marmara region.

Conclusion

The performance of a cascade refrigeration system using R290/R744 refrigerants at high and low temperature cycles, respectively, and a transcritical R744 system with an ejector device are examined by regarding the different ambient conditions of the cities in Turkey. Two temperature levels are determined as –10 °C for medium and –32 °C for low temperature in both systems. In cascade system, the evaporator at medium temperature stage is assumed to be operated with a pump. In both systems, refrigeration capacities of 50 kW and 25 kW are assumed for medium and low temperature levels, respectively.

It is concluded that the transcritical system shows better performance at lower ambient conditions. Performances of both system alternatives decrease with increasing the ambient temperature. However, the performance decreases more for transcritical system compared to cascade option. The decrease in performances between the highest and the lowest ambient conditions reaches about 35% and 21% for the transcritical and cascade systems, respectively. It is also found that the performance of the transcritical system is better at regions having lower ambient conditions near the Black Sea side and east of Turkey. On the other hand, the performances decrease at the south and the Agean Sea regions of Turkey. In mild and cold regions of Turkey, the transcritical ejector option is a better alternative due to having higher performance compared to the cascade system.

In order to make transcritical system model have better efficiencies at high temperature regions of Turkey, different configurations like suction line heat exchanger, liquid ejector or multi-ejector systems may be added for further investigations. Yearly analysis should be also performed for both systems to establish better comparison for Turkey.

Nomenclature

h – specific enthalpy, [kJ/kg]
$\dot{m}$ – mass flow rate, [kg/s]
P – pressure, [kPa or bar]
$\dot{Q}$ – heat transfer rate, [kW]
T – temperature, [°C or K]
$\dot{W}$ – power, [kW]

Greek symbol

η – efficiency, [%]

Subscripts

C – condensation
comp – compressor
cond – condenser
c – charge
dc – discharge
E – evaporation
evap – evaporator

Acronyms

CAS – cascade
DB – dry bulb
EER – energy efficiency ratio
EJ – ejector
HFC – hydro-flora-carbon
HP – high pressure
LP – low pressure
LT – low temperature
MP – medium pressure
MT – medium temperature
R290 – propane
R744 – carbon dioxide
RC – recirculation ratio

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