



COMPARISON OF SUPERHEATING EFFECT OF WATER AS A REFRIGERANT WITH THE OTHER REFRIGERANTS

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Abstract: The performance comparison of water as a refrigerant (R718) with some refrigerants including R717, R290, R134a, and R152a is presented for a vapor compression refrigeration cycle. A computer program simulating the vapor compression refrigeration cycle including superheating was developed to calculate the coefficient of performances (COPs) for the refrigerants. Evaporator temperatures which water yields a better COP over the other refrigerants are investigated for different superheating cases. The effects of degree of superheating on the COPs are observed. R718 shows the steepest increase in the relative COP gain with increasing evaporator temperature. At constant evaporator temperatures, the absolute COP of R718 increases as the degree of superheating increases. It is found that for evaporator temperatures above 26°C and high degree of superheating (6K-8K), R718 gives the highest relative COP increase at constant condenser temperature and polytropic efficiency.

Keywords: Soğutma, Soğutucu akışkanlar, Su, Kıyaslama, Aşırı ısıtma, Soğutma tesir katsayısı.

SU'YUN SOĞUTUCU AKIŞKAN OLARAK AŞIRI ISITMA ETKİSİNİN DİĞER AKIŞKANLARLA KIYASLANMASI

Özet: Su (R718) soğutucu akışkan olarak, R717, R290, R134a ve R152a gibi soğutucu akışkanlarla performans açısından buhar sıkıştırımlı bir soğutma sisteminde kıyaslanmıştır. Aşırı ısıtmanın olduğu buhar sıkıştırımlı soğutma çevriminin simülasyonunu yapan bir bilgisayar programı geliştirilerek kullanılan soğutucu akışkanlar için çevrimin soğutma tesir katsayısı (STK) hesaplanmıştır. Farklı aşırı ısıtma durumları için, su'nun diğer soğutucu akışkanlara göre daha yüksek STK değerlerini verdiği buharlaştırma sıcaklıkları araştırılmıştır. Aşırı ısıtmanın STK üzerindeki etkileri araştırılmıştır. Artan buharlaştırıcı sıcaklıklarında, R718'in bağıl STK değerinde keskin bir artma görülmüştür. Sabit buharlaştırıcı sıcaklıklarında, aşırı ısıtma derecesi arttığında R718'in mutlak STK değeri artmaktadır. Yoğuşturucu sıcaklığı ve politropik verim sabit tutulduğunda, 26°C'nin üzerindeki buharlaştırıcı sıcaklıklarında yüksek aşırı ısıtma değerlerinde (6K-8K), R718'in en yüksek bağıl STK artışını verdiği tespit edilmiştir.

Anahtar Kelimeler: Soğutma, Soğutucu akışkanlar, Su, Kıyaslama, Aşırı ısıtma, Soğutma tesir katsayısı.

NOMENCLATURE

c_p	Constant pressure specific heat of [kJ kg ⁻¹ K ⁻¹]
COP	Coefficient of Performance
COP_{abs}	COP of refrigerant absolute to the COP of R134a
COP_{rel}	Ratio of actual COP to the theoretical COP
E	Energy, kJ
g	Gravitational acceleration, m/s ²
h	Specific enthalpy of refrigerant [kJ kg ⁻¹]
k	Ratio of constant specific heats
\dot{m}	Mass flow rate, kg/s
η_{is}	Isentropic efficiency of the compressor
η_p	Polytropic efficiency of the compressor
q_r	Refrigeration effect [kJ kg ⁻¹]
\dot{Q}	Heat transfer rate, kW
P	Pressure [kPa]
s	Specific entropy of refrigerant [kJ kg ⁻¹ K ⁻¹]

T	Temperature [°C]
TD_{sc}	Degree of subcooling [K]
TD_{sh}	Degree of superheating [K]
\dot{W}	Power, kW
V	Velocity, m/s
Z	Height from the reference level, m

Subscripts

c	condenser
$comp$	compressor
$c.v$	control volume
e	evaporator
i	inlet
l	liquid
o	outlet
sc	subcooling
sh	superheating

<i>v</i>	vapor
1	compressor inlet
2	compressor outlet
3	condenser outlet
4	expansion valve inlet
5	evaporator inlet
6	evaporator outlet

INTRODUCTION

An actual vapor compression refrigeration cycle differs from the theoretical vapor compression cycle. Three main effects must be taken into consideration in the actual refrigeration cycle: subcooling, superheating, and pressure drops associated with refrigerant flow through the refrigeration cycle. Superheating is defined as a process of increasing the temperature of the refrigerant vapor above its saturation temperature.

Most of the studies, which are related to the actual vapor compression refrigeration cycles in the literature, have been based on the effect of subcooling rather than superheating on the coefficient of performance of the cycle (COP).

Prasad (1983) has presented a model simulating an actual compression refrigeration cycle with subcooling and superheating in order to find optimum interstage pressure in a two stage multistage refrigeration system. It has been found that the optimum interstage pressure increases with an increasing degree of superheating, while it decreases with the degree of subcooling. Thermodynamic models consisting of temperature and property dependent models have been developed to perform the design and performance evaluation in an integrated mechanical subcooling vapor compression refrigeration system (Khan and Zubair, 2000). A second law of thermodynamics based analysis has been presented for a theoretical and actual vapor compression refrigeration cycle with a mechanical subcooling loop (Zubair, 1994). There have been similar studies that are based on the effect of the subcooling on the performance of vapor compression refrigeration cycles (Zubair, 1990; Thornton *et al.*, 1994; Zubair and Khan, 1995; Zubair *et al.*, 1996; Khan and Zubair, 2000).

Because of having excellent features, being an environmentally safe (ODP=0 and GWP=0), non-toxic, non-flammable, non-explosive, easily available, and the most inexpensive refrigerant, water (R718) is one of the oldest natural refrigerants being used for refrigeration applications except freezing. It also has the highest COP over other refrigerants at higher evaporator temperatures. Despite these advantages, water as a refrigerant has also a few disadvantages such as high compression ratio and high compressor exit temperature. By using specially designed compressors in the vapor compression refrigeration cycle, these disadvantages have

been addressed (Elovic and Holmes, 1996; Koren and Ophir, 1996; Wight *et al.*, 2000; Müller 2001).

A few studies have been published in which water (R718) has been compared with other refrigerants in some aspects including COP and refrigeration capacity. In these studies, the performance comparison of water (R718) has been based on a theoretical vapor compression refrigeration cycle. Orshoven *et al.* (1993) have compared water as a refrigerant with other refrigerants (R12, R22, R502, and R717) in the respect to COP by using the computer codes. In their study, a theoretical vapor compression refrigeration model, with no pressure drop, no subcooling, and no superheating, has been used. By means of a dynamic computer model, traditional ammonia cooling plants were compared to the cooling plants in which water is used as a refrigerant (Madsboll and Elefsen, 1993). It has been observed that as much as 50% of energy was saved when the cooling plants were operated with water as a refrigerant. Chen *et al.* (1997) have examined the thermal storage part of an air-conditioning system in which water is used as a refrigerant by means of a numerical simulation model. They also have compared the COPs of R718 (water) and R22 under different operating conditions. Albring and Heinrich (1998) have compared the COP of water as a refrigerant (R718) with that of R290, R134a, R22, R152a, and R717 in a theoretical and an actual compression refrigeration cycle. In an actual cycle, for an evaporator temperature of 6°C, a condenser temperature of 30°C, and 5K subcooling case, it was observed that there was no benefit to use water as a refrigerant. It has been found that the COP of the water coolers was significantly increased by decreasing temperature difference between the external water flow and refrigerant water.

From the literature survey, it is clear that the studies that are related to the performance analysis of water as a refrigerant have been generally based on a theoretical vapor compression refrigeration cycle, but there have been also some studies in which the results for the subcooling cases of an actual vapor compression refrigeration cycle have been presented, but the studies based on investigating the effect of superheating on a vapor compression cycle is very scarce. However, the present study is different from the studies published by presenting a detailed analysis of an actual vapor compression refrigeration cycle with water as working fluid. Superheating is analyzed in detail and their effects on the system performance are discussed extensively. The main objective of this study is to compare the coefficient of performance (COP) of water (R718) with the COPs of the refrigerants R717, R134a, R152a, and R290 in an actual vapor compression refrigeration cycle for different superheating, polytropic efficiency, evaporator temperature cases by using the computer program developed and a commonly available refrigerant library.

THEORETICAL ANALYSIS

The theoretical analysis is based on the thermodynamic analysis of a vapor compression refrigeration system involving superheating. This refrigeration system consisting of compressor, condenser, thermostatic expansion valve and evaporator is shown in Figure 1.

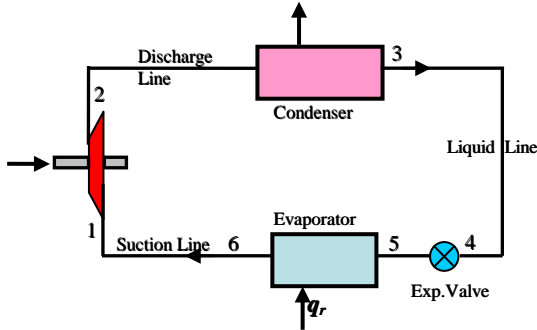


Figure 1. Representation of a vapor compression refrigeration system with superheating

In this vapor compression refrigeration cycle, the refrigerant enters the compressor at state 1 at a low pressure, low temperature, and a superheated vapor state. From state 1 to 2, the refrigerant is compressed by the compressor and is discharged at state 2 at a high pressure, high temperature and a superheated vapor state. At state 2, it enters the condenser where it rejects heat to the environment. It leaves the condenser at state 3 at a high pressure and a saturated liquid state. states 3 and 4 coincide because there is no subcooling. At state 4, the refrigerant enters the expansion valve where its pressure is reduced in a throttling process from high pressure (condenser pressure) to low pressure (evaporator pressure). After this it has state 5 and enters the evaporator where it absorbs heat from the refrigerated space. At state 6, it leaves the evaporator at a low pressure, low temperature, and a saturated vapor state.

Superheating is defined as the temperature difference between compressor inlet temperature and evaporator saturation temperature. In practice, some degree of superheating is necessary to make sure that a refrigerant entering the compressor is free from the liquid droplets to prevent damage to the compressor. By comparing the actual cycle including superheating to the theoretical refrigeration cycle, the desuperheating loss in the discharge line is increased, and an exergy loss is introduced (IIR, 1992).

In an actual refrigeration cycle, superheating a refrigerant can generally occur either in the last section of the evaporator or in the suction line. In a vapor compression refrigeration cycle, superheating occurs spontaneously because of absorbing heat from a refrigerated space or the section of a refrigeration system located outside the refrigerated space between evaporator exit and compressor inlet. When the suction line is located outside the refrigerated space, for every 2.5°C degree of superheating, the loss in the refrigeration capacity is about 1% (Dincer I, 2003). It is assumed that superheating occurs in the suction line and the suction line is placed inside the refrigerated space. It is also assumed that there is no subcooling and pressure drop in the refrigeration cycle. states 3 and 4 coincide due to no subcooling. Figure 2 shows pressure-enthalpy (P-h) and temperature-entropy (T-s) diagrams of an actual refrigeration cycle with superheating.

The each element in the vapor compression refrigeration cycle is treated as a control volume. First law of thermodynamics for a control volume can be expressed as (Van Wylen and Sonntag, 1985),

$$\dot{Q}_{c.v} + \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gZ_i \right) = \frac{dE_{c.v}}{dt} + \sum \dot{m}_o \left(h_o + \frac{V_o^2}{2} + gZ_o \right) + \dot{W}_{c.v} \quad (1)$$

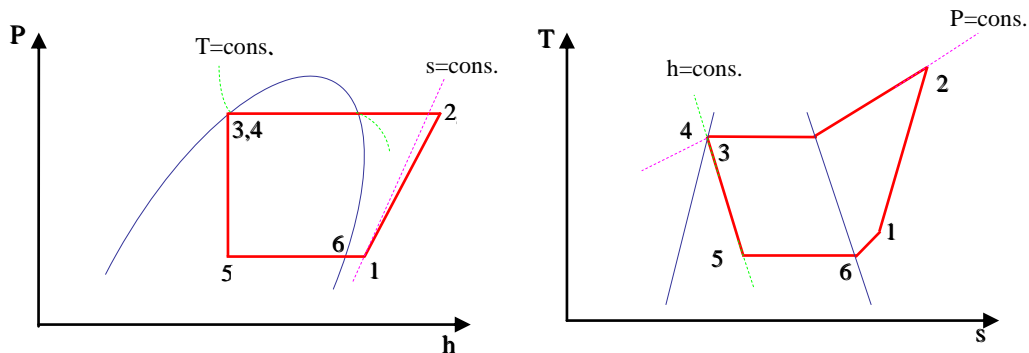


Figure 2. a) P-h diagram of a refrigeration cycle with superheating b) T-s diagram of the refrigeration cycle with superheating

$$\frac{dm_{c,v}}{dt} + \Sigma \dot{m}_o - \Sigma \dot{m}_i = 0 \quad (2)$$

where $\dot{Q}_{c,v}$, $\dot{W}_{c,v}$ and $E_{c,v}$ are the heat transfer rate to the control volume, the actual power done by the control volume and the energy within the boundary of the control volume, respectively while \dot{m} , h , $\frac{V^2}{2}$, gZ are the mass flow rate, specific enthalpy, specific kinetic energy and potential energy of the fluid at the inlet or outlet conditions. It is also presumed that steady-state and uniform flow conditions exist thorough the elements of the vapor-compression refrigeration cycle and changes in kinetic and potential energies and heat loss from the compressor are negligible.

When the above described assumptions are combined with equations (1) and (2), the change of energy and mass flow rate for the compressor of the compression refrigeration system,

$$\frac{dE_{c,v}}{dt} = 0 \quad (3)$$

$$\Sigma \dot{m}_o = \Sigma \dot{m}_i = \dot{m}$$

and the power required for the cycle, \dot{W} , can be expressed for superheating case

$$\dot{W}_{comp,sh} = \dot{m}(h_2 - h_1) \quad (4)$$

where h_1 and h_2 are the specific enthalpies at the inlet and outlet of the compressor, respectively. Specific work of compression w_{comp} for the compressor can be written

$$w_{comp,sh} = h_2 - h_1 \quad (5)$$

In the temperature range studied (0,01 - 50°C), the approximate values of compressibility factors of the refrigerants such as R718, R717, R134a, R152a and R290 are 0.98, 0.92, 0.85, 0.88 and 0.82, respectively. Therefore, the refrigerants under the study can be simulated as ideal gases during compression process. Hence the specific work of compression can also be expressed by

$$w_{comp} = \frac{c_p T_{comp,i}}{\eta_{is}} \left[\left(\frac{P_c}{P_e} \right)^{\left(\frac{k-1}{k} \right)} - 1 \right] \quad (6)$$

where P_c , P_e , $T_{comp,i}$ are the condenser pressure, evaporator pressure, and the temperature at compressor

inlet, respectively, while η_{is} is the isentropic efficiency of the compressor, c_p , and k are constant pressure specific heat, and specific heat ratio of the refrigerant, respectively. Isentropic efficiency of the compressor can be expressed in terms of polytropic efficiency η_p , pressure ratio and specific heat ratio.

$$\eta_{is} = \frac{\left(\frac{P_c}{P_e} \right)^{\frac{k-1}{k}}}{\left[\left(\frac{P_c}{P_e} \right)^{\left(\frac{k-1}{k\eta_p} \right)} - 1 \right]} \quad (7)$$

During the throttling process in the expansion valve, it is assumed that there is no heat transfer to the environment, which results in

$$h_4 = h_5 \quad (8)$$

where h_4 and h_5 are the enthalpies of refrigerant at the expansion valve inlet and evaporator inlet, respectively. The refrigeration effect of the vapor compression refrigeration cycle can be expressed as

$$q_{r,sh} = (h_1 - h_5) \quad (9)$$

The specific enthalpy of superheated vapor at state 1 can be expressed by

$$h_1 = h_6 + c_{p,v}(T_1 - T_e) \quad (10)$$

where h_6 is the enthalpy of saturated vapor at the evaporator outlet and $c_{p,v}$ is the constant specific heat of refrigerant in vapor state between states 6 and 1, T_e is the evaporator saturation temperature, and T_1 is the temperature at the compressor inlet.

The coefficient of performance of the refrigeration cycle COP_{sh} can be expressed by

$$COP_{sh} = \frac{q_{r,sh}}{w_{comp,sh}} \quad (11)$$

The performance of an actual vapor compression refrigeration cycle with superheating can be compared with that of a theoretical vapor compression refrigeration cycle by means of the following equation:

$$COP_{rel,sh} = \left(\frac{w_{comp}}{w_{comp,sh}} \right) \left[1 + \frac{c_{p,v}}{q_r} TD_{sh} \right] \quad (12)$$

where $COP_{rel,sh}$ is the ratio of COP_{sh} to the COP of theoretical refrigeration cycle, q_r is the refrigeration effect for the theoretical case and, TD_{sh} is the degree of superheating.

Based on the above model a computer program was developed calculating COPs for all refrigerants and their absolute differences to those COPs obtained for R134a. The computer code uses a commonly available data bank for refrigerant properties of such as P, T, h, s.

For the investigations, four parameters were varied or held constant. These include evaporator temperature T_e , condenser saturation temperature T_c , superheating TD_{sh} (the temperature difference between the evaporator saturation temperature and compressor inlet), and the polytropic efficiency of the compressor η_p . The refrigerants R718, R717, R290, R134a, and R152a are compared.

RESULTS AND DISCUSSIONS

Through the developed computer program, the evaporator temperature was increased from 0°C to 42°C while the condenser temperature and polytropic efficiency were held constant as 50°C and 0.9, respectively. The COP values of refrigerants in this study are given according to the absolute COP values ($COP_{abs} = COP(\text{of the related refrigerant}) - COP_{R134a}$) of the refrigerants, which were determined as a function of evaporator temperature. R-134a is the long term replacement refrigerant for R12. The determination of the absolute COP values are performed with different superheating values between 1K and 7K with 2K intervals while polytropic efficiency and condenser temperature are held constant as 0.9 and 50°C, respectively.

The effect of superheating of vapor refrigerant on the refrigeration effect changes to the place where the superheating of a refrigerant takes place. If the superheating takes place inside the refrigerated space, an additional amount of refrigeration is provided to the cycle. When it takes place outside the refrigerated space, there is no benefit for the cycle. This is called a useless cooling effect. As it was mentioned before, it has been assumed that superheating of vapor refrigerant occurs inside the refrigerated space.

At constant evaporator and condenser temperatures, the amount of heat for superheating vapor refrigerant is absorbed from the refrigerated space. This heat is added to the refrigeration effect of the cycle. This causes the specific volume of vapor refrigerant to increase. An additional amount of energy for the compressor is necessary to compress this extra specific volume of refrigerant. As a result, the compressor power increases. In the superheating process, both refrigeration effect and compressor power increases. The effect of superheating

vapor refrigerant on the coefficient of performance (COP) of the refrigeration cycle changes according to the ratio of increase of refrigeration effect to that of compressor power. The parameters ($w_{comp} / w_{comp,sh}$) and ($c_{p,v} / q_r$) in equation (8) are used to compare the effect of superheating on the COP of the cycle. When these parameters are analyzed, it is observed that the effect of the parameter ($c_{p,v} / q_r$) on the coefficient of performance of the refrigeration cycle (COP) is negligible.

The parameter ($w_{comp} / w_{comp,sh}$), in other words the ratio of the theoretical compressor work to the actual work with superheating, can be used as a deciding parameter in order to compare the relative performance of the refrigerants. The deciding parameters that are shown in Table.1 are tabulated for the refrigerants under study for four different superheating cases. These results are obtained for the specific conditions of 10°C evaporator temperature, 50°C condenser temperature, and 0.9 polytropic efficiency.

Figure 3 shows the variation of COP_{abs} as a function of evaporator temperature for different superheating values TD_{sh} , constant 0.9 polytropic efficiency and 50°C condenser temperature. As it is shown in Figure 3, the absolute COPs of the refrigerants increase as the evaporator temperature increases except for R290. R718 shows the steepest increase in all superheating cases in Figure 3.

Figure 3 also shows that the evaporator temperatures which R718 gives better COP increases with increasing values of superheating while the evaporator temperature is kept constant. The reason why R718 gives better COP_{abs} with increasing values of superheating is better understood by analyzing Table.1.

The deciding parameter for superheating ($w_{comp} / w_{comp,sh}$), in other words the ratio of theoretical compressor power to the actual power with superheating, is the biggest for R718 for the superheating cases 1K through 7K. It is also clear from Table.1 that the relative amount of deciding parameter of R718 increases with increasing values of superheating when it is compared to the other refrigerants. This indicates that R718 gives better COP relative to the other refrigerants under study for a constant evaporator temperature case. This result is shown in Figure 3.

The evaporator temperatures, which COP_{abs} of R718 is higher than that of the other refrigerants, are 31°C for $TD_{sh}=1K$, 29°C for $TD_{sh}=3K$, 27°C for $TD_{sh}=5K$, and 25°C for $TD_{sh}=7K$. Below these evaporator temperatures, R717 produces a better COP. However, despite that ammonia does not deplete the ozone layer (ODP=0) and does not directly contribute to the greenhouse effect, it still has a sharp rank smell, is toxic, and explosive in certain mixtures with air.

Table 1. The ratio of the theoretical compressor work to the actual work ($w_{comp}/w_{comp,sh}$) with superheating for the conditions of 10°C evaporator temperature, 50°C condenser temperature, and 0.9 polytropic efficiency.

Superheating	R290	R134a	R152a	R717	R718
TD _{sh} =1K	0.9889	0.9897	0.9913	0.9933	0.9965
TD _{sh} =3K	0.9677	0.9702	0.9734	0.9803	0.9895
TD _{sh} =5K	0.9475	0.9514	0.9570	0.9677	0.9826
TD _{sh} =7K	0.9283	0.9334	0.9413	0.9555	0.9759

Water (R718) is free of these serious disadvantages. For certain operating conditions at the lower evaporator temperatures, R718 still has advantageous over some of the refrigerants. For example, above 12°C for TD_{sh}=7K, COP values of R718 are better than R290 and R134a. The evaporator temperatures which R718 gives a better COP over the other refrigerants under study are shown for eight different superheating cases are shown in Table.2.

CONCLUDING REMARKS

The performance of water as a refrigerant (R718) is compared with some refrigerants including R717, R290, R134a, and R152a by using a computer program

developed for calculations of an actual vapor compression refrigeration cycle including superheating. R718 shows the steepest increase in COP with increasing evaporator temperature while keeping the condenser and evaporator temperature constant.

At constant evaporator temperatures, the absolute COPs of R718 increases as the degree of superheating increases. With increasing superheating, it is also concluded that the evaporator temperature range at which R718 performs better COP relative to the other refrigerants increases. For low range of superheating (1K-3K), above 32°C evaporator temperature, R718 has advantageous over the other refrigerants. At this range, above 12°C, it still has advantageous over R290 and

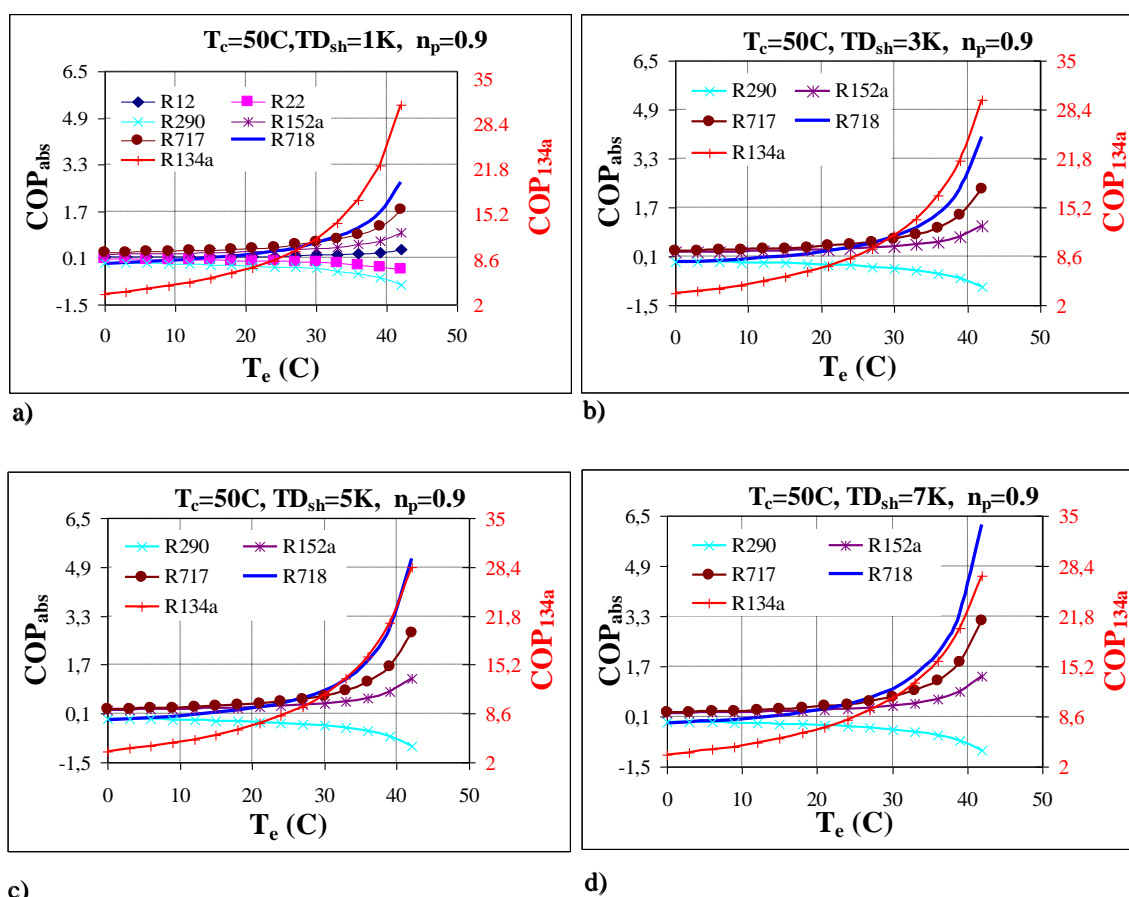


Figure 3. COP_{abs} as a function of evaporator temperature for different TD_{sh} values
a) $TD_{sh}=1K$, b) $TD_{sh}=3K$, c) $TD_{sh}=5K$, d) $TD_{sh}=7K$

Table 2. Evaporator temperatures which R718 gives a better COP than R290, R134a, R152a, and R717 in an actual refrigeration cycle with superheating, $\eta_p=0.9$ and $T_c=50^\circ\text{C}$

Superheating	R290	R134a	R152a	R717
TD _{sh} =1K	9	9	25	32
TD _{sh} =2K	3	9	24	30
TD _{sh} =3K	3	9	23	29
TD _{sh} =4K	3	9	22	28
TD _{sh} =5K	3	9	22	27
TD _{sh} =6K	4	9	21	26
TD _{sh} =7K	4	9	20	25
TD _{sh} =8K	4	9	20	24

R134a. In the case of high superheating (6K-8K), it is advantageous to use water as a refrigerant above 26°C evaporator temperature. In this study, for all superheating cases, R718 shows the steepest increase in the values of absolute COPs when it is compared to the other refrigerants.

The presented results show that the use of water as a refrigerant can result in a higher coefficient of performance (COP) relative to the other refrigerants under study in the case of maximizing superheating, and polytropic efficiency in an actual vapor compression refrigeration cycle.

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