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COPs OF R718 IN COMPARISION WITH OTHER MODERN REFRIGERANTS

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ABSTRACT
Water as a refrigerant (R718) is compared with other modern refrigerants (R717, R290 R134a, R12, R22, and R152a) regarding refrigeration capacity and COP (coefficient of performance). A computer program simulating a theoretical vapor compression refrigeration cycle was developed to calculate COPs, compression ratios, and discharge temperatures of the refrigerants from the compressor. The effects of temperature lift, which is the temperature difference between condenser and evaporator, and polytropic efficiency are also investigated. It is shown that for evaporator temperatures above 20˚C and small temperature lifts (5K), R718 gives the highest COP, assuming exactly the same cycle parameters. For medium temperature lifts (20…25K), this evaporator temperature is above 35˚C, whereas for even greater temperature lifts it decreases again. Furthermore this evaporator temperature at which R718 gives a greater COP than the other refrigerants decreases with increasing values of polytropic efficiency

INTRODUCTION
Water as a refrigerant is one of the oldest refrigerants being used for refrigeration applications down to freezing because of its easy availability and excellent thermodynamics and chemical properties. Beside these advantages, there are technical challenges that result from its high specific volume at low temperatures, necessary high pressure ratios across the compressor, and resulting high compressor outlet temperature. These challenges have been overcome by designing and manufacturing special compressors for water vapor compression applications.

Water vapor compression applications have been classified according to the compressor type used in the refrigeration cycle, which include single and multistage centrifugal, multistage axial, Roots, liquid ring, cycloid, and jet/ejector compressors Wight *et al.* [1]. Most of the studies in the literature have been based on the applications in which centrifugal compressors have been used Madsboll *et al.*[2]; Elovic and Holmes [3]; Madsboll and Minds [4]; Albring [5]; Albring and Heinrich [6]; Koren and Ophir [7]; Albring and Heinrich [8]; Müller [9]. Madsboll *et al.*[2] have presented the applications based on the production of vacuum ice and water chiller where the swept volume is very large (50 m^3/s - 360 m^3/s) in which centrifugal compressors are used to meet the requirements. A centrifugal compressor having very large volume flow rate and high pressure ratios with very thin and light radial blades has been designed and developed to produce ice slurries Elovic and Holmes [3]. Madsboll and Minds [4] have designed a new cooling system in which a centrifugal compressor is used to compress water vapor addressing environmental concerns, especially the Greenhouse effect. They also have described the results of computer simulation comparing water with other refrigerants. Centrifugal compressors having a suction volume 0.5 m^3/s and 5 m^3/s have been used in a refrigeration system that was designed for demonstration and experimental purposes Albring [5]. Centrifugal compressors have also been studied in a two-stage heat pump application and were also classified with respect to the pressure ratio across the compressor Albring and Heinrich [6] Koren and Ophir [7] have given information about the application of water vapor technology for commercial applications such as ice machines and chillers in which centrifugal compressors have been used to compress the water vapor. Design specifications for the centrifugal compressors that are employed in water chiller application have been presented Albring and Heinrich [8] A design algorithm for an advanced centrifugal compressor has also been created by Müller [9].

There have been a few studies about axial compressors and their applications Paul [10-11]. Claiming advantages over centrifugal compressors like smaller size, insensitivity to liquid droplet erosion and the possibility of building several stages on a single shaft, axial compressors have been projected for vapor compression refrigeration applications ranging from 150 kW to 3000 kW Paul [10]. An example study of applying a six-stage axial compressor has been presented by Paul [11].

Roots and liquid ring compressors have not been widely used in water-vapor compressor applications. Some advantages including smaller size, easy construction, low noise level, and reasonable price, etc. have been reported for ring compressors by Hackensellner and Jurisch [12]. Stene [13] has given information about special projects related to Roots and liquid ring compressors.

There have been several studies in the literature including the theory and experimentation involving cycloid compressors and jet ejectors Madsboll *et al*. [2]; Grazzini and Albero [14]; Sheer and Mitchley [15]; Nyvad and Elefsen [16]; Huang [17].

Furthermore, there have been some studies in which water as a refrigerant has been compared with other refrigerants in some aspects including COP, refrigeration capacity, compression ratio, and compressor outlet temperature. By means of a computer program, which has been developed to determine the thermodynamic properties of some working fluids used in vapor compression refrigeration cycle, water has been compared with other refrigerants de' Rossi and Mastrullo [18]. The study also has shown that water and ammonia are the best choices regarding the latent enthalpy. Orshoven *et al*. [19] have compared water as a refrigerant with other refrigerants including R12, R22, R502, and R717 in respect to COP by using several commercial programs. They have used a simple refrigeration cycle model consisting of assumptions of no pressure drop through the cycle, no subcooling between the condenser outlet and expansion valve, and no superheating between evaporator outlet and compressor inlet. Madsboll and Elefsen [20] have compared cooling plants using water as a refrigerant with traditional NH_3 cooling plants by means of a dynamic computer model that they developed. They also mentioned that energy saving in the cooling plant with water as refrigerant is 50% more as it is compared with traditional NH_3 cooling plants. By using a numerical simulation model, Chen [21] has analyzed a thermal storage system of an air-conditioning system with water as a refrigerant and also compared the COPs of R718 (water) and R22 under different evaporator and condenser temperatures.

The objective of the present study is to compare water as a refrigerant (R718) with the refrigerants R717 (ammonia), R12, R22, R134a, R152a, and R290 (propane). The comparison is based on coefficient of performance (COP) obtained by the refrigerant in the refrigeration cycle, and various cycle parameters: specific volume, pressure ratio, and discharge temperature. The effects of temperature lift and polytropic efficiency on the COPs are also investigated. For this study, a computer code was developed to calculate COPs, pressure ratios, compressor outlet temperatures of the refrigerants, and evaporator temperatures above which water as a refrigerant has a higher COP than the other refrigerants. A commonly available refrigerant library was used to calculate the thermodynamic properties of the refrigerants.

THERMODYNAMIC MODEL

The model used to compare water as a refrigerant with R717, R290, R134a, R12, R22, and R152a is based on a theoretical vapor compression refrigeration cycle consisting of compressor, condenser, thermostatic expansion valve, and evaporator. This refrigeration cycle is shown in Fig.1.

In this theoretical vapor compression cycle, the refrigerant enters the compressor at state 1 at low pressure, low temperature, and saturated vapor state. From state 1 to 2, the refrigerant is compressed by the compressor and is discharged at state 2 at high pressure, high temperature, and superheated vapor condition. At state 2, it enters the condenser where it rejects heat to the environment.

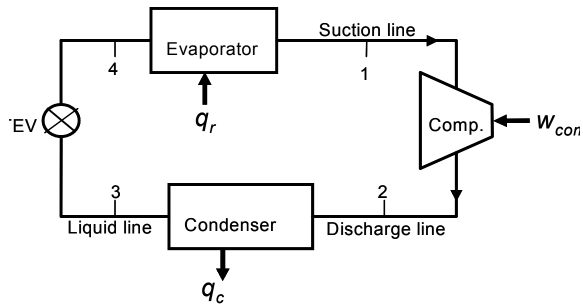


Fig.1. Schematic of one stage vapor compression refrigeration system

It leaves the condenser at state 3 at high pressure and saturated liquid state. From state 3, 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 is at state 4 and enters the evaporator where it absorbs heat from the refrigerated space; and it leaves the evaporator at low pressure, low temperature, and saturated vapor state. In the theoretical cycle, it is also assumed that there is no superheating in the suction line, no subcooling in the liquid line and no pressure drop throughout the cycle.

Figure 2 shows a pressure-enthalpy diagram with the above described states for the theoretic simple vapor compression refrigeration cycle.

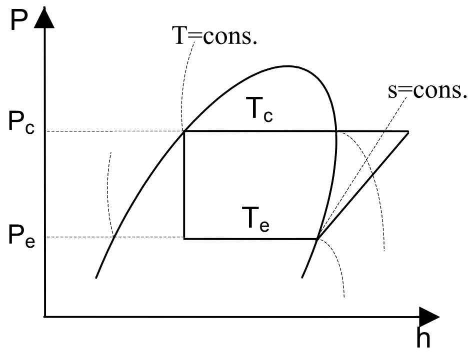


Fig.2. P-h diagram of a simple vapor-compression refrigeration cycle

Also, it is assumed that steady-state and uniform- flow conditions exist in all elements of this simple vapor compression refrigeration cycle and changes in kinetic, potential energies, and heat loss from the compressor are neglected. Therefore, specific work of compression w_{comp} for the compressor can be written as

$$w_{comp} = h_2 - h_1 \tag{1}$$

where h_1 and h_2 are the enthalpies of refrigerant at compressor inlet and exit, respectively. The refrigerants are simulated as ideal gases during compression process. Hence the specific work of compression can also be expressed by

$$w_{comp} = \frac{c_p T_1}{\eta_{is}} \left[\left(\frac{P_c}{P_e} \right)^{\left(\frac{K-1}{K} \right)} - 1 \right] \tag{2}$$

where P_c, P_e, T_1 are 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. Isentropic efficiency of the compressor can be expressed in terms of polytropic efficiency η_p, pressure ratio and specific heat ratio

$$\eta_{is} = \frac{\left(\frac{P_c}{P_e} \right)^{\frac{K-1}{K}}}{\left[\left(\frac{P_c}{P_e} \right)^{\left(\frac{K-1}{K\eta_p} \right)} - 1 \right]} \tag{3}$$

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

$$h_3 = h_4 \tag{4}$$

The refrigeration capacity of the cycle can be calculated from the rate of enthalpy change in the Evaporator

$$q_r = (h_1 - h_4) \tag{5}$$

where q_r is the specific refrigeration load of the refrigeration cycle. The coefficient of performance (COP) of the refrigeration cycle is then calculated by

$$COP = \frac{q_r}{w_{comp}} \tag{6}$$

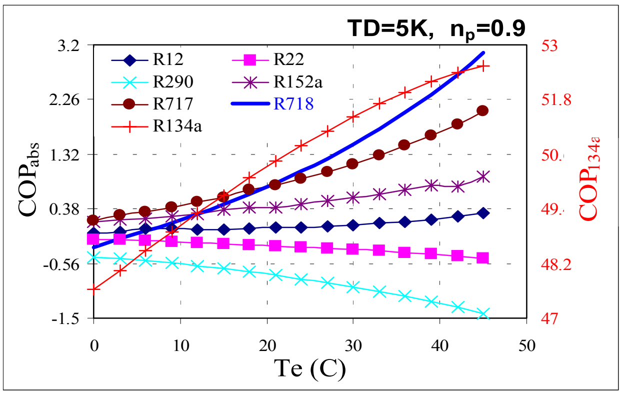
Based on the above model, a computer program was developed calculating COPs for all refrigerants and their absolute differences to those COPs obtained for R134a. Further compressor outlet temperature, pressure ratio, and the temperatures at which water as a refrigerant (R718) gives a better COP than other refrigerants. The computer code uses a commonly available data bank for various refrigerant properties: P, T, h, and s.

For the investigations, three parameters were primarily varied or held constant. These are the evaporator temperature T_e; temperature lift TD, which is the temperature difference between condenser and evaporator; and the polytropic efficiency of the compressor η_p.

RESULTS AND DISCUSSION

With the computer code the evaporator temperature was increased from 0˚C to 45˚C while the temperature lift and polytrophic efficiency were held constant. Absolute COP values (and their absolute differences to the COPs of R134a) of the refrigerants were determined as a function of evaporator temperature. This has been performed with different temperature lifts between 5K and 30K and polytropic efficiencies between 0.5 and 0.9.

Figure 3 shows the variation of COP_{abs} (the absolute difference between the COP of the refrigerant and the COP of the reference refrigerant R134a) versus evaporator temperature for different TD values and constant polytropic efficiency. While the COP of R134a (COP_{R134a}) is given by the right ordinate, the absolute COPs for the other refrigerants can be calculated by adding COP_{abs} from the left ordinate to COP_{R134a}.



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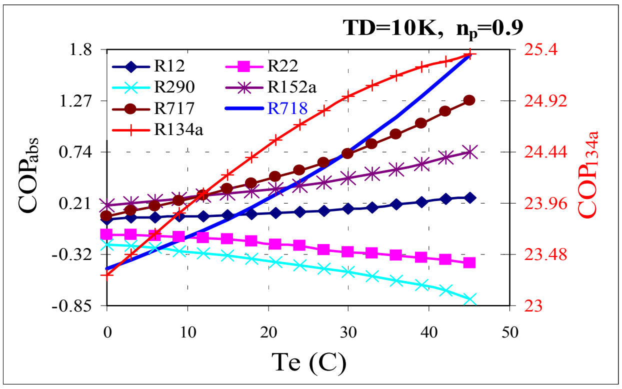

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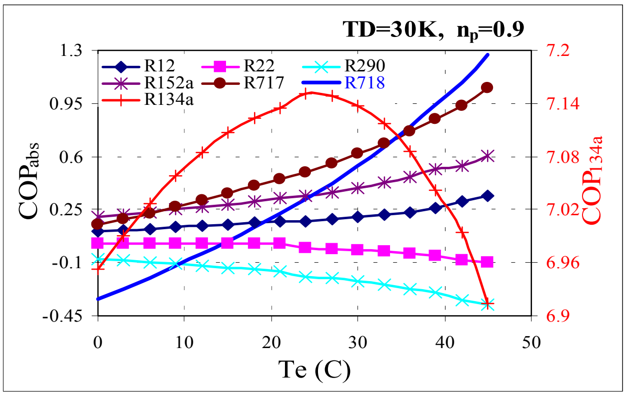

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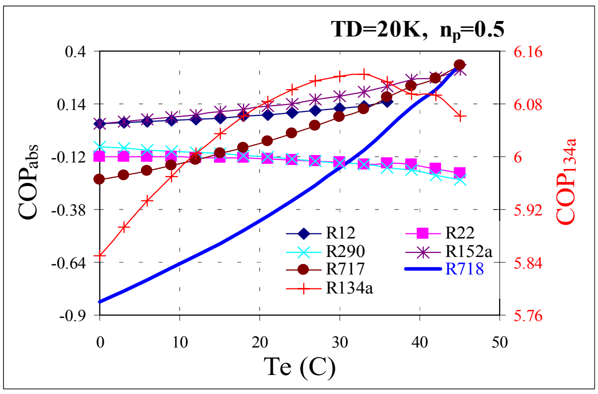
Fig.3.COP $_{abs}$ as a function of evaporator temperature for different TD values a) TD=5K, b) TD=10K, c) TD=30K

For the shown temperature range, with increasing evaporator temperature, the COP_{abs} of the refrigerants increases except those of R22 and R290. As shown in Fig. 3c, for a high TD=30K, the COP of 134a also decreases at evaporator temperature above 23˚C. Furthermore, R718 shows the steepest increase in COP_{abs} for all TD ranges in Fig. 3, which shows the potential for higher economic benefits than with other refrigerants if the evaporator temperature can be raised. As TD values decrease, the temperature range at which R718 shows the best COP spreads out to low temperatures. The evaporator temperatures above which COP_{abs} of R718 is higher than that of the other refrigerants are

34˚C for TD=30K, 30˚C for TD=10K, and 20˚C for TD=5K. Below these evaporator temperatures, R717 produces a better COP. However, despite the fact 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 is explosive in certain mixtures with air. 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 9˚C and for TD=5K, COP values of R718 are better than R12, R22, R290, and R134a.

As TD increases, the pressure ratio increases and, in turn, the compressor power. Simultaneously, with increasing pressure ratio the refrigeration effect decreases. These actions together result in a reduced coefficient of performance for all refrigerants as can be seen by comparing the plots in Fig. 3.

The variation of COP_{abs} values with respect to the evaporator temperature for three different polytropic efficiencies and constant TD=20K is shown in Fig. 4. Isentropic efficiency is mainly a function of pressure ratio and polytropic efficiency. Isentropic efficiency can be determined as a function of the compressor outlet temperature as the pressure ratio is held constant at constant evaporator temperature. As polytropic efficiency increases at constant evaporator temperature, compressor power and compressor outlet temperature decreases. The COP of the cycle increases. Comparing the plots in Fig. 4 for constant TD, at a constant evaporator temperature value, in other words, for constant evaporator and condenser



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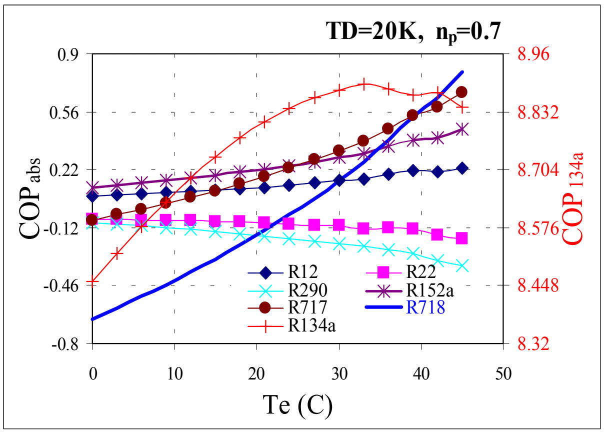

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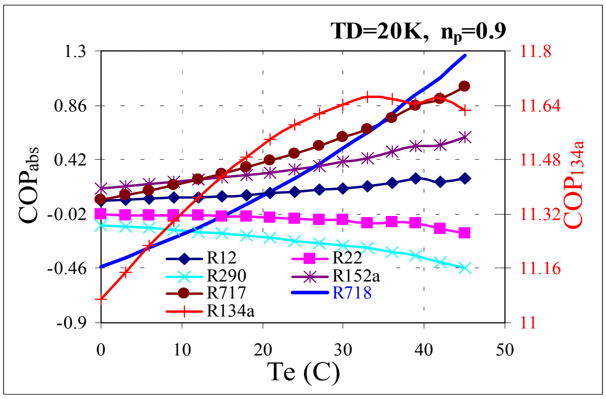

c)

Fig.4. COP_{abs} as a function of evaporator temperature for different η_p values a) η_p=0.5, b) η_p=0.7, c) η_p=0.9

temperature (no change in refrigeration capacity), COP of all the refrigerants increases as polytropic efficiency η_p increases. R718 still shows the steepest increase in COP values between any two successive polytropic efficiencies (0.5-0.7 or 0.7-0.9) as it is compared with other refrigerant. This shows that for R718 that requires high pressure ratio, the development of high quality compressors with high η_p pays off the most. As η_p increases, the temperature range at which R718 has advantages over the other refrigerants increases. For a TD=20K, the evaporator temperatures, above which the calculated COP_{abs} of R718 is higher than those of the other refrigerants, are 45˚C for η_p=0.5, 39˚C for η_p =0.7, and 33˚C for η_p =0.9, knowing that such a temperature lift is a least favorable for R718 as shown further below.

The increase of refrigerant temperature at the compressor outlet (discharge temperature T_2) with increasing evaporator temperature is shown in Fig. 5 for a temperature lift TD=20K and a polytropic efficiency η_p=0.9.

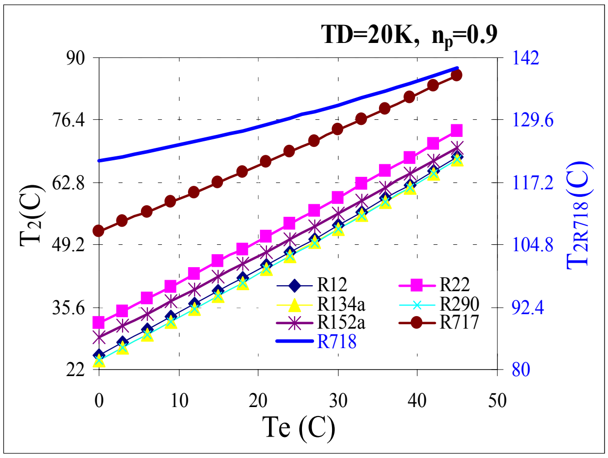

Fig.5. Discharge temperature versus evaporator temperature

The compressor-outlet temperature of water (R718) is given from the right ordinate in Fig. 5. R134a gives the lowest compressor-outlet temperature, while R718 gives the highest. The temperatures of R134a, R290, and R12 are very close to each other. The high compressor-outlet temperature of water is mainly due to the high pressure ratios, which require high compressor work. This disadvantage can be reduced by designing special compressors and applying suitable cooling methods like intercooling for multistage compressors. As mentioned above, this has been realized with centrifugal compressors for water vapor compression applications.

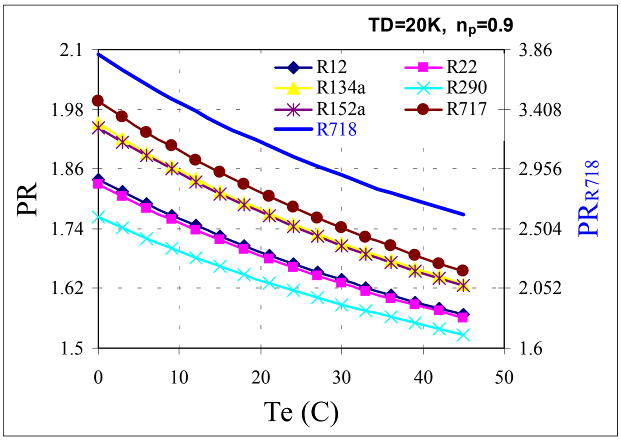

Fig.6. Pressure ratio as a function of evaporator temperature

Figure 6 shows the cycle pressure-ratio versus evaporator temperature for a temperature lift TD=20K and polytrophic efficiency η_p=0.9 . The high pressure ratio for R718 is shown on the right ordinate. When the evaporator temperature increases, the evaporator pressure increases, as does the condenser pressure too, because of constant temperature lift. But the ratio of the increase in evaporator pressure to that of increase in condenser pressure is always greater than 1. Therefore, the ratio of condenser pressure to the evaporator pressure decreases with increasing evaporator temperature. R290 has the lowest pressure ratio, while R718 has the highest pressure ratio. R22 and R12 as well as R134a and R152 respectively show almost the same pressure ratios in the lower mid range in Fig. 6.

Table 1 summarizes the evaporator temperatures above which the theoretical cycle calculation gives a higher COP for R718 than for the other considered refrigerants for a polytropic efficiency n_p=0.9 and temperature lifts TD=5…40K.

Table 1. Evaporator temperatures above which R718 gives a better COP than R290, R22, R134a, R12, R152a, R717. TD=5…40K.and. n_p=0.9.

Evaporator Temperature (˚C)						
R718						
TD	R290	R22	R134a	R12	R152a	R717
5K	0	3	6	7	12	20
10K	7	10	14	16	23	29
15K	10	14	16	20	26	33
20K	11	15	16	21	26	35
25K	11	15	15	20	26	35
30K	10	14	14	19	25	34
35K	8	13	11	17	23	33
40K	6	11	9	15	21	32

CONCLUSIONS

Water as a refrigerant (R718) is compared with current refrigerants including R717, R290, R134a, R12, R22, and R152a by using a created computer code for calculations of a simple vapor compression refrigeration cycle

The computed results show that the use of water as a refrigerant can result in a higher coefficient of power (COP) than if the other refrigerants are used. From the presented results, it can be concluded that for evaporator temperatures above 35˚C the highest COP can always be obtained with R718. The COP is then even greater than if R717 (ammonia) were used.

Also, at lower evaporator temperatures the use of water can result in a higher COP than if other refrigerants were used. This is especially true if the temperature lift (temperature difference between condenser and evaporator) is either relatively small (≲10K) or if it is relatively high (≳30K). The temperature range at which R718 gives a better COP than other refrigerants increases with increasing values of polytropic compressor efficiency. This encourages very much the further development of high quality compressors for R718.

The disadvantages of water as a refrigerant are its high specific volume, the required high pressure ratio, and the resulting high compressor outlet temperature. It has been demonstrated that these technical challenges can be overcome with specifically developed compressors, especially multi-stage turbo compressors with intercoolers between stages. While in today's world a high COP is a key target, it is not the only value that decides the choice of the refrigerant. Environmental parameters like ozone depletion potential (ODP) and global warming potential (GWP) become more and more restrictive. Further price and safety properties of refrigerants are also heavily taken into consideration. In all these aspects, water is the superior refrigerant.

NOMENCLATURE

COP coefficient of performance
c_p constant pressure specific heat (kJ/kgK)
h_1 specific enthalpy of refrigerant at the compressor outlet (kJ/kg)
h_2 specific enthalpy of refrigerant at the compressor outlet valve (kJ/kg)
h_3 specific enthalpy of refrigerant at the condenser outlet (kJ/kg)
h_4 specific enthalpy of refrigerant at the evaporator inlet (kJ/kg)
K ratio of constant specific heats
η_{is} isentropic efficiency of the compressor
η_p polytropic efficiency of the compressor
q_c condenser capacity (kJ/kg)
q_r refrigeration capacity (kJ/kg)
P pressure (kPa)
PR pressure ratio
s specific entropy (kJ/kgK)
R ideal gas constant (kJ/kgK)
T temperature (ºC)
T_2 compressor outlet temperature (ºC)
TD temp. lift (Temperature difference between condenser and evaporator)(K)
w specific work (kJ/kg)

Subscripts
c condenser
comp compressor
e evaporator

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