An Investigation of Water as a Refrigerant

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This paper explores some of the basic thermodynamic and technical considerations involved in using water as a working fluid for refrigeration and heat pump cycles down to its freezing point of 0°C. It is first shown how the integration of the functions of refrigerant and heat transfer fluid can lead to energy savings, especially for the case of ice production. Next, the two fundamental requirements that the compressor must fulfill—handling a very large volume flow and achieving a large compression ratio—are described. A thermodynamic analysis of multistage compression follows to investigate the adiabatic head requirements and the large desuperheating irreversibility. It is concluded that a radically new type of vacuum compressor must be developed in order for water to be used as working fluid in vapor compression refrigeration cycles.

Introduction

In refrigeration and air conditioning systems, water can function both as a heat transfer fluid and as a thermal storage medium. In the last decade, mechanical vapor compression of steam has been developed for open cycle heat pumps in evaporative processes (concentration, drying, etc.). In this instance, water vapor produced by evaporation is compressed so that it condenses at a temperature that is higher than the boiling point of the solution, thereby furnishing energy for further evaporation. These processes usually operate around 100°C. In principle, it is possible to extend mechanical vapor compression of water vapor to lower operating temperatures. Refrigeration is obtained when the evaporation temperature is below the ambient temperature. Alternatively, the cycle can operate as a heat pump with the environment as heat source.

McLinden and Didion (1987) summarize the requirements that a substance must satisfy to function as a refrigerant. A fundamental requirement is chemical stability over the entire range of operating temperatures of the system. Furthermore, it is highly desirable that the refrigerant is non-toxic, non-flammable and environmentally benign. Water completely fulfills all of these demands, distinguishing it from all other refrigerants. The other requirements listed by McLinden and Didion relate to the thermodynamic and transport properties that determine the performance of the refrigeration cycle and the sizing of the equipment. In this respect, water differs fundamentally from conventional refrigerants.

At the triple point, the vapor pressure of water is 611 Pa, which is less than 1% of the atmospheric pressure. This low operating pressure results in a very large specific volume posing a demand on the compressor that has prevented water from being used as the refrigerant in mechanical systems. The combined use of water both as a refrigerant and a heat transfer fluid (or thermal storage medium) permits direct contact heat transfer in the evaporator and/or condenser with potential energy savings.

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System Description

Figure 1 is a schematic of a vacuum ice maker with water as the refrigerant. The vacuum freeze evaporator operates at triple point conditions. The vessel is approximately adiabatic so that the energy transfer during evaporation causes the remaining water to freeze; freezing is thus obtained by causing the water to boil. The latent heats of freezing (~333 kJ/kg) and vaporization (~2500 kJ/kg) are such that the mass of ice produced is 7.5 times the mass of vaporized water. To keep the process going, it is necessary to continuously agitate the water. One way of achieving this agitation is by a set of submerged nozzles and a circulation pump. The resulting slurry of fine ice crystals is readily pumpable. If chilled water is the desired product, the vessel operates as a simple flash tank. Direct contact condensation can be applied in water-cooled condensers. Surface condensers may be used in those cases where distilled water is a useful by-product or where contamination of the chilled water/ice loop cannot be tolerated. Air is introduced with the water entering the system so that the vacuum pump is required to continuously deaerate the ice maker and maintain the triple point pressure.

Early vacuum freeze evaporators were developed for desalination (e.g., Snyder, 1966), and more recently they have been applied for heat pumping with the latent heat of freezing as the heat source (Andersen and Boldvig, 1987). In both cases, the water vapor undergoes only a moderate compression so that it condenses at a temperature slightly above the freezing point, typically 3°C. The remainder of the temperature lift is provided by a refrigeration cycle with a conventional refrigerant. Alternatively, Collet et al. (1985) have investigated direct vapor deposition of the triple-point vapor onto the evaporator coils of the conventional cycle. The ice layer formed during the process is periodically removed by melting.

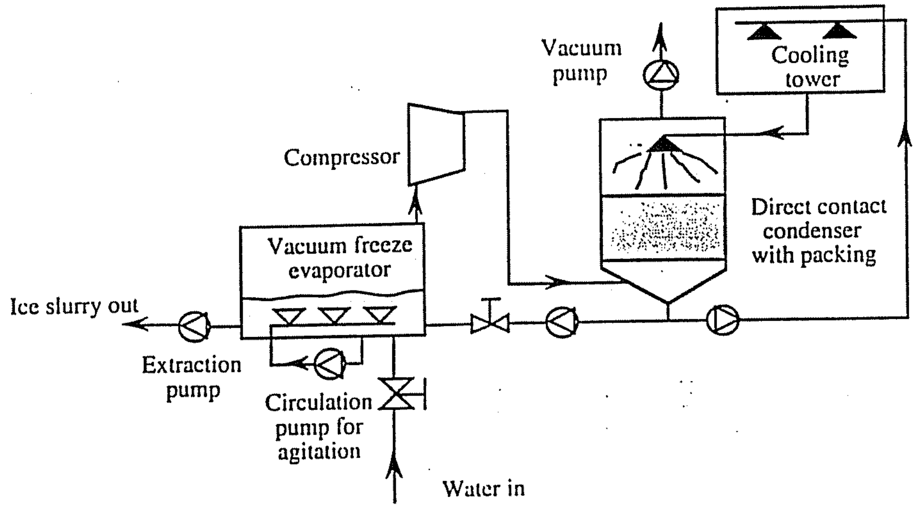

Fig. 1 Vacuum ice maker with water as refrigerant and direct contact condensation

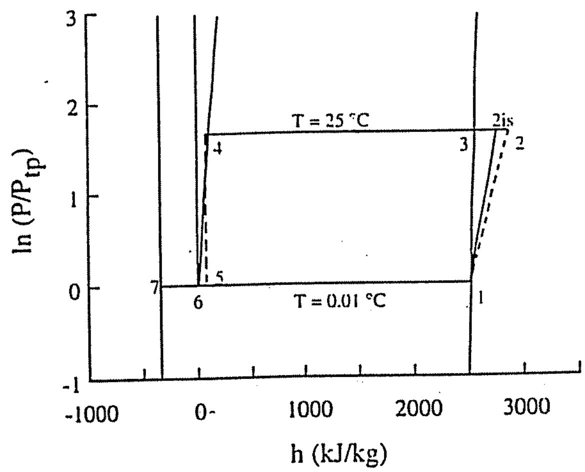

Fig. 2 Vacuum ice making cycle on a pressure-enthalpy-diagram. P_{tp} is the vapor pressure of water at the triple point.

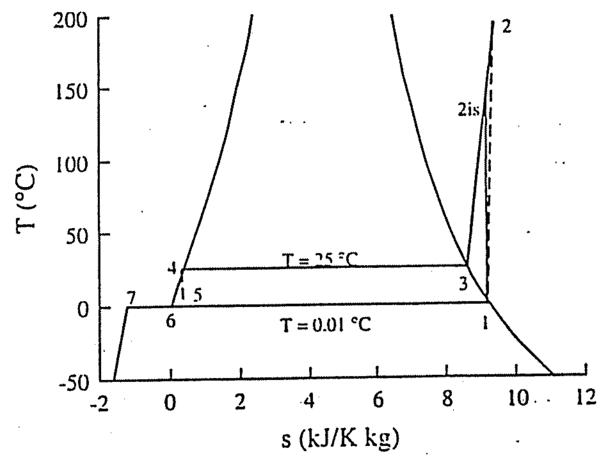

Fig. 3 Vacuum ice making cycle on a temperature-entropy diagram

Figures 2 and 3 show the thermodynamic states of a vacuum ice-making cycle in pressure-enthalpy and temperature-entropy diagrams. The pressure in Fig. 2 is normalized by the triple-point pressure and the scale is logarithmic. The enthalpy of the liquid phase at the triple point is set to 0 kJ. Saturated vapor at the triple point has an enthalpy of approximately 2500 kJ, which is equal to the latent heat of vaporization of water at the triple point. To the left is the saturated solid phase, having an enthalpy of approximately −333 kJ, the latent heat of fusion. At lower pressures, the liquid phase does not exist and the difference between the saturated ice and vapor lines is the latent heat of sublimation. At higher pressures, the liquid and solid phases can be in equilibrium. The enthalpy at saturated conditions is nearly constant in the range of interest to refrigeration cycles. The temperature-entropy diagram has similar features.

The vacuum ice making cycle is similar to an ordinary refrigeration cycle, except that the thermodynamic state in the evaporator corresponds to triple-point conditions. The freezing of the water corresponds to a transition from state 6 (saturated liquid) to state 7 (saturated solid) in Figs. 2 and 3.

Thermodynamic Analysis of Simple Cycles

The thermodynamic calculations presented in this paper were done using an equation-solving program which has built-in

Nomenclature

g = gravitational acceleration (m^2/s)
h = specific enthalpy (kJ/kg)
H_{ad} = adiabatic head (m)
q = heat transfer per; unit mass (kJ/kg)
s = specific entropy (kJ/kg − K)
P = pressure (kPa)
T = temperature (K)
v = specific volume (m^3/kg)
w = work per unit mass (kJ/kg)

Subscripts

cd = condenser pressure
ev = saturation at evaporator pressure
des = for desuperheating process
gen = with s-entropy generation
is = for isentropic process
scd = saturation at condenser pressure
tp = triple point condition

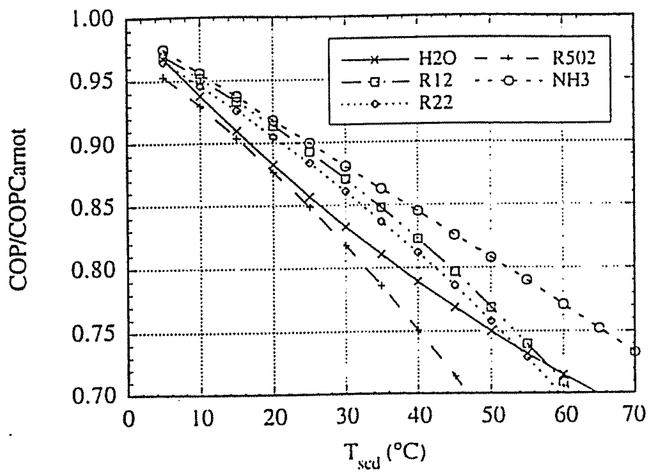

Fig. 4 COP as a function of the condensation temperature for an ideal cycle with an evaporation temperature of 0.01°C

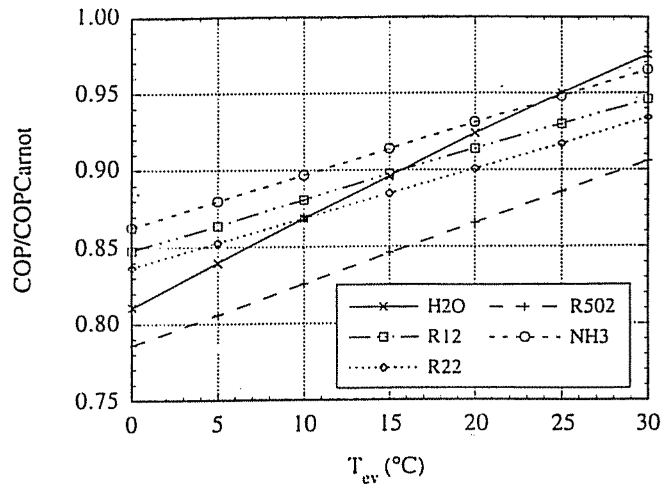

Fig. 5 COP as a function of the evaporation temperature for an ideal cycle and a condensation temperature of 35°C

functions for the thermodynamic properties of refrigerants including water (Klein and Alvarado, 1992). Properties of water were calculated following Hyland and Wexler (1983) for the saturated liquid and ice phases and Young (1988) for the gas phase.

The performance of different refrigerants is first compared for an idealized refrigeration cycle, i.e., with isentropic compression and no pressure drop in the heat exchangers. It is assumed that the refrigerant enters the throttling valve as saturated liquid (no subcooling), and that it is saturated vapor (no superheating) at the compressor inlet.

Figure 4 shows the coefficient of performance (COP) for five different refrigerants at an evaporation temperature of 0.01°C as a function of the condensation temperature (saturation temperature of the refrigerant in the condenser). In this and following figures, the coefficient of performance (COP) is divided by the COP of a Carnot cycle that operates between the same evaporation and condensation temperatures. Since the compressor is isentropic and pressure losses in the heat exchangers are neglected, the deviation from the ideal performance is due only to internal irreversibilities in the cycle caused by throttling and desuperheating. Desuperheating is considered an internal irreversibility in this analysis because the degree of superheating at the compressor outlet is dependent on the choice of refrigerant. Figure 4 shows that the COP for water is lower than for ammonia, R-12 and R-22 over most of the range, but is better than for R-502. Only around 60°C does water match R-12 and R-22, but it still remains far below ammonia.

Figure 5 shows the COP as a function of evaporation temperature for a condensation temperature of 35°C. The COP of water rises more rapidly with increasing evaporation temperature than does the COP of the other refrigerants investigated, surpassing all other refrigerants at high evaporator temperatures. Figures 4 and 5 lead to the conclusion that the thermodynamic properties of water make it a less optimal working fluid at temperatures close to its triple point, i.e., in the normal operating range of cooling cycles. This disadvantage does not persist at temperatures from about 30°C upwards. When a high-temperature heat source is available, water is comparable to other working fluids, at least as far as theoretical energy requirements are concerned.

The danger of freezing inside the evaporator excludes water from being used in traditional cycle configurations. However, the use of a vacuum freeze evaporator or a flash evaporator deals effectively with the problem of freezing and can actually result in a performance advantage.

Ice-producing methods commonly employ surface-freezing techniques with ice forming on one side of a heat transfer surface and an evaporating refrigerant or a colder secondary coolant removing thermal energy on the other side. Due to heat transfer resistances, a temperature difference is needed between the refrigerant and the water; an evaporator temperature of −6°C is assumed for refrigeration cycles using conventional refrigerants in the following calculations. For water it is assumed that the vapor enters the compressor in the saturated state at −0.5°C to account for the heat transfer through the finite liquid-vapor interface. The condensation temperature for water is taken to be the same as for conventional refrigerants, which implies that a surface rather than a direct contact condenser is used. The isentropic efficiency of the compressor is set equal to 0.7 for all cases. The reference Carnot cycle is based on a low temperature of 0.01°C and a high temperature equal to the condensation temperature. The deviation from ideal performance is now caused by 3 factors: internal cycle irreversibilities, nonisentropic compression, and heat transfer at the low temperature side.

Figure 6 shows that reduced heat transfer resistance for the vacuum ice maker results in a significant improvement in the COP, especially at the lower condensation temperatures. At higher temperatures, the internal cycle irreversibilities tend to dominate the heat transfer irreversibility, and the COP difference between alternative refrigerants narrows. Each of the curves shows a maximum. The condenser heat transfer irreversibility varies weakly with condensing temperature. At low temperatures, the reversible work is moderate and the extra work caused by the heat transfer irreversibility is relatively large. Its relative importance decreases with increasing condensation temperature. For still higher condensation temperatures, the internal cycle irreversibilities (throttling and desuperheating) become more important and the relative COP again decreases. The heat transfer penalty for water is very small. The maximum COP occurs at low condensation temperatures. At a typical condensation temperature at 35°C, the larger COP of a vacuum ice maker over surface ice makers corresponds to an energy saving of 11 percent. If the evaporation temperature of the traditional system is −10°C, the savings increases to 21 percent.

Physical Properties of Water and Compressor Selection

The two fundamental thermophysical properties that determine the behavior of a refrigerant are its specific heat and the shape and position of its vapor pressure curve in the pressure-

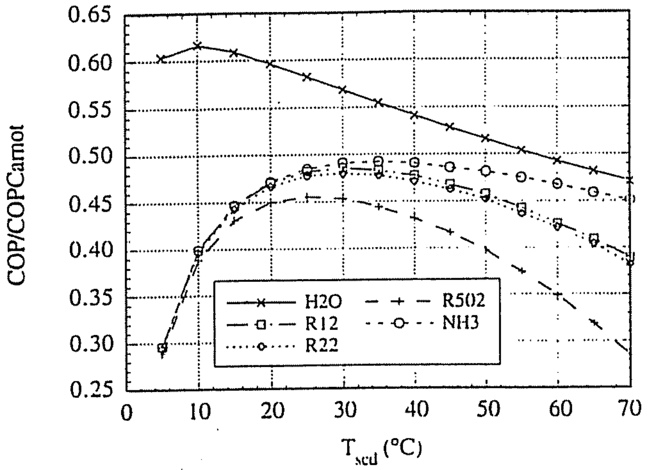

Fig. 6 COP of vacuum ice making versus traditional methods

Table 1 Comparison of the thermophysical properties of water and R22 for an evaporation temperature of 0.01°C and a condensation temperature of 35°C

	H_2O	R22	H_2O/R22
P_{ev} (kPa)	0.611	497.9	$1.23 \cdot 10^{-3}$
v (m^3/kmole)	3714	4.065	914
q_{ev} (kJ/kmole)	42411	13989	3.03
spec. flow (m^3/kJr)	$87.6 \cdot 10^{-3}$	$0.291 \cdot 10^{-3}$	301
P_{cd} (kPa)	5.628	1355	$4.15 \cdot 10^{-3}$
$P_{cd} - P_{ev}$ (kPa)	5.017	927.1	$5.41 \cdot 10^{-3}$
P_{cd} / P_{ev}	9.201	2.722	3.38
w_{is} (kJ/kg)	372	24.8	15.0
H_{ad} (m)	37920	2530	15.0

temperature domain. Table 1 compares some values of water with those of R-22.

For temperatures at or below ambient, the saturation pressure of water is far below atmospheric pressure; at the triple point it is 0.611 kPa and at 35°C it is only 5.628 kPa. Consequently, the specific volume is more than 900 times larger (on a molar basis) than saturated R-22 vapor for an evaporation temperature of 0.01°C. This effect is somewhat compensated by the larger latent heat of vaporization of water, which is about 3 times larger than that of R-22 (again on a molar basis). The volume flow that needs to be processed for a given cooling effect is therefore a factor of 300 larger. The flow handling capacity of even the largest positive displacement compressors is small in comparison to the needs of low temperature water vapor compression. In order to achieve a reasonable cooling capacity, dynamic compressors must be used. The conventional refrigerant that probably most closely approaches the characteristic of high flow is R-11; it also operates under vacuum conditions (but at about 40 kPa at 0°C) and it is normally used in combination with centrifugal compressors.

In addition to the low vapor pressure at evaporator conditions (0°C), the change in the vapor pressure of water with temperature is steeper than for most traditional refrigerants. As a result, the ratio of the condenser pressure to the evaporator pressure is larger than usual. This ratio determines the compression work per unit mass, which for isentropic compression is given by

$$w_{is} = h_{2is} - h_1 \tag{1}$$

where h the specific enthalpy and states 1 and 2 are shown in Fig. 2. As indicated in Table 1, the isentropic work is 15 times larger for water than for R-22 under identical operating temperatures. An equivalent quantity, which is commonly used in compressor technology, is the adiabatic head, defined as:

$$H_{ad} = \frac{W_{is}}{g} \tag{2}$$

where g is the gravitational acceleration. The values found in the literature for the adiabatic head that can be achieved per stage in centrifugal compressors vary from about 6,000 m to as high as 20,000 m in special applications. Thus, the corresponding number of stages required to achieve a condensation temperature of 50°C in a vacuum ice maker (i.e., an adiabatic head of almost 60,000 m) lies between 10 and 3. Multistaging is thus an inevitable necessity if water is to be used as refrigerant. A conventional refrigerant with a similar high adiabatic head is ammonia. Since its volume flow per unit cooling capacity is very small, its is always used in conjunction with positive displacement compressors, which can easily attain high pressure ratios.

A vacuum ice making cycle with water as refrigerant requires a vacuum compressor that can at the same time process a very large volume flow and achieve a large compression ratio. The combination of both demands is so extreme that it has prevented the use of this ice making configuration thus far. Application of standard compressors makes the system prohibitively expensive. However, the vacuum operating conditions result in a very small pressure difference between the condenser and evaporator, as seen in Table 1. This small difference results in small aerodynamic forces on internal compressor parts, offering the opportunity for very light construction and possibly unconventional materials such as fiber-reinforced synthetics (as in the blades of modern windmills) which can lower the capital costs of vacuum compression systems. Only if this opportunity is fully exploited can water become an economically competitive refrigerant.

A compressor development in this respect was reported by Ophir and Paul (1991). It concerns an extended design of the compressors used in desalination applications. The centrifugal compressor has a diameter of 2.5 m and flexible, titanium alloy steel blades with a thickness of 1.5 mm. The flow rate reaches 300 m^3/s and the pressure ratio is between 2 and 3. At triple-point inlet conditions, these values correspond to a refrigeration capacity of 3.3 MW and to an adiabatic head of approximately 16,500 m (for a pressure ratio of 3). Ophir and Paul propose a vacuum ice maker with two such compressors (driven by separate electric motors) in series, with different contact condensation and with intercooling by means of water spray. For this configuration the maximum condensation temperature can be estimated to be about 35°C. This system appears to be the first ever to achieve a refrigeration cycle by means of mechanical vapor compression with only water as refrigerant. The authors claim that the light compressor construction makes the design economically viable.

Another important thermophysical feature of water is the low specific heat of the vapor on a molar basis. At 25°C, only molecular translation and rotation (and not vibration as for most refrigerants which have a somewhat more complex molecular structure) contribute significantly to the specific heat. This low specific heat, in combination with the large work input per unit mass, leads to high superheat temperatures for the case of adiabatic compression. Figure 7 shows the compressor outlet temperatures, T_{CO}, for an idealized isentropic refrigeration cycle. Even ammonia, a refrigerant known for

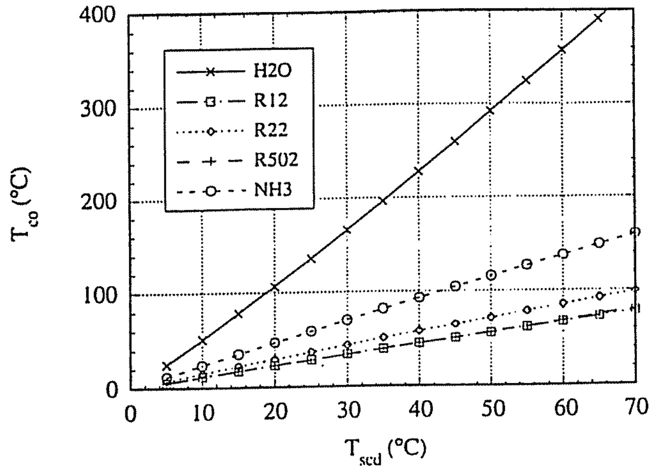

Fig. 7 Compressor outlet temperatures as a function of the condensation temperature for an evaporation temperature of 0.01°C

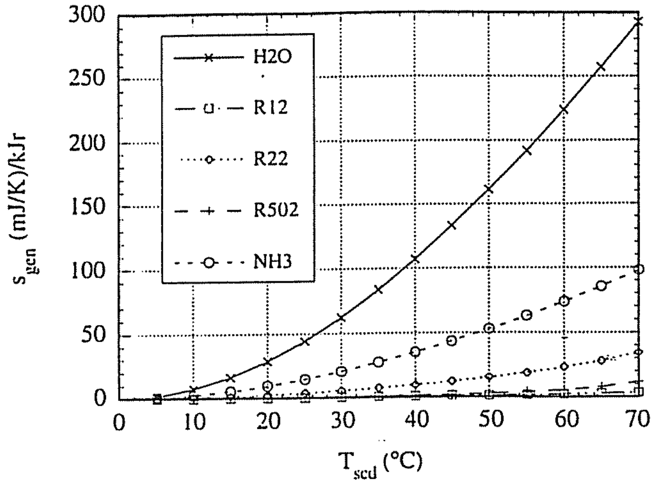

Fig. 9 Entropy generation due to desuperheating

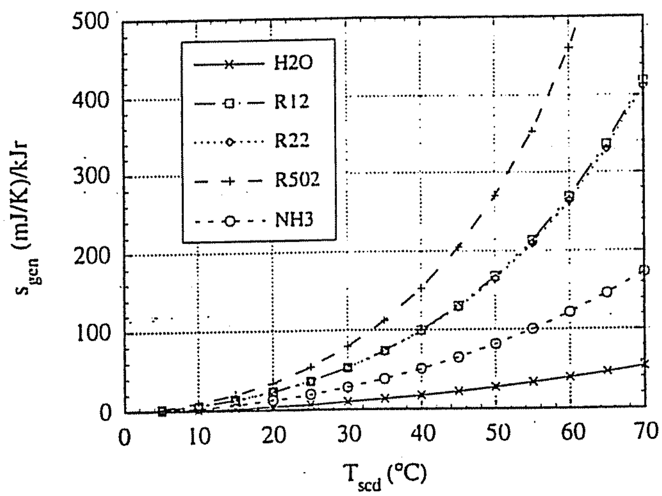

Fig. 8 Entropy generation during the throttling process

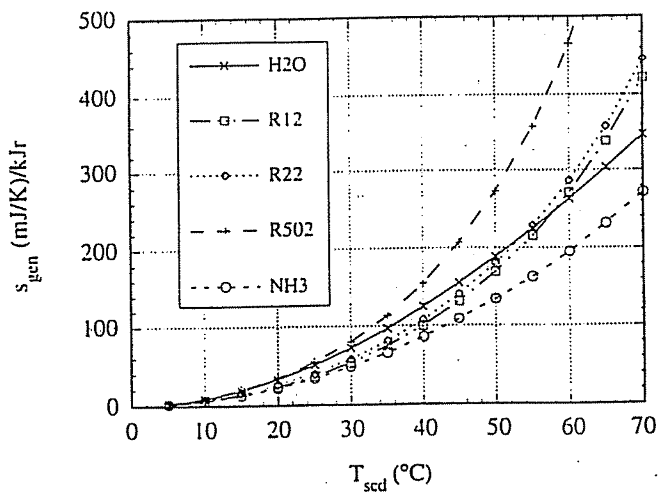

Fig. 10 Total entropy generation for an ideal cycle

its high superheat, has a far lower compressor outlet temperature.

The internal cycle irreversibilities in an ideal refrigeration cycle are caused by the irreversible throttling and the desuperheating process (i.e., heat is rejected to the environment at a temperature higher than the condensing temperature). The entropy generated during the throttling process can easily be determined from the inlet and outlet states of the throttling valve. In order to account for the different mass flows of different refrigerants, the generated entropy, s_{gen}, is plotted per unit of cooling capacity in Fig. 8. On this basis, water has a smaller entropy generation than the four other refrigerants. The exceptional performance of water in this respect can also be seen in Fig. 3 where line 4–5, representing the throttling process, is almost vertical. This relatively small irreversibility can be attributed to the small vapor fraction of water after throttling, which results from its high latent heat and small molar specific heat (in the liquid state). Throttling liquid between two pressure levels results in a much smaller irreversibility than the same process in the gas phase.

The entropy generation caused by desuperheating is a function of the thermodynamic properties of the refrigerant. Refrigerants having a low vapor specific heat achieve a high compressor outlet temperature. The entropy generation caused by desuperheating can be calculated as follows. The heat transfer during desuperheat (q_{des}) is

$$q_{des} = h_2 - h_3 \tag{3}$$

with states 2 and 3 as defined in Figs. 2 and 3. If this energy transfer were to occur at the condensation temperature, the entropy rejection associated with q_{des} under this condition would be

$$s_{scd} = \frac{q_{des}}{T_{scd}} \tag{4}$$

The actual entropy change during the superheating process (2)-(3) is lower. The difference between s_{scd} and the actual entropy change is a measure of the entropy generation resulting from desuperheating.

$$s_{gen} = s_{scd} - (s_2 - s_3) \tag{5}$$

In Fig. 9, the entropy generation due to superheating per unit cooling capacity is plotted versus the condensation temperature for different refrigerants. The high desuperheating irreversibility of water is evident.

Figure 10 shows the sum of the throttling and desuperheating irreversibilities. Water has larger internal cycle irreversibilities than the other conventional refrigerants, with the exception of R-502. Only at a condensation temperature of 60°C does water match R-12 and R-22. These conclusions agree with those drawn concerning the COP of ideal cycles in Fig. 5.

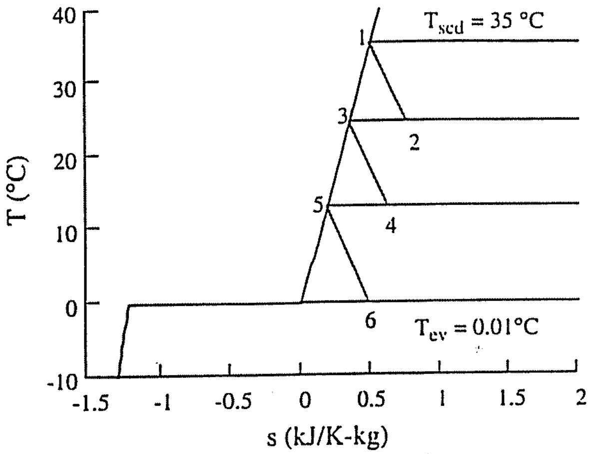

Fig. 11 Three-stage throttling process

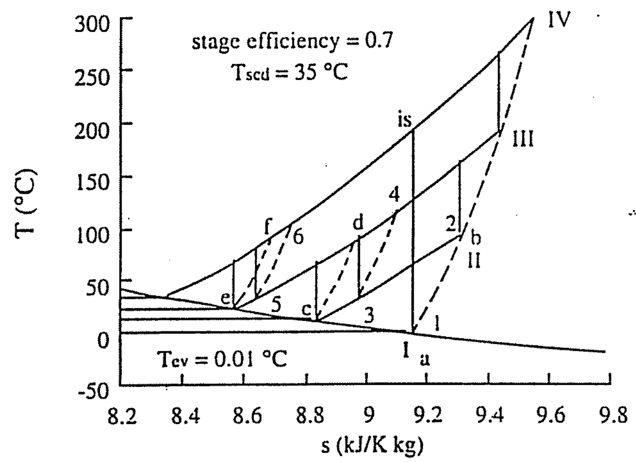

Fig. 12 Three-stage compression without intercooling, with external intercooling and with refrigerant intercooling

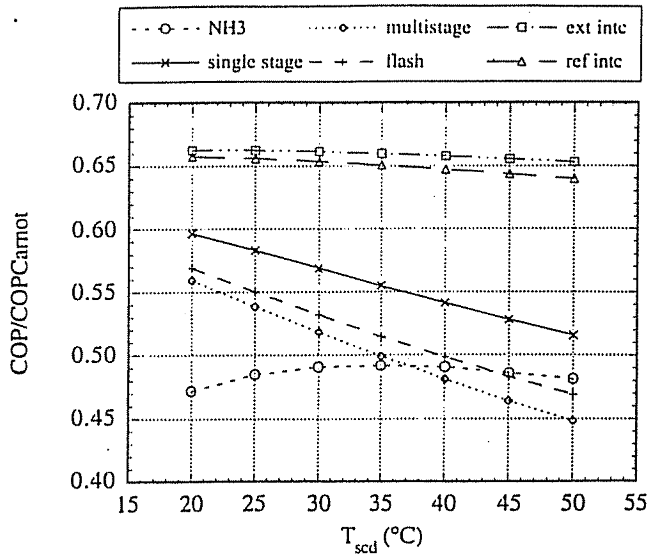

Fig. 13 COP/COP$_{Carnot}$ for different multistage configurations

Multistaging

Although the need for multistaging definitely is a severe drawback, it permits some modifications to the simple cycle that can lead to a significant performance improvement. In the following analysis, the compression is assumed to occur in 7 consecutive stages. This number is chosen so that a condensation temperature of 50°C can be reached for a modest isentropic work input of 90 kJ/kg per stage, corresponding to an adiabatic head of about 9150 m per stage. The intermediate pressures are determined such that, for isentropic compression, each state contributes an equal amount to the overall work (per unit mass), i.e., all stages deliver the same head. Every stage is modeled with an isentropic efficiency of 0.7.

Different modifications to a simple cycle are illustrated in Figs. 11 and 12. For clarity, the processes are drawn for 3 stages only. A simple modification consists of replacing of the single throttling process by a number of small expansion steps, as depicted in Fig. 11. The intermediate pressures correspond to the pressures between two compressor stages. After each flashing, the generated vapor is separated and led to the corresponding interstage compressor duct, where it mixes with the bulk vapor flow. The saturated liquid undergoes another expansion process, and so forth. In this way, unnecessary expansion of all the vapor to the evaporator pressure is eliminated and the flow through the lowest compressor stages reduced. This mode of operation is called economizer operation.

A further modification is the cooling of the superheated vapor between two stages. Adiabatic multistaging with no intercooling corresponds to a process along states I-II-III-IV in Fig. 12. Intercooling can be done by injecting liquid water or by rejecting heat to cooling water in a heat exchanger. With water injection, the vapor is cooled down to its saturation temperature as the liquid vaporizes. In Fig. 12, the vapor leaves the evaporator at state a and is nonisentropically compressed to point b in the first stage. The liquid spray cools the vapor to the saturated state c. A second compression to state d follows with another intercooling to state e. After a final compression to state f, the steam enters the condenser. By adding the spray water, the mass flow increases in each stage. If the impeller is susceptible to erosion damage from two phase flow, special precautions must be taken to avoid droplets from entering each stage (e.g., louvers or demisters).

In the case of intercooling with a heat exchanger, the superheated vapor cannot be cooled below the ambient temperature (or inlet temperature of the cooling water). In the following calculations, it is assumed that the vapor could be desuperheated to a temperature equal to the condensation temperature. Since the heat transfer coefficient for gas is much lower than for condensation and since a large interstage heat exchanger may result in a significant pressure drop, this assumption will be optimistic. In Fig. 12, the processes with external intercooling go through stages 1-2-3-4-5-6. The temperature at points 3 and 5 is 35°C, as is the condensation temperature. With 7-stage compression, the exit temperature of the first stage is lower than the condensation temperature. External intercooling was therefore modeled with intercooling starting after the second stage.

Figure 13 shows the performance of the different configurations. An ice making cycle with ammonia is used for comparison for which the compressor inlet is saturated vapor at −6°C. For all water cycles, the compressor inlet stage is saturated vapor at −0.5°C. The nonisentropic compression with an overall efficiency of 0.7 (labeled "single stage") also corresponds to the same case in Fig. 6. This cycle will be considered the base case for further comparisons.

The COP for 7 stage compression with each stage having an isentropic efficiency of 0.7 (labeled 'multistage' in Fig. 13) is lower than that for the base case. At a condensation temperature of 35°C, the isentropic efficiency of the overall 7 stage compression process is 0.63. The higher work input per stage also causes the design head (90 kJ/kg) to not reach the desired condensation temperature of 50°C. Multistaging in a

Table 2 Number of stages required to obtain a condensation temperature of 50°C as a function of the work input or adiabatic head per stage

W_{is} (kJ/kg)	H_{ad} (m)	# stages w/o intercooling T_{scd} = 50°C	# stages with intercooling T_{scd} = 50°C
60	6116	10	7
90	9174	7	5
150	15291	4	3
190	19368	3	2.5

further unmodified cycle thus not only destroys much of the gain of eliminating the heat transfer in the evaporator, but, in addition, aggravates the already extreme compressor requirements.

The simple cycle modification to multistage flashing (labled "flash") only partially compensates for the loss in performance of multistage compression. The small improvement is primarily due to a reduction of the mass flow through the lower stages of the compressor. Also, the interstage mixing of saturated flash vapor with the superheated vapor somewhat reduces the inlet temperature of the next stage. In Fig. 11, the flash modification would result in a slight shift of points II and III to the left. However, since the amount of flash vapor is relatively small, this cooling effect is limited: under most conditions it is barely a few degrees Celsius. It is expected that multistage flashing results in an only moderate performance improvement since it was found that for water the throttling irreversibilities are only a small fraction of the desuperheating irreversibilities, and desuperheating irreversibilities are not affected by the multistage throttling.

Interstage desuperheating (in addition to multistage flashing) has a much larger impact, as seen in Fig. 13. Under the assumptions made above, intercooling by means of an external heat exchanger (labled "ext intc") turns out to be slightly better than intercooling through injection of liquid refrigerant (labeled as "ref intc"). The ratio of the COP to the Carnot COP is nearly constant with increasing condensation temperature. Since, for the case of external intercooling, the mass flows through the different stages remain unchanged as compared to simple flashing, the improvement is solely due to the much lower stage inlet temperatures (taken equal to the condensation temperature). The pressure lines converge towards the left in the temperature-entropy diagram, corresponding to a lower compression work per unit mass. Interstage desuperheating not only improves the COP, but also it allows a higher condensation temperature to be reached. Maximum superheat temperatures are on the order of 50°C above the condensation temperature. As can be seen in Fig. 12, the high superheat spike is replaced by a seesaw of much smaller triangles, effectively reducing the irreversibility and more closely approaching the ideal isothermal compression. In the case of liquid intercooling, the extra, but small, irreversibility of throttling an additional amount of liquid water clearly is more than outweighed by the strong reduction of the desuperheating irreversibility. The superheat temperature remains below approximately 30°C (in contrast to the hundreds of degrees Celsius for the simple adiabatic compression in Fig. 7). The increasingly larger mass flow through the higher stages (on the average some 15 percent extra in the last stage) is more than compensated for by a lower work per unit mass due to lower inlet temperatures. For a given head per stage, the configuration with spray intercooling therefore requires the lowest number of stages to reach a specified condensation temperature. The results in Table 2 indicate that the number of stages can be reduced by roughly one fourth as compared to the case without intercooling.

Another feature of multistage is the progressively smaller volume flow through the higher stages. As the pressure increases, the specific volume decreases. For the cases of adiabatic compression and multistage flashing, the volume flow of the seventh stage is only approximately 1/4th of that of the first stage. With intercooling a reduction by a factor of roughly 6 is observed. The flow requirements for the highest compressor stages are thus somewhat reduced.

Conclusions

Water possesses highly desired qualities of a modern refrigerant: it is chemically stable, nontoxic, nonflammable and environmentally benign. In this last respect it is unrivaled by any traditional refrigerant. Water can function as working fluid for heat pump and refrigeration cycles down to evaporation temperatures of 0°C. In simple cycle configurations, its thermophysical properties (specific heat, shape and position of the vapor pressure curve) make it a slightly less efficient medium than conventional refrigerants. However, by integrating the functions of refrigerant and heat carrier, i.e., by applying direct contact heat transfer in the condenser and/or evaporator, the energy consumption of water-based systems can be less than systems with conventional refrigerants. The improvement is especially significant for ice making, but less important for the case of water chilling. Vacuum ice makers have the additional advantage that the produced ice is in the form of a readily pumpable slurry. Further energy savings evolve from multistage compression in combination with multistage expansion and intercooling. Intercooling effectively cuts down the otherwise large desuperheating irreversibility.

The very low operating pressure and the steepness of the vapor pressure curve result in the extreme combination of two compressor demands: the vacuum compression system must at the same time process huge volume flows and deliver a large adiabatic head. Only multistage, dynamic compressors can achieve a reasonable refrigeration capacity. In order to be economically competitive, it is necessary to develop an entirely new type of compressor specifically for the purpose of efficiency, high-flow vacuum compression. Full advantage must be taken of the low aerodynamic load which allows lightweight construction.

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