Simulation investigation into the comparative performance of different vacuum ice refrigeration systems and a conventional ammonia system for the cooling of deep mines

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Nomenclature

B	Split parameter
COP	Coefficient of performance
c_p	Specific heat [kJ/kg°C]
h	Enthalpy [kJ/kg]
m	Mass flow [kg/s]
P	Pressure [Pa]
Q	Heat flow [kJ]
s	Entropy [kJ/kgK]
T	Temperature [°C]
Δt	Temperature difference across hybrid system heat exchanger [°C]
U	Total heat transfer coefficient [W/m²°C]
x	Quality

Greek letters

η	Efficiency of compressor
ϵ	Condenser efficiency

Subscripts

a	Ammonia
c	Condenser
e	Evaporator
fg	Difference between fluid and gas
f	Fluid
g	Gas
i	Intermediate
s	Constant entropy
t	Triple point
w	Water
wi	Water inlet
wo	Water outlet

A simulation model for vacuum-ice refrigeration systems was developed. The model was extended to deal with three variants of the vacuum-ice refrigeration system, i.e. single-stage compression, two-stage compression, and a hybrid system. The model was then used to compare the performance of these systems with a conventional ammonia refrigeration system. The hybrid vacuum-ice system was found to have the highest COP of the vacuum-ice systems, while the COP of the ammonia system was higher than those of any of the vacuum-ice systems.

Introduction

One of the recent trends in the cooling of deep mines is the use of ice instead of chilled water. The advantage of using ice stems from the fact that ice has a cooling capacity approximately 5 times that of water, which implies that 5 times less refrigerant has to be moved down the mine and back to the surface when using ice instead of water.[1]

Due to the higher production cost and low capacities of traditional ice machines, slurry ice making processes were developed by the mining industry. Sheer *et al.* pointed out that there are three different processes for the manufacturing of slurry ice. These are the indirect method, the vacuum-ice process, and the secondary refrigerant process. The vacuum process seems to have a cost advantage over the other processes and is already used on at least one mine in southern Africa.[3] This process will therefore be the focus of this paper.

There are at least three variations of the basic vacuum-ice process. These are single stage compression, two-stage compression, and the hybrid system which consists of a single stage vacuum-ice system coupled to a conventional ammonia system. The purpose in this paper is to compare the theoretical performance of the three variations and also to compare them with a conventional ammonia system. The performance of the different systems is calculated, using simplified simulation models, which are discussed.

The vacuum-ice process

This process is based on the common vapour compression refrigeration cycle. It differs, however, from the conventional cycle in that it incorporates a direct contact evaporation process, using the water itself as the refrigerant. Evaporation occurs at the triple point where water exists as a solid, a liquid, and a vapour in equilibrium. For pure

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water this point is at an absolute pressure of 611 Pa and a temperature of 0.001°C.

The water to be frozen is contained in a vessel at low pressure. Water vapour is withdrawn, producing a partial vacuum and resulting in the evaporative cooling of the bulk of the water. Ice is formed at the pressure corresponding to the triple point. With continuing withdrawal of the vapour more ice will form. The ice slurry can be removed from the vessel and replaced with water in a continuous process, as shown in Figure 1.

Simulation models

Single-stage compression model

Both the vacuum-ice system and the conventional ammonia systems are examples of the vapour compression cycle. Figure 2 shows a simplified temperature entropy diagram for this cycle.

The purpose of the simulation model is to determine the states, 1 to 5, from which the heat transfer rates in the condenser and evaporator and the power input to the compressor can be calculated. This can be accomplished by developing component models for the compressor, evaporator, and condenser with which the required states can be obtained.

In a vacuum-ice system, the evaporator temperature, T_e, is fixed at the triple point temperature which eliminates the requirement for an evaporator model.

The compression process can be modelled by using the definition of isentropic efficiency. State point 2 is calculated from the following equation derived from the definition of isentropic efficiency:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_s} \tag{1}$$

The condenser is modelled with the NTU-effectiveness method. Before this model can be explained, it is necessary to examine the heat transfer in the condenser. Consider a counter-flow tube-in-tube condenser as shown in Figure 3. Superheated vapour enters the inner tube while the cooling liquid flows through the annulus. Figure 4 shows the variation of temperature along the length of the condenser. Positions 2 and 4 in Figure 4 correspond with the same points on the $T - s$ diagram of Figure 2.

Desuperheating normally takes place in a small section of the condenser. For the sake of simplicity, it was decided not to model the desuperheating process, but to assume that all the heat transfer in the condenser occurs at the constant temperature T_c. The resulting calculation error is usually smaller than 5%.

In the condenser model, the unknown to be solved for is the condensing temperature T_c. This can be done with the NTU-effectiveness method described by Holman.[4] With this method, the effectiveness of a heat exchanger is defined as

$$\epsilon = \frac{(mc_p)_{\min} \cdot \Delta T_{\min}}{(mc_p)_{\min} \cdot \Delta T_{\max}} \tag{2}$$

where the subscript $_{\min}$ refers to the fluid with the minimum (mc_p) value, and $\Delta T_{\max}$ is the maximum temperature difference in the heat exchanger. In the case where one fluid is changing phase, the other fluid has the minimum (mc_p) value. The reason is that during a phase changing process the fluid acts as if it has infinite specific heat. The effectiveness of the condenser can therefore be expressed as:

$$\epsilon = \frac{T_{\mathrm{wo}} - T_{\mathrm{wi}}}{T_c - T_{\mathrm{wi}}} \tag{3}$$

where T_{wi} and T_{wo} are the cooling water inlet and outlet temperatures, respectively. An analytical expression for ϵ can be derived in terms of the number of transfer units N. This expression is given by Holman:[4]

$$\epsilon = 1 - e^{-N} \tag{4}$$

with N defined as

$$N = \frac{UA}{(m.c_p)_{\text{cooling fluid}}} \tag{5}$$

where U and A are the total heat transfer coefficient and heat transfer area of the condenser, respectively.

The value of the condenser temperature, T_c, is obtained by solving equations (3) and (4) simultaneously. Figure 5 shows the logical flow diagram of the simulation model.

The model starts by reading the input data. An initial value is guessed for the condenser temperature, T_c, which is denoted as $T_c{}^*$. This value is used to calculate the enthalpy at state points 2 and 4 which in turn is used to calculate the condenser heat transfer q^*. The number of transfer units N is calculated with (5) which is used to calculate condenser effectiveness (4). Equation (3) is then used to calculate a new condenser temperature from which a new condenser heat transfer q is calculated. The process is repeated until the difference between q and q^* is sufficiently small.

Two stage compression model

The first variant of the vacuum-ice system is the two-stage compression system (TSC). In this system the water vapour is compressed in two stages with intercooling between the stages.

The simulation model for the TSC differs from the single stage compression system (SSC), only with respect to the compressor model. The $T - s$ diagram for the TSC system is shown in Figure 6. It follows that the vapour is compressed to an intermediate state 6 after which it is cooled to the saturated state 7 and again compressed to the final state 8. The inter-cooling between the compressors reduces the compressor power required and results in a higher COP[5]. The cooling required can be provided by an external source or by means of an economizer where refrigerant from an intermediate state is used to cool the superheated vapour, as shown in Figure 6.

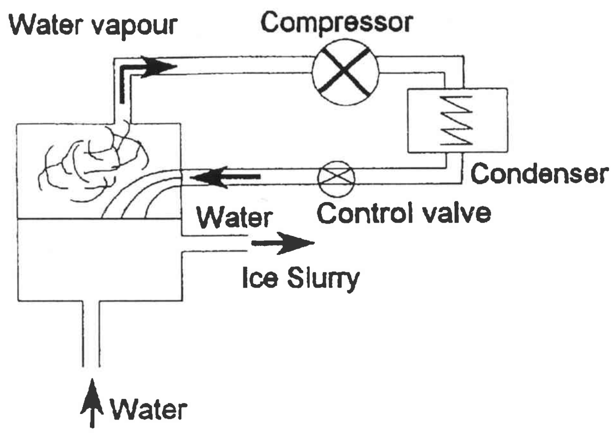

Figure 1 Slurry ice with the vacuum process

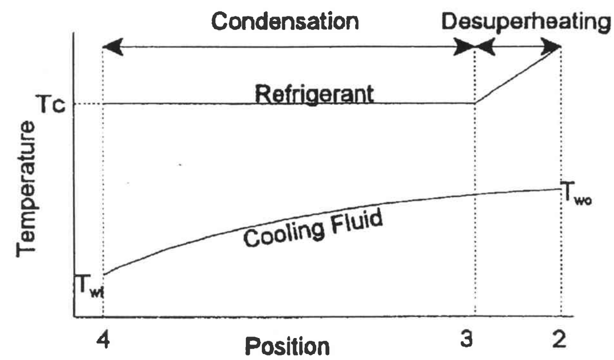

Figure 4 Temperature profile through condenser

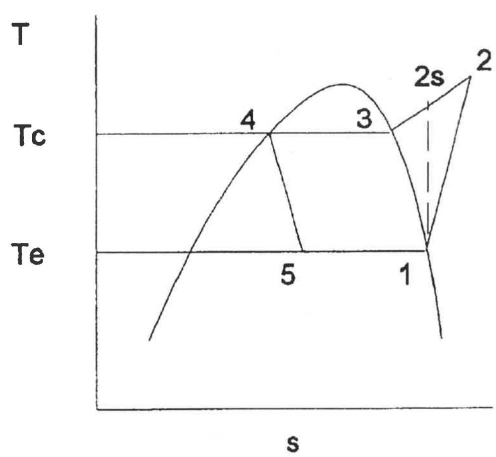

Figure 2 The $T-s$ diagram for the vapour compression cycle

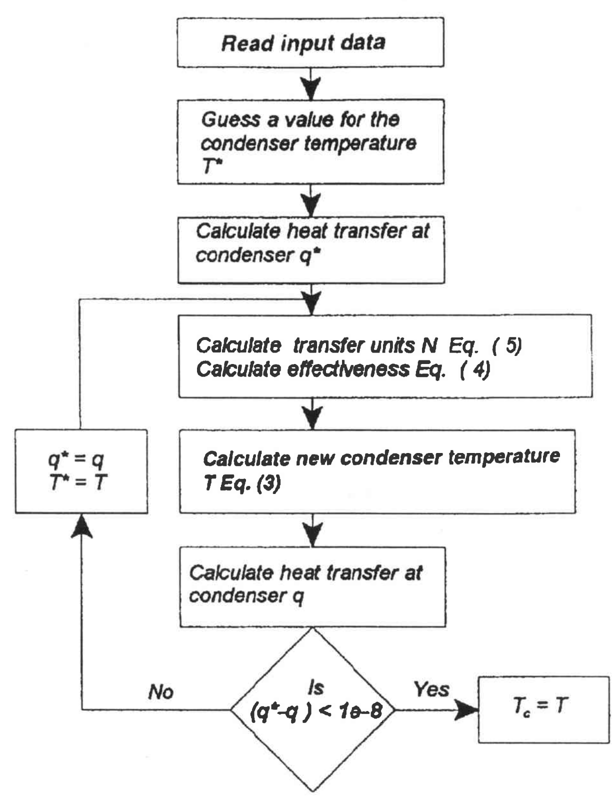

Figure 5 Logic flow diagram for single-stage compression simulation model

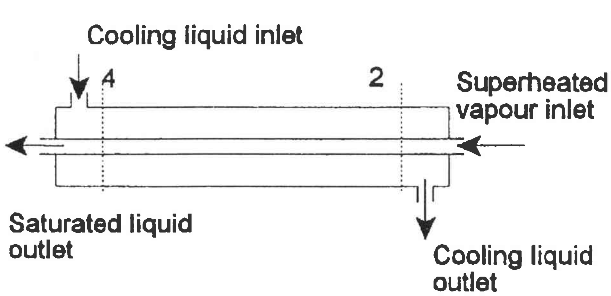

Figure 3 Liquid flow in condenser

As the saturated liquid is expanded from 4 to 5, it is first expanded to an intermediate pressure where all the flash gas and a sufficient quantity of the saturated liquid is removed and mixed with the superheated vapour at 6 to cool it to the saturated state 7.

The cycle is solved as follows: firstly, a value for the intermediate pressure P_6 is assumed. With T_1 and P_1 known, state 6 can be solved by applying the definition of isentropic efficiency between states 1 and 6. The enthalpy at 9 is equal to the enthalpy at 4 which enables us to write

$$\begin{aligned} h_4 = h_{f_4} &= h_9 \\ &= h_{f9} + x_9 h_{fg_9} \end{aligned} \tag{6}$$

Therefore

$$x_9 = \frac{h_4 - h_{f_9}}{h_{g9} - h_{f9}} \tag{7}$$

State 7 is known to be the saturated state and is obtained by mixing $x_9$kg of saturated vapour at state 9_v and m_9 kg of saturated liquid at state 9_f with $(1 - x_9 - m_9)$ kg of superheated vapour at state 6. The only unknown therefore is the mass of the saturated fluid m_9. Applying the energy equation between states 6, 7, and 9, we get

$$\begin{aligned} &h_7 = x_9 h_{g_9} + m_9 h_{f_9} + (1 - x_9 - m_9)\, h_6 \\ &\text{therefore} \\ &m_9 = \frac{h_7 - x_9 h_{g_9} - (1 - x_9) h_6}{h_{f_9} - h_6} \end{aligned} \tag{8}$$

State 8 can be determined by applying the definition of isentropic efficiency between 7 and 8.

The hybrid system model

The third derivative of the vacuum cycle is the hybrid or cascaded system. In such a system two independent cycles are coupled via a heat exchanger. The first cycle is a single stage vacuum-ice system whilst the second is a conventional ammonia cycle. The second cycle is used to absorb the heat rejected at the condenser of the first cycle, as shown in Figure 7.

The nett effect is the same as in the case of the TSC system where the required pressure ratio across each compressor is reduced, which results in lower compressor and power costs.[6]

The $T-s$ diagram for the hybrid system is a combination of the $T-s$ diagrams for two conventional vapour compression cycles. The left-hand diagram in Figure 8 shows the water cycle whilst the diagram on the right represents the ammonia cycle. The figure shows a temperature difference between the water side condenser temperature and the ammonia side evaporator temperature. This temperature difference is necessary to allow heat transfer between the evaporator and condenser.

The system can be modelled by using a simple cycle analysis between the triple point temperature and the chosen condenser temperature for the water cycle. The ammonia cycle is modelled as a single stage compressor model with ammonia as refrigerant.

The ammonia side evaporator temperature will be less than the waterside condenser temperature. The specific value depends on the geometry of the evaporator and condenser and, for the purpose of this paper, will be chosen as fixed. The ammonia evaporator temperature can thus be given as:

$$T_{ea} = T_{cw} - \Delta t \tag{9}$$

where T_{ea} is the ammonia evaporator temperature, T_{cw} the water side condenser temperature and Δt the fixed temperature difference. Although Δt is fixed the optimum value of T_{cw} and thus T_{ea} will be determined in this paper.

Ammonia cycle model

The ammonia cycle is modelled with the single stage compression vacuum-ice model, with the only difference that ammonia instead of water is used as refrigerant. The purpose of the ammonia system model is to provide a benchmark with which to compare the performance of the vacuum ice systems.

Simulation investigation

In the simulation investigation, the COP and specific capacity of the three vacuum-ice variations and the ammonia system are compared. As a first step the influence of the intermediate pressure in the case of the two-stage compression system and the intermediate temperature in the case of the hybrid system on the COP are evaluated. Secondly, the influence of the temperature difference across the heat exchanger of the hybrid system is investigated. This allows for optimum two-stage and hybrid systems to be compared with the other systems.

The optimum intermediate pressure for the two-stage compression system

It is convenient to express the intermediate pressure as a fraction, which is defined as

$$B = \frac{P_i - P_t}{P_c - P_t} \tag{10}$$

where P_i is the intermediate pressure, P_c is the condensing pressure, and P_t is the triple point pressure. Before the optimum fraction can be calculated, it is necessary to establish that an optimum point does exist. The optimum fraction is where the COP is maximum. Figure 9 compares the COP as a function of the intermediate pressure fraction for a two-stage system without intercooling, a two-stage system where the intercooling is done with flash gas, and a two-stage system where the superheated gas after the first compressor is cooled to saturated state. A condenser temperature of 30°C and an average compressor efficiency of 65% was used.[7;8]

The interesting result is that without intercooling, or intercooling with flash gas, the COP exhibits a minimum rather than a maximum. The reason for this can be found

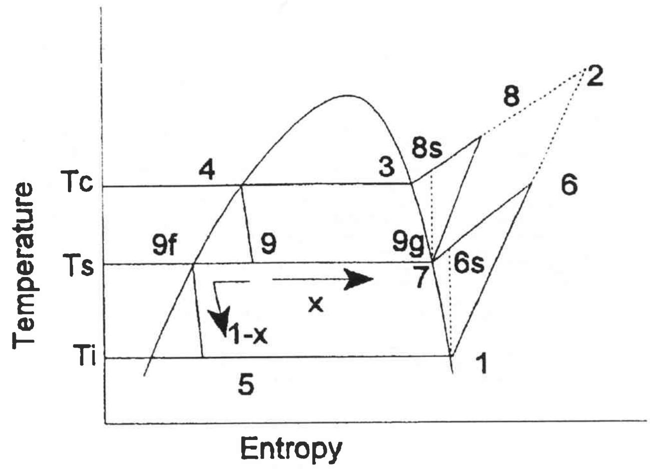

Figure 6 $T-s$ diagram for TSC system with economizer

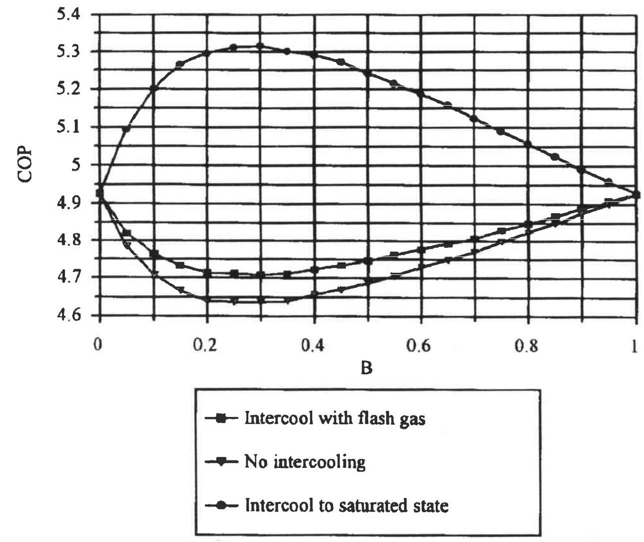

Figure 9 COP against intermediate pressure fraction for a two stage compression system with various inter-cooling strategies

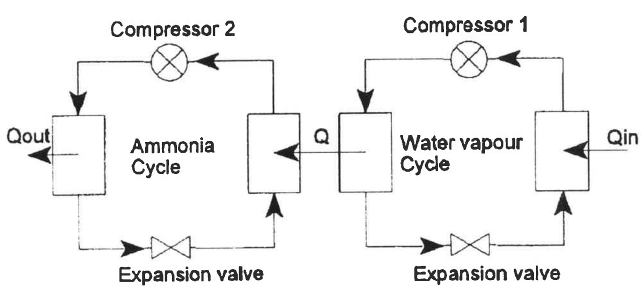

Figure 7 Hybrid system layout

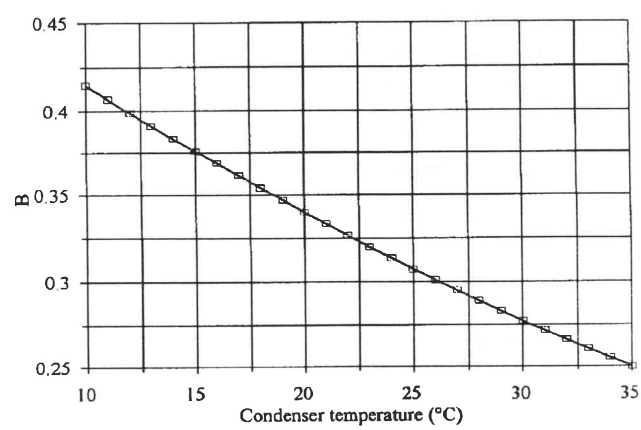

Figure 10 Optimum intermediate pressure fraction against condensert temperature for a two-stage compression system with equal compressor efficiencies

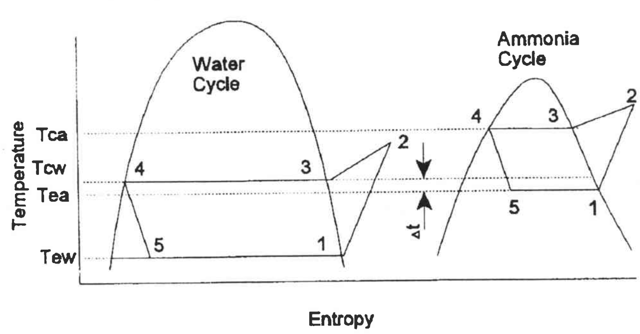

Figure 8 $T-s$ diagram for hybrid system in the definition of isentropic efficiency, and the shape of the isobars on the $T-s$ diagram for water. The isobars diverge in the superheated region. If the isentropic efficiency definition is applied twice, the resulting overall isentropic efficiency is lower than that of a single process. The lower efficiency leads to higher compressor power which explains the lower COP compared with single stage compression. This effect is only noticeable when very little or no intercooling is used.

Figure 9 shows that if intercoofing to the saturated state is used, the COP increases to a maximum value at intermediate pressure fraction of 0.3.

With the existence of an optimum intermediate pressure fraction proven, the Fibonacci search routine of Bunday & Garside[9] was used to calculate the optimum over a range of condenser temperatures, as shown in Figure 10.

The figure shows that the optimum intermediate pressure fraction varies nearly linearly from 0.41, at a condensing temperature of 10°C, to 0.25 at a condenser temperature of 35°C.

The optimum intermediate pressure fraction results in an equal compressor ratio over each stage. This is generally true if both stages have the same efficiency. We now investigate the optimum intermediate temperature for the hybrid system.

The optimum intermediate temperature for the hybrid system

Figure 11 shows the optimum intermediate temperature against condenser temperature for a hybrid system with the efficiency of the ammonia compressor at 80% while the water vapour compressor efficiency was varied.

It follows from the graph that as the efficiency of the water vapour compressor is reduced, the ammonia cycle has to do more of the work. The actual case where the ammonia compressor efficiency is approximately 80% and the water vapour compressor efficiency is approximately 65%[7] is shown by the bottom line in Figure 11. In this case, the ammonia cycle does all the work up to a condenser temperature of 26°C. This situation is not realistic because the water cycle must be used for ice generation. It does, however, show that the ammonia cycle can take over at the lowest practical attainable intermediate temperature. The added advantage of such a system is that the water vapour compressor ratio can be substantially decreased which implies a cheaper water vapour compressor.

It was therefore decided to use a value of 5°C for the intermediate temperature which gives a reasonable Δt from heat transfer considerations.

Temperature difference across the heat exchanger of the hybrid system

The COP of the hybrid system is influenced by the temperature difference across the heat exchanger between the condenser of the water side and the evaporator of the ammonia side, as shown in Figure 12.

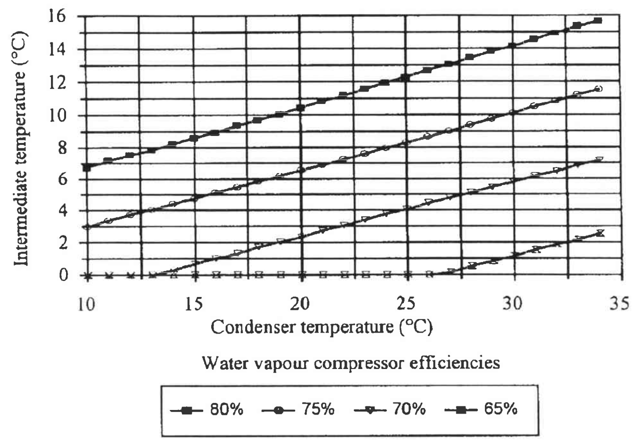

Figure 11 Optimum intermediate temperature against condenser temperature for a hybrid system with various water vapour compressor efficiencies and an ammonia compressor efficiency of 80%

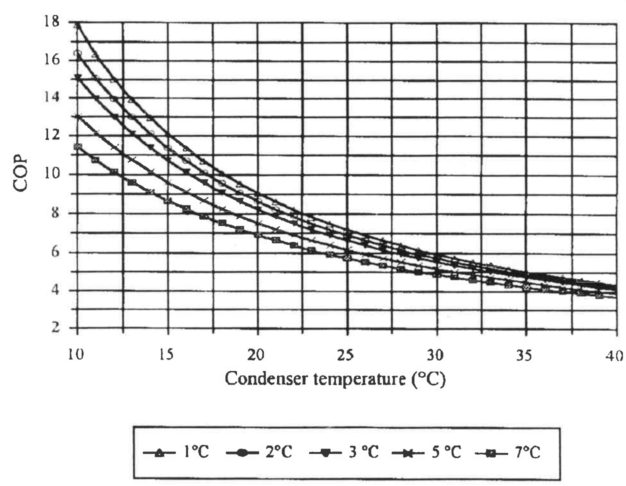

Figure 12 COP against condenser temperature for the hybrid system with various temperature differences over the heat exchanger

Figure 12 shows that the COP decreases with an increase in temperature difference. This is the result of the extra irreversibility when heat is transferred from the condenser of the water cycle to the evaporator of the ammonia cycle. It is assumed that a sufficiently large heat exchanger is available so that the temperature difference is 2°C. This value will be used in the following investigations.

Performance comparison

In this section the COP values of the four systems are compared as a function of condenser temperature as well as the specific cooling capacity. A compressor efficiency of 65% is used for the water vapour compressors and 80% for the ammonia compressors. Figure 13 shows the COP against condenser temperature for the different cycles.

The COP of all three water vapour systems is approximately the same, with the COP of the ammonia system approximately 28% higher. This is mainly due to the higher compressor efficiency of the ammonia cycle. With similar compressor efficiencies, the COP of the ammonia system is only slightly higher.

The single-stage compression system has the lowest COP of the water vapour systems, with the two-stage compression system slightly higher. The COP of the hybrid system is initially lower than that of the two-stage system, but increases relative to the other systems and intersects the COP of the two-stage compression system at a condenser temperature of 13°C.

The higher COP of the two-stage compression system is mainly due to the reduced work required by the two compressors, compared to a single compressor. The initial lower COP of the hybrid system is the result of the extra irreversibility when heat is transferred from the condenser of the water cycle to the evaporator of the ammonia cycle. This irreversibility is directly proportional to the temperature difference over the heat exchanger as is shown in Figure 13.

The heat transfer per unit volume flow at the compressor inlet, the specific cooling capacity, for the different systems is also compared. As these values vary very little with condenser temperature, they are only shown for a condenser temperature of 20°C. Table 1 shows the specific cooling capacity for the different systems.

Table 1 Specific cooling capacity for the different systems

System	Specific cooling capacity (kJ/m^3)
Single-stage compression	11.722
Two-stage compression	11.163
Hybrid system	10.391
Ammonia system	4010.6

As can be seen from the table, the specific cooling capacity for the ammonia cycle is approximately 350 times higher than that of water systems, which explains the much larger compressors required for the water vapour systems.

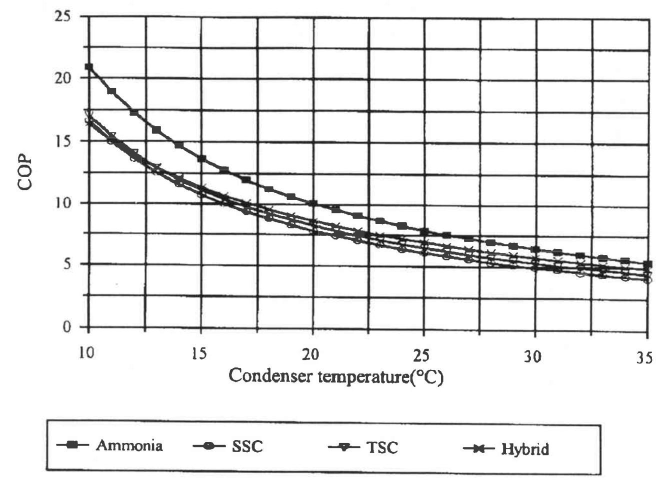

Figure 13 COP against condenser temperature for single compression, two-stage compression, hybrid and ammonia systems

The advantage of hybrid and two-stage compression systems is that the requirement of high pressure ratios at very high volumetric flow rates is relaxed thus decreasing the cost of the compressor.

Summary

In this paper simulation models were developed for three variations of the vacuum-ice system and for a conventional ammonia system. These models were used to compare the performance of the above systems. It was found that there is very little difference between the COP of the three water vapour systems. The single-stage compression system was found to be the least efficient vacuum-ice system. The hybrid system has the potential to be the most efficient water vapour system, provided that the temperature difference across the heat exchanger between the water cycle and ammonia cycle can be kept low. The two-stage compression system displayed a COP initially higher and then lower than that of the hybrid system. The conventional ammonia system displayed an higher overall COP which is largely due to the higher compressor efficiency.

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