

Comparison of the performances of absorption refrigeration cycles

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Abstract: This paper compares the performance of absorption refrigeration cycles that are used for refrigeration temperatures below 0°C. Since the most common vapor absorption refrigeration systems use ammonia-water solution with ammonia as the refrigerant and water as the absorbent, research has been devoted to improvement of the performance of ammonia-water absorption refrigeration systems in recent years. In this paper the performances of the ammonia-water and possible alternative cycles as ammonia-lithium nitrate, ammonia-sodium thiocyanate, monomethylamine-water, R22-DMEU, R32-DMEU, R124-DMEU, R152a-DMEU, R125-DMEU, R134a-DMEU, trifluoroethanol (TFE)-tetraethylglycol dimethylether (TEGDME), methanol-TEGDME and R134a-DMAC are compared in respect of the coefficient of performance (COP) and circulation ratio (f). The highest COP and the lowest f , were found as a function of the generator, condenser, absorber and evaporating temperature.

Key-Words: Absorption, refrigeration, working fluids, COP, circulation ratio, waste heat

1 Introduction

The European Union (EU) energy policy is an important part of the strategy for sustainable development. The ultimate objective of the strategy paper is the security of energy supply, encompassing environmental principles. The priority of the energy supply field lies in the reduction of global atmosphere warming with emphasis on efficient energy use and on renewable energy sources (RES) [1].

Simultaneous heating and cooling are required in many industries such as dairy plant pharmaceuticals chemical etc. Absorption systems have been extensively paid attention in recent years due to the potential for CFC and HCFC replacements in refrigeration, heating and cooling applications [2]. Furthermore, thanks to the progressive reduction of both installation and maintenance cost and energy consumption, their employment may become more and more diffuse [3].

Most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat for the purpose. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to a useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle [4]. Despite a lower coefficient of performance (COP) as compared to the vapor compression cycle,

absorption refrigeration systems are promising for using inexpensive waste energy from industrial processes, geothermal energy, solar energy etc.

Performance of an absorption refrigeration systems is critically dependent on the chemical and thermodynamic properties of the working fluid [4]. Thermodynamic properties of presented working fluids can be obtained from publications [5 - 14]. Evaluation of potential working fluid for the absorption cycle is a problem because of a lack of published thermodynamic data. The ideal absorbent-refrigerant pair does not exist, all possible combinations present advantages and disadvantages [13]. Many working fluids are suggested in literature but for the refrigeration temperatures below 0°C the most common working fluid is NH₃-H₂O. NH₃-H₂O system exhibits a relatively low COP, therefor efforts are being made to search for better refrigerant-absorbent pairs that can improve system performance [14]. Among different options of working fluids that can be used as alternative to NH₃-H₂O the following working fluids: NH₃-LiNO₃, NH₃-NaSCN, monomethylamine-water, R22-DMEU, R32-DMEU, R124-DMEU, R152a-DMEU, R125-DMEU, R134a-DMEU, TFE-TEGDME, methanol-TEGDME and R134a-DMAC are presented in this paper. As a result, COP was used to evaluate the performances of working fluids.

2 Cycles description

Figures 1 - 3 describes the cycles compared in this paper. Figure 1 illustrates the main components of the Single-effect absorption refrigeration cycle. High-pressure liquid refrigerant (2) from the condenser passes into the evaporator (4) through an expansion valve (3) that reduces the pressure of the refrigerant to the low pressure existing in the evaporator. The liquid refrigerant (3) vaporizes in the evaporator by absorbing heat from the material being cooled and the resulting low-pressure vapour (4) passes to the absorber, where it is absorbed by the strong solution coming from the generator (8) through an expansion valve (10), and forms the weak solution (5). The weak solution (5) is pumped to the generator pressure (7), and the refrigerant in it is boiled off in the generator. The remaining solution (8) flows back to the absorber and, thus, completes the cycle. In order to improve system performance, a solution heat exchanger is included in the cycle. If the cycle operates on $\text{NH}_3\text{-H}_2\text{O}$, an analyser and a rectifier need to be added to remove water vapour from the refrigerant mixture leaving the generator before reaching the condenser [14].

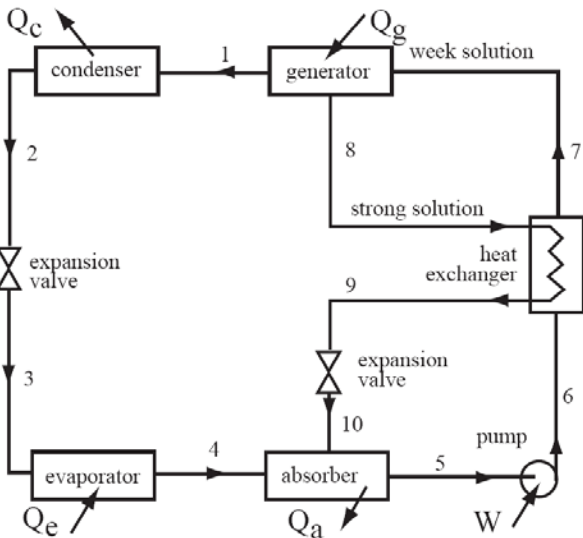


Fig. 1: The schematic of the single-effect absorption refrigeration cycle [14]

Figure 2 illustrates single-stage triple pressure level (TPL) absorption cycle.

Figure 3 illustrates double-lift and half-effect cycle respectively. A half-effect absorption cycle is a combination of two single-effect cycles but working at different pressure levels. Letting heat source temperature be lower than the minimum temperature is necessary for a single-effect cycle working at the same pressure level. The half-effect

absorption system was introduced for an application with a relatively low-temperature heat source. It must be noted that COP of the half-effect absorption system is relatively low as it rejects more heat than a single-effect absorption cycle around 50%. However, it can be operated with the relatively low temperature heat source [4].

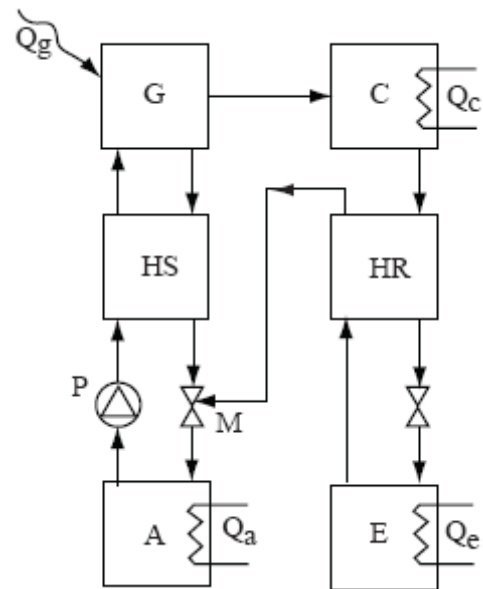


Fig. 2: Schematic illustration of a single-stage TPLAC absorption cycle. G-generator, A-absorber, C-condenser, E-evaporator, HR-refrigerant heat exchanger, HS-solution heat exchanger, P-solution pump, M-jet ejector [15]

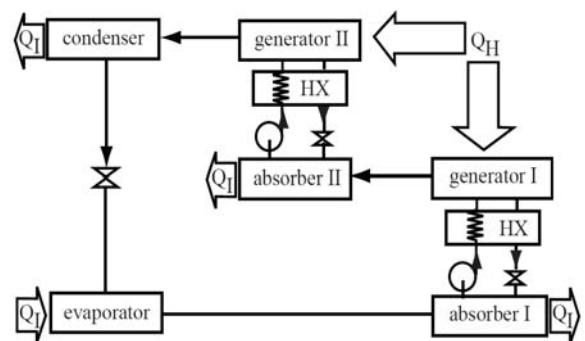


Fig. 3: A half-effect absorption cycle [4]

The COP of the single-effect absorption refrigeration cycle is defined as the ratio between the heat removed at the evaporator to the heat supplied to the generator (Eq. 1).

$$\text{COP} = \frac{\Phi_e}{\Phi_g} \quad (1)$$

The COP of the half-effect absorption refrigeration cycle is defined as the ratio between the heat removed at the evaporator to the heat supplied to the low generator and high generator (Eq. 2).

$$\text{COP} = \frac{\Phi_e}{\Phi_{lg} + \Phi_{hg}} \quad (2)$$

The circulation ratio (f) is defined as the ratio between the mass flow rate of the strong solution and the mass flow rate of the refrigerant (Eq. 3).

$$f = \frac{m_s}{m_r} \quad (3)$$

3 Comparison of absorption refrigeration cycles

Figure 4 shows the comparison of COP values vs. generator temperatures for $\text{NH}_3\text{-H}_2\text{O}$, $\text{NH}_3\text{-LiNO}_3$ and $\text{NH}_3\text{-NaSCN}$ absorption cycles. The COP values for these three cycles increase with generator temperatures. There exists a low generator temperature limit for each cycle. Each cycle cannot be operated at generator temperatures lower than its limit. For the $\text{NH}_3\text{-LiNO}_3$ cycle a lower generator temperature can be used than for the others. This is an important point for utilizing solar energy since fluid temperatures for flat plate solar collectors are generally below 90°C . It is shown that, for generator temperatures higher than 80°C , the $\text{NH}_3\text{-NaSCN}$ cycle gives the best performance, and the $\text{NH}_3\text{-H}_2\text{O}$ cycle has the lowest COP. However, the differences among them are not very remarkable. For low generator temperatures, the $\text{NH}_3\text{-LiNO}_3$ cycle gives the best performance [14].

Figure 5 shows the corresponding comparison of circulation ratios vs. generator temperatures. It is illustrated that the circulation ratio for the $\text{NH}_3\text{-NaSCN}$ cycle is higher than for the other two cycles. This means that either the solution pump needs to run faster or a bigger pump is required. If the generator temperature approaches its low temperature limit, the circulation ratio increases dramatically. Therefore, it is highly impractical to

operate a cycle at a generator temperature too low [14].

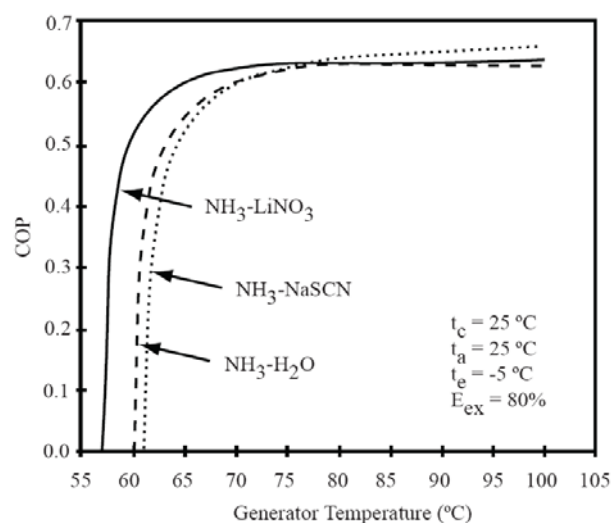


Fig. 4: Comparison of the effect of COP values on generator temperatures [14]

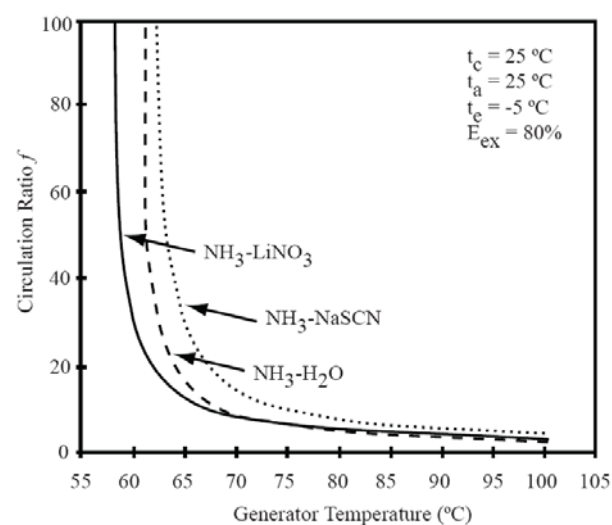


Fig. 5: Comparison of the effect of circulation ratio values on generator temperatures [14]

Figure 6 gives the comparison of COP values vs. evaporator temperatures for $\text{NH}_3\text{-H}_2\text{O}$, $\text{NH}_3\text{-LiNO}_3$ and $\text{NH}_3\text{-NaSCN}$ absorption cycles.

With the increase in evaporator temperature, the COP values for each cycle increase. For evaporator temperatures lower than zero, which is the temperature range for refrigeration, the $\text{NH}_3\text{-NaSCN}$ cycle gives the best performance, and the $\text{NH}_3\text{-H}_2\text{O}$ cycle has the lowest COP values. However, for high evaporator temperature, the

performance of the NH₃-H₂O cycle is better than that of the NH₃-LiNO₃ cycle [14].

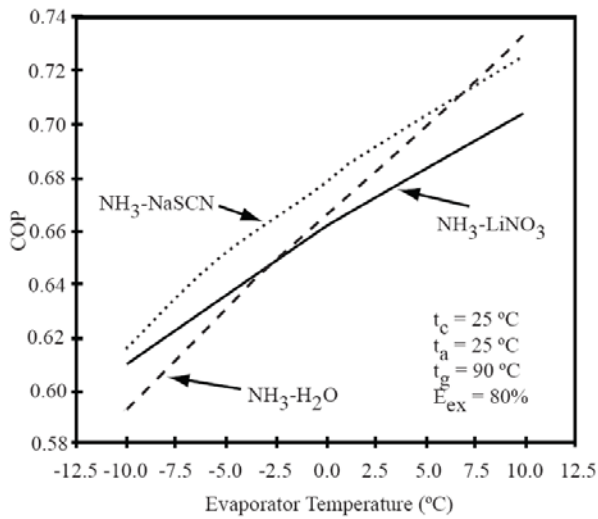


Fig. 6: Comparison of the effect of COP values on evaporator temperatures [14]

The corresponding comparison of circulation ratios vs. evaporator temperatures is given in figure 7. Again, it is shown that the circulation ratio for the NH₃-NaSCN cycle is higher than the other two cycles [14].

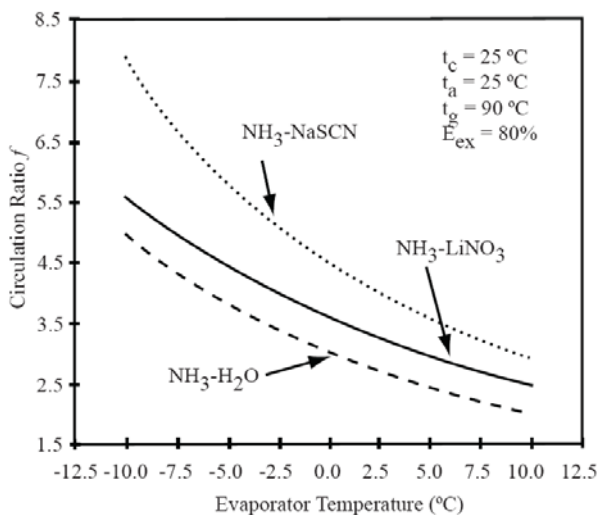


Fig. 7: Comparison of the effect of circulation ratio values on evaporator temperatures [14]

Figure 8 illustrates the comparison of COP values vs. condenser temperatures for NH₃-H₂O, NH₃-LiNO₃ and NH₃-NaSCN absorption cycles.

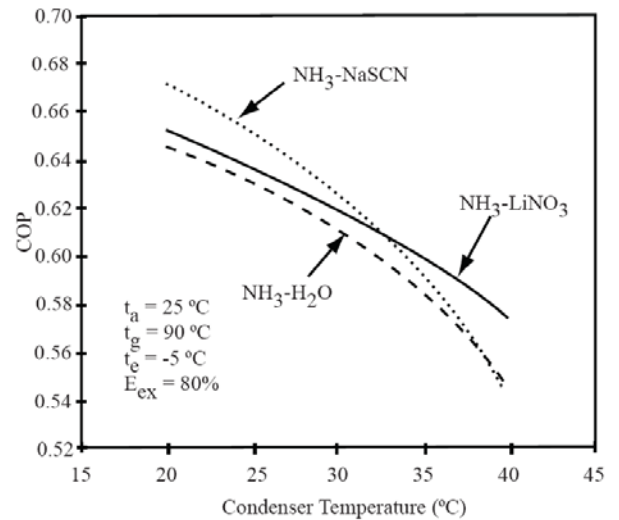


Fig. 8: Comparison of the effect of COP values on condenser temperatures [14]

Increasing condenser temperatures cause a decrease in system performance for each cycle. For condenser temperatures ranging from 20°C to 40°C, both the NH₃-NaSCN and NH₃-LiNO₃ cycles show better performance than the NH₃-H₂O cycle. Figure 8 shows that, for low condenser temperatures, the COP values for the NH₃-NaSCN cycle are the highest, while for high condenser temperatures, the NH₃-LiNO₃ cycle has the highest COP values [14].

Figure 9 illustrates the corresponding comparison of circulation ratios vs. condenser temperatures.

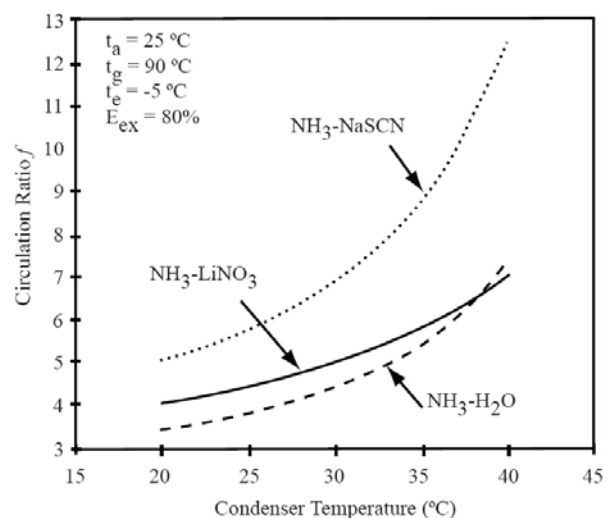


Fig. 9: Comparison of the effect of circulation ratio values on condenser temperatures [14]

The circulation ratio for the NH₃-NaSCN cycle is still higher than for the other two cycles [14]. The comparison of COP values vs. absorber temperatures for NH₃-H₂O, NH₃-LiNO₃ and NH₃-NaSCN absorption cycles is shown in figure 10.

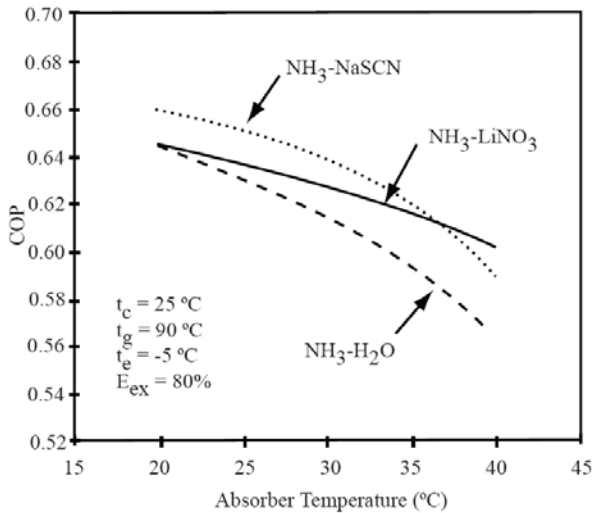


Fig. 10: Comparison of the effect of COP values on absorber temperatures [14]

The effect of absorber temperature is similar to that of condenser temperature. Generally speaking, the condenser and absorber temperatures should be at a similar level.

The corresponding comparison of circulation ratios vs. absorber temperatures is given in figure 11.

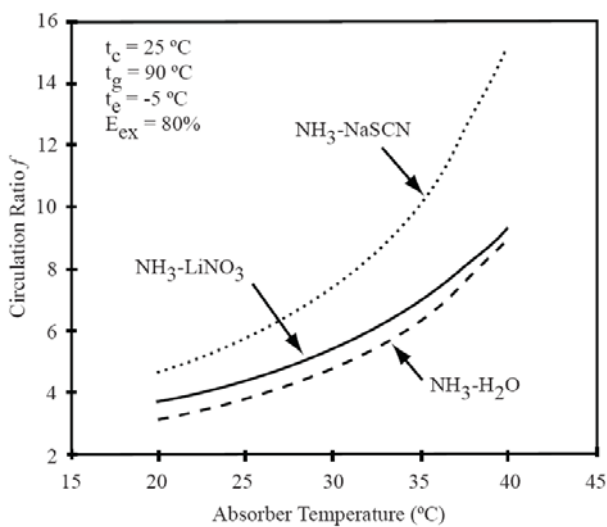


Fig. 11: Comparison of the effect of circulation ratio values on absorber temperatures [14]

Figures 4 - 11 shows that the system performance for the NH₃-NaSCN and NH₃-LiNO₃ cycles is better than that for the NH₃-H₂O cycle, however the improvement is not very remarkable. Considering the fact that, for the NH₃-NaSCN and NH₃-LiNO₃ cycles, no analysers and rectifiers are needed, these two cycles are suitable alternatives to the NH₃-H₂O cycle. The advantages for using the NH₃-NaSCN and NH₃-LiNO₃ cycles are very similar, however, for the NH₃-NaSCN cycle, it cannot operate below -10°C evaporator temperature because of the possibility of crystallization [14].

Figure 12 compares the performances of monomethylamine-water and ammonia-water working pair as a function of generator temperature at three different absorber and condenser temperatures (25, 30 and 35°C for monomethylamine-water and 30, 40°C for ammonia-water). It shows that the COP values increase sharply until a maximum value is reached and after that the value diminishes smoothly on increasing the generation temperature and it also diminishes on increasing the condensation and absorption temperatures. In the case of the ammonia-water solutions, the values of the COP are higher for generation temperatures above 80°C corresponding to $t_c = t_a = 30^\circ\text{C}$ and t_g above 97°C for temperatures of $t_c = t_a = 40^\circ\text{C}$. However, for generator temperatures between 60 and 83°C the monomethylamine-water solutions have high values of COP, which makes possible the use of sources of low enthalpy, as solar energy, industrial waste heat, among others [13].

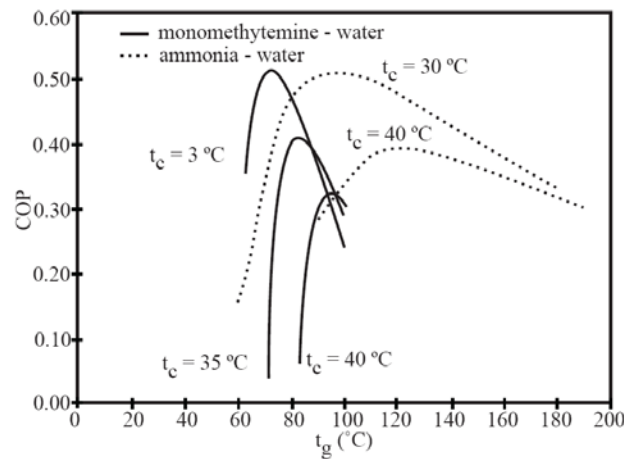


Fig. 12: Coefficient of performance for monomethylamine-water and ammonia-water [13]

The corresponding comparison of circulation ratio as function of evaporation temperature at three

different absorber and condenser temperatures (25°C, 30°C and 35°C) and a fixed generation temperature, $t_g = 70^\circ\text{C}$ is given in figure 13 [18].

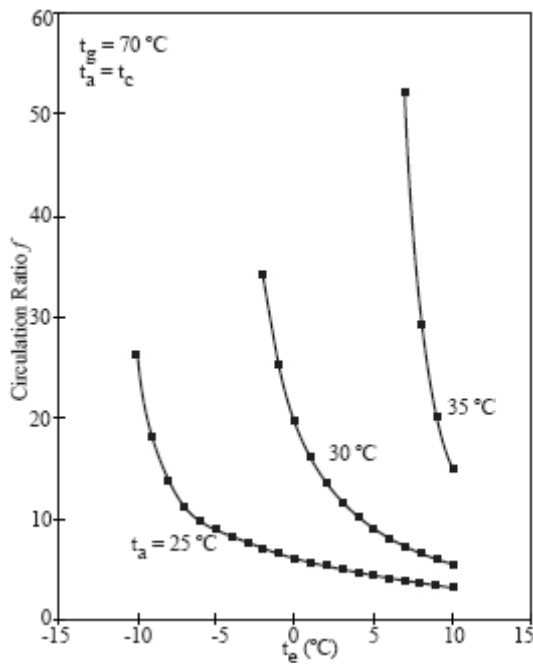


Fig. 13: Circulation ratio f for monomethylamine-water mixture as function of evaporation temperature at three different absorber and condenser temperatures (25°C, 30°C and 35°C) and a fixed generation temperature, $t_g = 70^\circ\text{C}$ [18]

It can be observed that the higher values of COP for the monomethylamine-water system is found in a short range of generation temperatures between 63 and 80°C, with COP values from 0.35 to 0.51, these are bigger than the corresponding ones in the ammonia-water system. The ammonia-water system has a higher COP at higher temperatures and it declines as well when the generation temperature increases. The monomethylamine-water system is a good potential pair for refrigeration cycles for absorption which can be operated at lower generation temperatures that allow the use of heat sources like solar, geothermal, industrial waste or others. An additional advantage of the monomethylamine-water system is the lower vapour required pressures. This capability would allow slighter devices to require smaller wall thickness in the components of the system. Due to the normal boiling point of the monomethylamine (-6°C) and to avoid vacuum operation problems,

this system can be used for air conditioning and product conservation purposes [13].

Figures 14 - 20 compares the performances of a single-stage triple pressure level (TPL) absorption cycle with four HFC refrigerants namely: R32, R125, R134a and R152a which are alternative to HCFC, such as R22 and R124, in combination with the absorbent dimethylethylenurea (DMEU) [15].

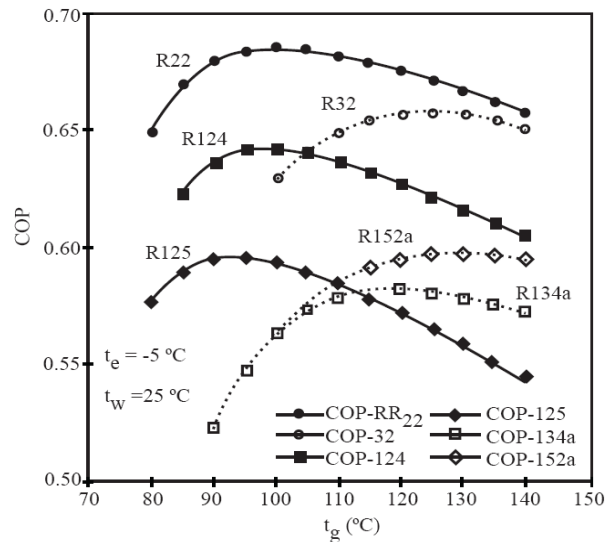


Fig. 14: Variation of the COP with generator temperature, t_g , for evaporator temperature of -5°C and cooling water temperature of 25°C [15]

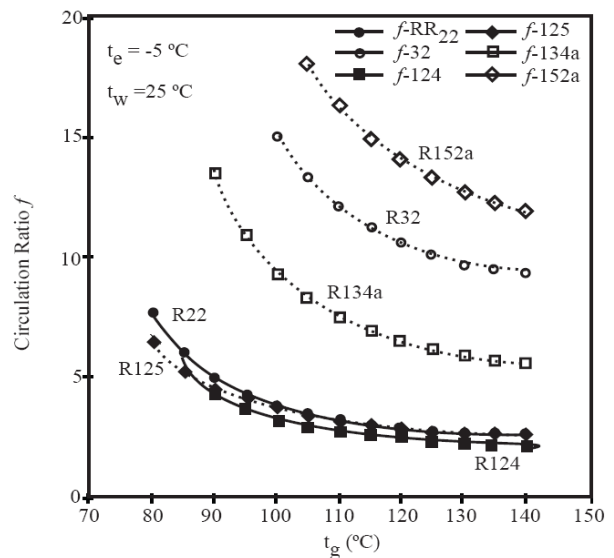


Fig. 15: Variation of the f with generator temperature, t_g , for evaporator temperature of -5°C and cooling water temperature of 25°C [15]

Variation of COP and f with evaporator temperature can be seen in figure 16 and 17.

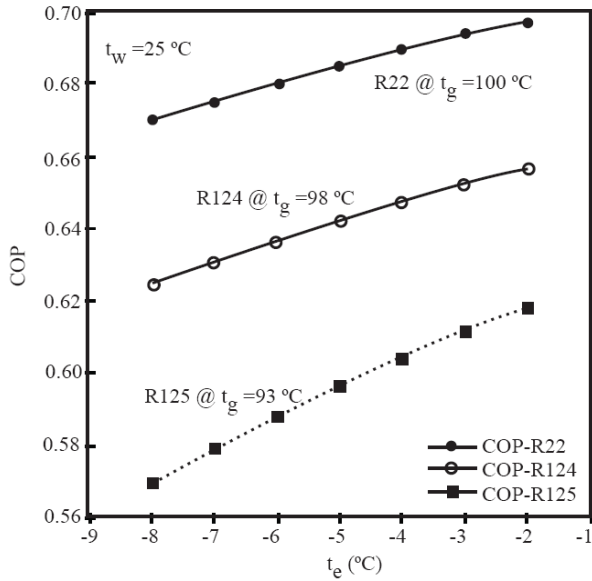


Fig. 16: Variation of the COP with evaporator temperature, t_e , for the cooling water temperature of 25°C and the generator temperature, t_g [15]

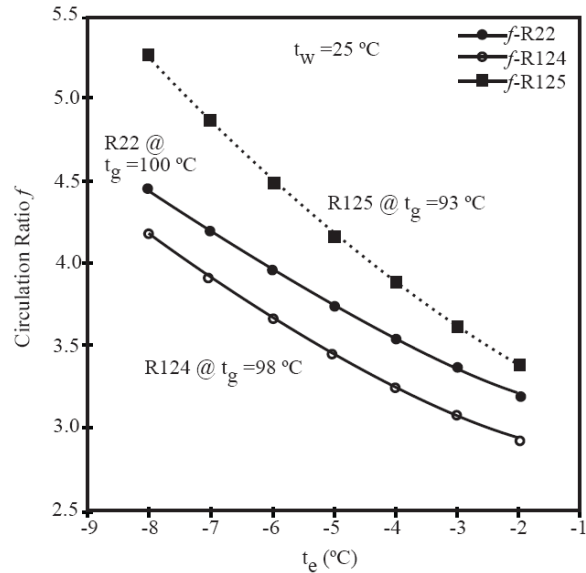


Fig. 17: Variation of the f with evaporator temperature, t_e , for the cooling water temperature of 25°C and the generator temperature, t_g [15]

Variation of the COP and f with cooling water temperature can be seen in figure 18 and 19.

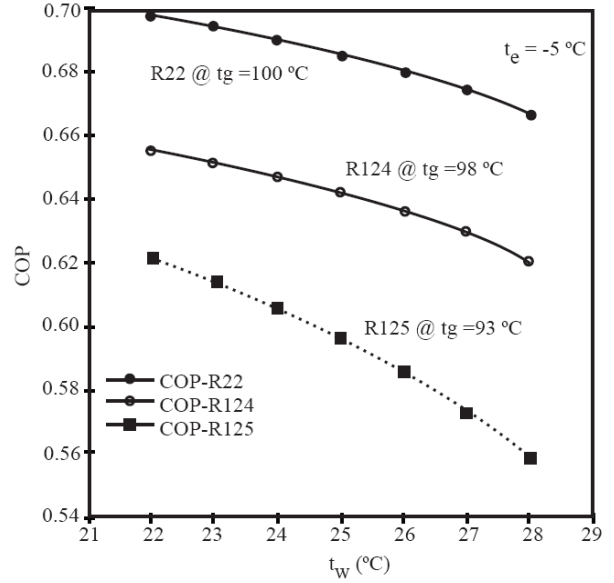


Fig. 18: Variation of the COP with cooling water temperature, t_w , for evaporator temperature of -5°C and the generator temperature, t_g [15]

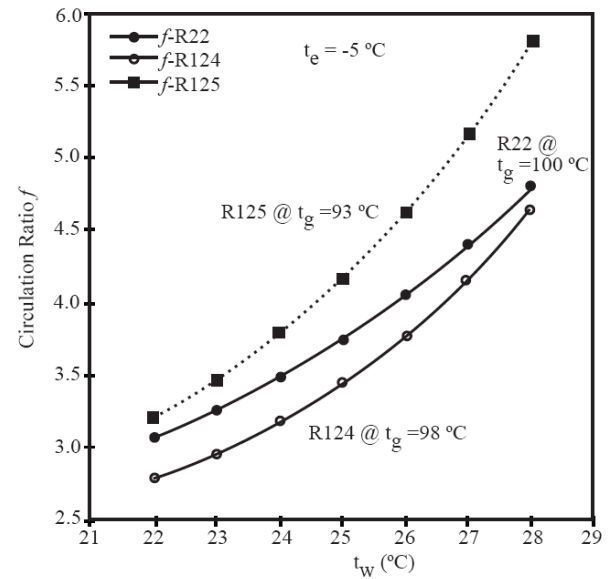


Fig. 19: Variation of the f with cooling water temperature, t_w , for evaporator temperature of -5°C and the generator temperature, t_g [15]

The range of operation of the TPLAC with the recommended working fluid R124-DMEU in term of generator temperature at maximum COP as a function of evaporator and cooling water temperatures are shown in figure 20.

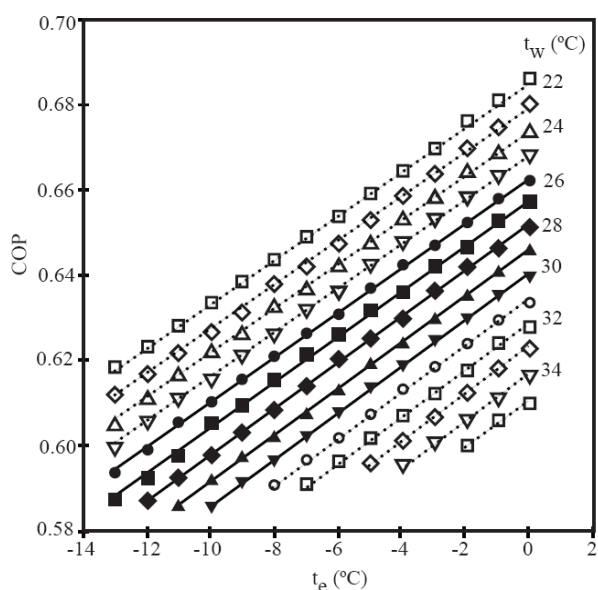


Fig. 20: The range of operation of the TPL absorption cycle with the recommended working fluid R124-DMEU in term of the maximum COP as a function of the evaporator temperature, t_e , and the cooling water temperature, t_w [15]

As can be seen in figure 14, the highest maximum of COP was achieved with the solution R22-DMEU followed by R32-DMEU, R124-DMEU, R152a-DMEU, R125-DMEU and R134a-DMEU. The lowest generator temperature at maximum COP was achieved by R125-DMEU followed by R124-DMEU, R22-DMEU, R134a-DMEU, R32-DMEU and R152a-DMEU. The lowest circulation ratio was achieved by R124-DMEU followed by R22-DMEU and R125-DMEU (which are very close), R134a-DMEU, R32-DMEU and R152a-DMEU. As can be seen in figure 14 there are two groups in terms of the generator temperature and f at maximum COP. The lower generator temperature obtained with the working fluids R22-DMEU, R124-DMEU and R125-DMEU which includes refrigerants from group 1 followed by the working fluids R32-DMEU, R134a-DMEU and R152a-DMEU which include refrigerants from group 2 and group 4. As can be seen in figure 14, the working fluids of group 1 shows much lower generator temperature and circulation ratio than those associated with group 2 and group 4. The solutions R22-DMEU followed by R124-DMEU and R125-DMEU matches the definition of the preferable working fluid. Among the HFC refrigerants, the solution R125-DMEU (group 1) is the preferable despite the solution R32-DMEU (group 2) that shows the second highest COP but at

much higher generator temperature and circulation ratio. The solutions R134a-DMEU and R152a-DMEU (group 2) showed the worst performances. Based on this analysis it can be said that R124-DMEU is the preferable pair among the compared working fluids in figures 14 - 20 while among working fluids based on HFC the preferable pair is the R125-DMEU. The figures 14 - 20 shows that maximum value of COP is obtained at different generator temperatures depending on the working fluid. The preferable working fluid can be considered as a solution with the highest COP, lower required generator temperature and f as low as possible. R124-DMEU is found to be the preferable pair among the compared working fluids in figures 14 - 20 while among working fluids based on HFC the preferable pair is the R125-DMEU [15].

Figure 21 shows the effect of the generator temperature on the COP for the vapour exchange double-lift cycle working with TFE-TEGDME, MeOH-TEGDME and ammonia-water. First of all, it can be observed that the vapour-exchange cycle can be driven by a low-temperature energy source (60 – 100°C) for refrigerating at 0°C. The cycle performances of the three mixtures shows that the working pair TFE-TEGDME has the highest COP (0.45) in the stable range. It is about 15% higher than that of MeOH-TEGDME and ammonia-water. MeOH-TEGDME also requires at least 80°C at the generators, whereas TFE-TEGDME and ammonia-water can operate at lower generator temperatures of about 65°C [16].

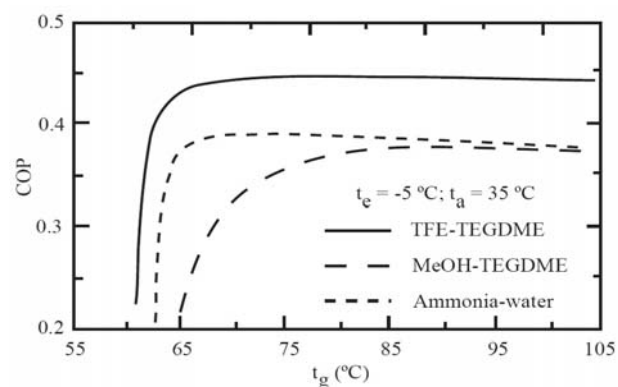


Fig. 21: Effect of generator temperature on COP for the double-lift vapour-exchange cycle [16]

The influence of the generator temperature on the circulation ratio per unit of refrigeration load ($f/Q_{I,E}$) can be seen in figure 22 for the vapour-exchange cycle. The working pair MeOH-TEGDME requires a considerably higher

circulation ratio because, in the selected operating conditions, the concentration difference between the rich and poor solutions is very low. The parameter ($f/Q_{l,E}$) is a minimum for ammonia-water due to the excellent vapour-liquid equilibrium of this working pair and ammonia's high latent-heat of vaporisation. In the stable COP zone, the ($f/Q_{l,E}$) of the TFE-TEGDME working pair is about three times higher than that of ammonia-water [16].

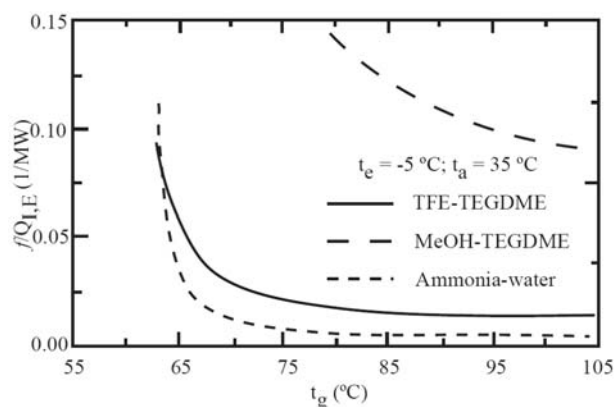


Fig. 22: Effect of generator temperature on circulation ratio per unit refrigeration load for the double-lift vapour-exchange cycle [16]

The performances of vapour-exchange absorption double-lift cycles working with the organic fluid mixtures TFE-TEGDME, MeOH-TEGDME and ammonia-water were analysed for refrigerating applications driven by low-grade thermal energy (70 – 100°C). The double-lift absorption cycles can operate in this range of heat source temperatures with a COP of about 0.45 for both TFE-TEGDME and MeOH-TEGDME, which is slightly higher than for the working pair ammonia-water. The COP of the vapour exchange double-lift cycle is better for the TFE-TEGDME fluid mixture with a minimum generator temperature of about 65°C. The COP of MeOH-TEGDME, however, is best in the series-flow double-lift cycle, but the circulation ratio per unit of refrigeration load is about twice, and the minimum generator temperature is 10°C higher, than with TFE-TEGDME. The vapour-exchange double-lift cycle using TFE-TEGDME as the working pair seems to be the most promising combination in terms of COP and the minimum generator temperature required for the operation of the cycle [16].

Figures 23 - 27 shows the performance of the half effect vapour absorption refrigeration cycle using R134a and DMAC. The working fluid pair

was evaluated for evaporator temperatures varying from -5°C to 10°C in steps of 5°C. The absorber outlet temperatures were varied from 25°C to 40°C in steps of 5°C. The condenser temperatures were varied from 20°C to 30°C in steps of 5°C. The generator temperatures were varied from 50°C to 70°C in steps of 10°C [17].

Figure 23 shows the variation in COP with both low and high absorber temperatures at different evaporator temperatures. It is observed that for given low and high absorber temperatures, the higher is the evaporator temperature, the higher is the COP obtained, as the temperature limit between which the heat is pumped is less at higher evaporator temperatures. For a given condenser, evaporator, generator and low absorber temperatures, an increase in the low absorber temperature results in a lower degassing width of the low pressure stage, which, in turn, results in a lower COP. This decrease is negligible at higher evaporating temperatures, but at lower evaporating temperatures, the decrease is appreciable. The reason is that at low evaporating temperature, the degassing width of the LP stage is more sensitive to the absorber temperatures. The same trend is obtained by keeping the temperature of the low absorber constant and varying the temperature of the high absorber, but the decrease is less when compared to the former condition [17].

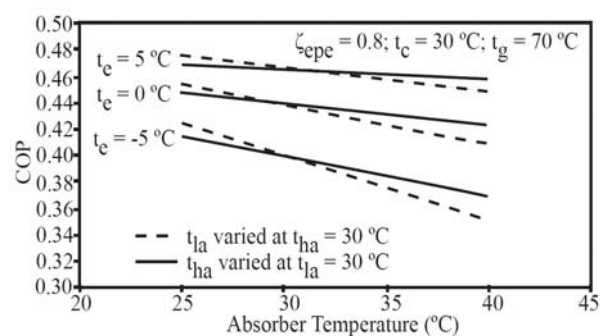


Fig. 23: Variation in COP with respect to absorber temperature at different evaporating temperatures [17]

Figure 24 shows the effect of the condensate pre-cooler on the performance of the cycle. Use of a condensate pre-cooler with an effectiveness of 0.8 is found to give an improvement of about 13.5% in COP for a R134a-DMAC cycle for different absorber temperatures when the temperatures of all the other components are kept constant. For the ammonia-water cycle, the improvement in COP is only 5.6% under the same conditions. The COP of the R134a-DMAC cycle is 25% higher than that of

the ammonia-water cycle without the condensate pre-cooler and 35% higher with the condensate pre-cooler. Further, half effect ammonia-water cycles are not feasible with low absorber temperatures above 30°C [17].

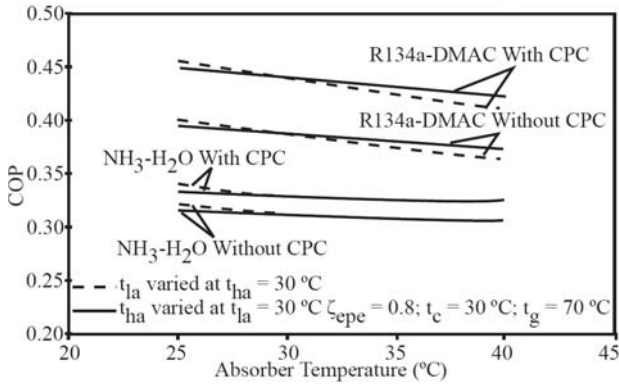


Fig. 24: Effect of condensate pre-cooler on COP [17]

Figure 25 shows the effect of condenser temperature on the performance of the cycle for both R134a-DMAC and ammonia-water at different high absorber temperatures. As the condenser temperature increases, the COP decreases as it increases the overall temperature of heat rejection. This decrease per degree is high at higher values of the high absorber temperatures. As discussed earlier, for a given condenser temperature, the higher is the absorber temperature, the lower is the COP obtained [17].

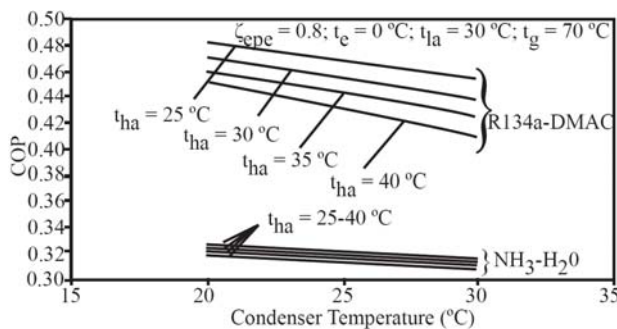


Fig. 25: Variation in COP with condenser temperature at different high absorber temperature [17]

Figure 26 shows the effect of generator temperature on the performance of the cycle for both R134a-DMAC and ammonia-water at different high absorber temperatures. At low generator temperature, the absorber temperature is found to be more significant, and its effect becomes negligible at high temperature. It can be concluded

that the ammonia-water system cannot be operated with source temperatures below 70°C under these operating conditions [17].

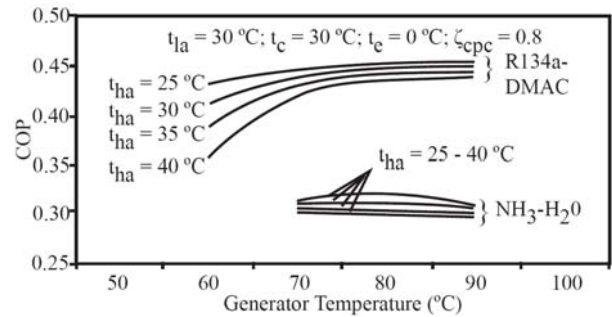


Fig. 26: Variation in COP with generator temperature at different high absorber temperature [17]

Figure 27 shows the variation in circulation ratio of the high and low pressure stages for different high and low absorber temperatures keeping the temperatures of all other components constant.

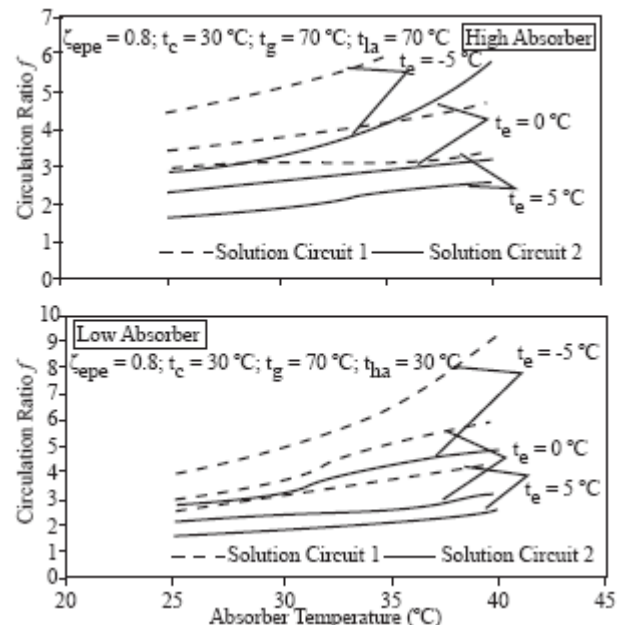


Fig. 27: Variation in circulation ratio with absorber temperature at different evaporating temperatures [17]

For a given high absorber temperature, the circulation ratio is high for the lower evaporator temperature as the degassing width decreases when the evaporator temperature is lowered. For a constant evaporating temperature when the high absorber temperature is increased, the circulation ratio of both stages increases, but above a certain

high absorber temperature, the increase in high circulation ratio becomes higher for solution circuit 2 than that of solution circuit 1. For a given low absorber temperature, the circulation ratio of solution circuit 1 is higher than that of solution circuit 2. Also, the increase in circulation ratio for a given increase in low absorber temperature is higher for solution circuit 1 than that for solution circuit 2. The reason for this is attributed to the relative p-t-x variation of the working fluid at high and low pressures. It can be concluded that from the circulation point of view, lowering the low absorber temperature is preferred relative to lowering the high absorber temperature [17]. Simulation studies on a half effect vapour absorption refrigeration cycle for solar energy based cold storage systems using R134a-DMAC as working fluids shows that the COP of this cycle is found to be about 0.35-0.46 for an evaporating temperature of -5 to 5°C with a heat input at 70°C with a condensing temperature of 20 – 30°C and absorber temperatures at 25°C. Improvement in COP up to 13% has been found with the use of a condensate pre-cooler. When compared to ammonia-water, R134a-DMAC gives a marginally higher COP in the half effect cycle at low heat source temperatures. An average increase in COP of 33% can be obtained within the evaporator temperature range of -5°C to 5°C for a typical operating condition. From these, it is evident that the R134a-DMAC refrigerant absorbent combination may be considered as one of the most favorable working fluids when a half effect system is to be operated with low temperature heat sources [17].

4 Conclusion

The performance of NH₃-H₂O, NH₃-LiNO₃, NH₃-NaSCN, monomethylamine-water, R22-DMEU, R32-DMEU, R124-DMEU, R152a-DMEU, R125-DMEU, R134a-DMEU, TFE-TEGDME, methanol-TEGDME and R134a-DMAC as working fluids for refrigeration temperature below 0°C were presented in this paper. The preferable working fluid can be considered as a solution with the highest COP, lower required generator temperature and circulation ratio as low as possible [15]. It is evident that COP strongly depends on working conditions such as generator, absorber, condenser and evaporating temperature.

Each cycle cannot be operated at generator temperatures lower than its limits. If the low-temperature heat source is used the half-effect absorption cycle gives the best performance.

Among presented working fluids it is evident that R124-DMEU, R125-DMEU, NH₃-LiNO₃ and NH₃-NaSCN are possible alternatives in terms of COP compared to NH₃-H₂O if single-effect cycle is used and R134a-DMAC and TFE-TEGDME respectively if half-effect cycle is used.

References:

- [1] Poberžnik S.; Goricanec D.; Krope J., Traditional vs. alternative energy house heating source, *Proceedings of the 2nd IASME / WSEAS International Conference on Energy & Environment (EE'07)*, Portoroz, Slovenia, May 15-17, 2007.
- [2] Mohideen S. T.; Renganarayanan S., Heat and mass transfer studies on 134 A-DMAC based falling film absorbers for absorption refrigeration system, *Proceedings of the 4th WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING and ENVIRONMENT*, Elounda, Greece, August 21- 23, 2006, pp. 342-350.
- [3] Rossi F.; Filipponi M.; Presciutti A., Ultrasonic vibrations for improving absorption chillers performances, *Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT*, Corfu, Greece, August 20 - 22, 2005, pp. 420-423.
- [4] Srihirin P.; Aphornratana S.; Chungpaibulpatana S., A review of absorption refrigeration technologies, *Renewable and Sustainable Energy Reviews*, 5, 2001, pp. 343-372.
- [5] Coronas A.; Vallrs M.; Chaudhari S. K.; Patil K. R., Absorption heat pump with the TFE-TEGDME and TFE-H₂O-TEGDME systems, *Applied Thermal Engineering*, 16, 1996, pp. 335-345.
- [6] Sencan A., Artificial intelligent methods for thermodynamic evaluation of ammonia-water refrigeration systems, *Energy Conversion and Management*, 47, 2006, pp. 3319-3332.
- [7] Lopez E. R.; Garcia J.; Coronas A.; Fernandez J., Experimental and predicted excess enthalpies of the working pairs (methanol or trifluoroethanol + polyglycol ethers) for absorption cycles, *Fluid Phase Equilibria*, 133, 1997, pp. 229-238.
- [8] Patek J.; Klomfar J., Simple functions for fast calculations of selected thermodynamic properties of the ammonia-water system,

- International Journal of Refrigeration*, 18, 1995, pp. 228-234.
- [9] Rogdakis E. D.; Antonopoulos K. A., Thermodynamic cycle, correlations and nomograph for $\text{NH}_3\text{-NaSCN}$ absorption refrigeration systems, *Heat Recovery Systems & CHP*, 15, 1995, pp. 591-599.
- [10] Pilatowsky I.; Rivera W.; Best R.; Holland F. A., Thermodynamic design data for absorption heat pump systems operating on monomethylamine-water. part 1: cooling, *Heat Recovery Systems & CHP*, 15, 1995, pp. 563-570.
- [11] Feng X.; Goswami D. Y., Thermodynamic properties of ammonia-water mixtures for power-cycle application, *Energy*, 24, 1999, pp. 525-536.
- [12] Borde I., Working fluids for an absorption system based on R124 (2-chloro-1,1,1,2-tetrafluoroethane) and organic absorbents, *International Journal of Refrigeration*, 20, 1997, pp. 256-266.
- [13] Romero R. J.; Guillen L.; Pilatowsky I., Monomethylamine-water vapour absorption refrigeration system, *Applied Thermal Engineering*, 25, 2005, pp. 867-876.
- [14] Sun D., Comparison of the performances of $\text{NH}_3\text{-H}_2\text{O}$, $\text{NH}_3\text{-LiNO}_3$ and $\text{NH}_3\text{-NaSCN}$ absorption refrigeration systems, *Energy Conversion and Management*, 39, 1998, pp. 357-368.
- [15] Jelinek M.; Levy A.; Borde I., The performance of a triple pressure level absorption cycle (TPLAC) with working fluids based on the absorbent DMEU and the refrigerants R22, R32, R124, R125, R134a and R152a, *Applied Thermal Engineering*, 28, 2008, pp. 1551-1555.
- [16] Medrano M.; Bourouis M.; Coronas A., Double-lift absorption refrigeration cycles driven by low-temperature heat sources using organic fluid mixtures as working pairs. *Applied Energy*, 68, 2001, pp. 173-185.
- [17] Arivazhagan S.; Murugesan S.N.; Saravanan R.; Renganarayanan S., Simulation studies on R134a-DMAC based half effect absorption cold storage systems, *Energy Conversion and Management*, 46, 2005, pp. 1703-1713.
- [18] Pilatowsky I.; Rivera W.; Romero R. J., Thermodynamic analysis of monomethylamine-water solutions in a single-stage solar absorption refrigeration cycle at low generator temperatures, *Solar Energy Materials & Solar Cells*, 70, 2001, pp. 287-300.