THE AQUA-AMMONIA ABSORPTION SYSTEM: AN ALTERNATIVE OPTION FOR FOOD REFRIGERATION

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ABSTRACT

Food refrigeration is one of the largest applications in the refrigeration market. At present, this market is dominated by mechanical vapor compression systems. The availability of aqua-ammonia absorption refrigeration systems provides an alternative to vapor compression systems. The study describes the aqua-ammonia absorption refrigeration system. A detailed mathematical model for simulating its operation is presented with complete sources of H_2O*-*NH_3 *mixture thermodynamic data. Based on the model, a computer software is developed. With this software, the performance characteristics of the aqua-ammonia absorption system are analyzed. Optimizing the system performance is also carried out and some optimum results are presented.*

INTRODUCTION

The largest overall application of refrigeration is the prevention or retardation of microbial, physiological, and chemical changes in food. During storage, foods may deteriorate through the growth of microorganisms, through changes caused by enzymes, or through chemical reactions. Holding foods at low temperatures can significantly reduce the rate at which these changes take place. The current food refrigeration market is largely dominated by the mechanical vapor compression machine. The alternative systems are absorption and ejector refrigeration machines (Eames *et al.* 1995; Sun *et al.* 1996; Sun 1996; 1997a,b).

Absorption refrigeration systems are currently attracting increasing interests. These systems are heat-powered with the potential in utilizing inexpensive waste

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heat, solar, biomass or geothermal energy sources. This waste energy is widely available and the cost of its supply is negligible in many cases. Therefore, the refrigerating costs will be low.

Heat-powered absorption systems also bring in two additional advantages. The dominant mechanical vapor-compression systems for several decades have depended on chlorofluorocarbon (CFC) and hydrochloroflurocarbon (HCFC) refrigerants. Growing international concern about damage caused by these refrigerants to the ozone layer has highlighted the need for development of new refrigerants. Companies including ICI and Du Pont have developed "ozone-friendly" hydrofluorocarbon (HFC) refrigerants such as HFC-134a. However, the deployment of these new refrigerants require the modifications of the existing systems with additional cost between 40% and 60% of a new machine (Butler 1994). Furthermore, production costs for new refrigerants are significantly high with an example of the cost of HFC-134a being about 3 to 4 times that of CFC-12. Absorption refrigeration systems use environmentally friendly refrigerant/absorbent combinations without causing any stratospheric ozone layer depletion. On the other hand, the dominant mechanical vapor compression systems are powered by electricity. The generation of electricity produces a large amount of carbon dioxide which is the main source for global warming. Therefore, heat-powered systems reduce demand for electricity and improve environmental conditions.

Lithium bromide-water and water-ammonia are the most common combinations used in absorption systems. In LiBr-H_2O systems, water is the refrigerant, they are not suitable for refrigeration temperature below 0C. H_2O-NH_3 systems use NH_3 as refrigerant, they can be applied to wide refrigeration temperature range. Therefore H_2O-NH_3 systems are suitable for food refrigeration and ice making.

In spite of the above advantages, research into the application of H_2O-NH_3 systems to the food industry is limited. The absorption system always suffers from its first cost. However, its operating cost is often less than vapor compression cycles because of low gas rates compared to electrical rates. This is especially true for the food processing situation where a relatively high-temperature waste heat sources is available, which can be used to drive the absorption system. For example, a water-ammonia system powered by wood or biomass waste was built to produce ice for fish storage (Monerasinghe 1985); solar energy, coupled with methane generated from cattle herd waste, as energy source to power an aqua-ammonia absorption system was also investigated for refrigeration at dairy farms in order to improve the rate and quality of milk production (Alvares and Trepp 1987), and finally a pilot plant for fishery industry powered by low quality waste heat from commercial manufacturing operations showed that operating costs were only 12% of those of a mechanical compression system (Xu 1983). Furthermore, H_2O-NH_3 absorption cooling systems can be used to replace the cooling systems using groundwater in the dairy industry (Stam 1994).

Simulation of an aqua-ammonia system is important for optimizing the system design and an efficient absorption system will enhance the opportunity for its application in the food industry. Early simulation work can be traced back to 1970s (Stoecker 1975), since then, various simulation studies have been carried out (Kaushik and Sheridan 1981; Alvares and Trepp 1987; Bulgan 1995). In these simulations, similar thermodynamic analyses were used and therefore, the accuracy of the simulation results mainly depended on the H_2O-NH_3 thermodynamic data used which were published more than ten years ago (Jain and Gable 1971; Shulz 1971; Ziegler and Trepp 1984). In the current study, the water-ammonia absorption refrigeration system is described, a computer simulation model is established based on the latest thermodynamic data available (Patek and Klomfar 1995) and used to analyze its performance characteristics. Hopefully, the paper will attract public interests in applying absorption refrigeration systems to food industry.

MATHEMATICAL MODEL

Figure 1 shows a schematic of the absorption refrigeration cycle. An absorption cycle employs a condenser, evaporator, generator, absorber, together with pressure-reducing valves (expansion valves) and a liquid pump. The heat exchangers shown in the figure are energy savers but are not essential to the successful operation of the cycle. The right half of Fig. 1 is identical with that of a regular mechanical vapor-compression cycle through which the refrigerant alone travels. At the end of this journey, the refrigerant at state 6 enters the absorber and is absorbed by the strong solution (strong ability in absorbing refrigerant) at state 12 coming back from the generator via the solution heat exchanger and the pressure-reducing valve. The dissolving process occurring in the absorber is exothermic; thus heat must be removed from the absorber in order to keep it at a constant low temperature. The weak solution (weak ability in absorbing refrigerant) emerging from the absorber at state 7 is pumped through the solution heat exchanger to extract some energy from the strong solution and then flows into the generator at state 9. In the generator heat is added to the weak solution to boil off some of the refrigerant. The remaining strong solution at state 10 flows back to the absorber. The refrigerant vapor leaving the generator at state 1 travels through the condenser and condenses into liquid state 2. The liquid refrigerant releases some of its heat to the vapor at state 5 from the evaporator via the liquid subcooler and then undergoes pressure reduction to state 4 via the expansion valve. It then evaporates in the evaporator to produce cooling effect. Finally the refrigerant vapor at state 6 returns to the absorber to complete the cycle.

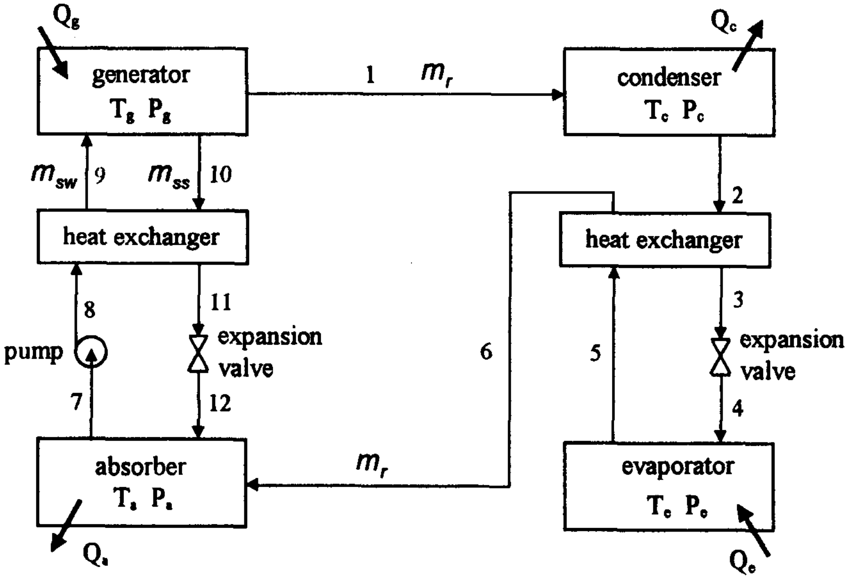

FIG. 1. THE SCHEMATIC OF THE AQUA-AMMONIA ABSORPTION REFRIGERATION CYCLE

The drawback of a water-ammonia system is that, since both ammonia and water are volatile, the vapor leaving the generator would contain too much water, and freezing of water in the expansion valve and evaporator would cause damage to the machine. To achieve efficient elimination of water vapor from the ammonia vapor being sent to the condenser, a rectifying column with a dephlegmator is placed between the generator and the condenser. The operation of the rectifier consists of the heat removal in the dephlegmator section to such a temperature as to cause condensation of water vapor. This condensate, while trickling downward through the trays in the column, joins the incoming weak solution and comes into intimate contact with the vapor rising from the generator. The ascending vapor develops a progressively higher ammonia concentration, while the descending liquid carries additional condensed water back to the generator. The rectifying process can produce vapor of any desired ammonia concentration, approaching very nearly 1.0 in the ideal case. Considering this fact, in the establishment of the mathematical model, the rectifying column will not be considered, and the ammonia concentration at the generator outlet is assumed to be a known constant value.

The operation of the absorption cycle is characterized by the generator, condenser, absorber and evaporator temperatures and the refrigerant mass flow flowing through the evaporator or the required refrigerating load. Therefore, the cycle can be analyzed as follows.

The operating pressures in the above components are determined by the saturated liquid and vapor concentrations in the condenser and evaporator, respectively and their temperatures, that is

$$P_c=P_g=P(T_2,X_2),\ P_e=P_a=P(T_5,Y_5) \tag{1}$$

For the strong solution at state (10) from the generator:

$$T_{10}=T_g,\ P_{10}=P_g,\ X_{10}=X(T_{10},P_{10}),\ h_{10}=h_l(T_{10},X_{10}) \tag{2}$$

$$T_1=T_g,\ P_1=P_g,\ h_1=h_v(T_1,Y_1) \tag{3}$$

For the weak solution at state (7) from the absorber:

$$T_7=T_a,\ P_7=P_a,\ X_7=X(T_7,P_7),\ h_7=(T_7,X_7) \tag{4}$$

A mass balance at the generator yields:

$$X_9=X_7,\ m_9X_9=m_{10}X_{10}+M_1Y_1,\ m_1=m_9-m_{10} \tag{5}$$

From Eq. (5), the mass flow rates of the strong and weak solutions at states (9) and (10), respectively are found to be:

$$m_{10}=\frac{X_9-Y_1}{X_{10}-X_9}m_1,\ m_7=m_9=\frac{X_{10}-Y_1}{X_{10}-X_9}m_1 \tag{6}$$

The weak solution from the absorber is pumped to the solution heat exchanger. As a result, the enthalpy at (8) is increased:

$$T_8-T_7,\ P_8=P_g,X_8=X_7,\ v_8=v_l(T_8,X_8),\ h_8=h_7+(P_8-P_7)v_8 \tag{7}$$

If the effectiveness of the solution heat exchanger is E_{ex}, the fluid at states (11) and (9) is derived by using the above values as follows:

$$T_{11}=E_{ex}T_8+(1-E_{ex})T_{10},\ X_{11}=X_{10},\ h_{11}=h_l(T_{11},T_{11}) \tag{8}$$

$$X_9=X_8,\ m_8=m_7,\ h_9=h_8+\frac{m_{10}}{m_8}(h_{10}-h_{11}),\ T_9=T_l(X_9,h_9) \tag{9}$$

Through the expansion valve, the fluid pressure is reduced from high pressure to low pressure, therefore

$$X_{12}=X_{11},\ h_{12}=h_{11},\ T_{12}=T_l(X_{12},h_{12}) \tag{10}$$

Liquid refrigerant at (2) from the condenser can be specified:

$$P_2=P_c,\ X_2=Y_1,\ T_2=T_l(P_2,X_2),\ h_2=h_l(T_2,X_2) \tag{11}$$

Vapor refrigerant at (5) from the evaporator can be specified:

$$P_5=P_e,T_5=T_e,Y_5=Y_1,h_5=h_v(T_5,Y_5) \tag{12}$$

The liquid refrigerant then exchanges energy with vapor refrigerant in the liquid subcooler. If the effectiveness of the subcooler is E_{sc}, the fluid in states 3 and 6 is derived as follows:

$$T_3=E_{sc}T_5+(1-E_{sc})T_2,\ X_3=X_2,\ h_3=h_l(T_3,X_3) \tag{13}$$

$$Y_6=Y_5,\ h_6=h_2+h_5-h_3,\ T_6=T_v(Y_6,h_6) \tag{14}$$

The liquid at (3) undergoes a pressure reduction via the expansion valve and enters the evaporator:

$$P_4=P_e,\ h_4=h_3,\ T_4=T_l(h_4,X_4) \tag{15}$$

In order to obtain the COP of the cycle, energy balances at the generator and evaporator are required, thus:

$$Q_g=m_1h_1+m_{10}h_{10}-m_9h_9,\ Q_e=m_1(h_5-h_4),\ W_{me}=(P_8-P_7)v_8 \tag{16}$$

The performance of the system is measured by the Coefficient of Performance (COP), it can be calculated as

$$COP=\frac{Q_e}{Q_g+W_{me}} \tag{17}$$

The circulation ratio of the system is defined as

$$f=\frac{m_7}{m_1} \tag{18}$$

The heat rejected by the absorber and condenser can be also found by taking the heat balances as:

$$Q_a=m_6h_6+m_{12}h_{12}-m_7h_7,\ Q_c=m_1(h_1-h_2) \tag{19}$$

THERMODYNAMIC PROPERTIES

The saturation pressure of the water-ammonia mixture as a function of saturation temperature is given by (Bourseau and Bugarel 1986):

$$\log P=A-\frac{B}{T} \tag{20}$$

where

$$A=7.44-1.767X+0.9823X^2+0.3627X^3 \tag{21}$$

$$B=2013.8-2155.7X+1540.9X^2-194.7X^3 \tag{22}$$

The original pressure unit in Eq. (20) was in Pascal, therefore the coefficients of A in Eq. (21) has been modified in the current study so that Eq. (20) gives pressure prediction in kPa.

The relations among pressure, temperature and concentration in saturated liquid phase and gas phase are found by the following fitted equations (Patek and Klomfar 1995):

$$T_l(P,\bar{X})=100\sum_{i=1}^{14} a_i(1-\bar{X})^{m_i}\left[\ln\left(\frac{2000}{P}\right)\right]^{n_i} \tag{23}$$

$$T_v(P,\bar{Y})=100\sum_{i=1}^{17} a_i(1-\bar{Y})^{m_i/4}\left[\ln\left(\frac{2000}{P}\right)\right]^{n_i} \tag{24}$$

The relationships among enthalpy, temperature and concentration in saturated liquid phase and gas phase are also found by the following fitted equations (Patek and Klomfar 1995):

$$h_l(T,\bar{X})=100\sum_{i=1}^{14} a_i\left(\frac{T}{273.16}-1\right)^{m_i}\bar{X}^{n_i} \tag{25}$$

$$h_v(T,\bar{Y})=1000\sum_{i=1}^{16} a_i\left(1-\frac{T}{324}\right)^{m_i}(1-\bar{Y})^{n_i/4} \tag{26}$$

All the coefficients in Eq. (23)-(26) are taken from the published paper (Patek and Klomfar 1995). X and Y in Eq. (23)-(26) are ammonia mole fraction in liquid and gas phase, respectively. The relationship between mole fraction and mass fraction is given by (Bogart 1981):

$$\bar{X}=\frac{X}{X+(1-X)\Phi} \tag{27}$$

where

$$\Phi=\frac{18.015}{17.03} \tag{28}$$

The specific volume of saturated water-ammonia solution is fitted by the author using a Numerical Algorithms Group (NAG) Fortran library routine E04FDF (Nag 1990). The source data were taken from ASHRAE handbook (ASHRAE 1993). The fitted equation is given as:

$$v(T,X)=\sum_{j=0}^{3}\sum_{i=0}^{3} a(i,j)(T-273.15)^i X^j \tag{29}$$

Table 1 lists the coefficients of Eq. (29). The standard error for the above equation is 4.058×10^{-6} and the mean deviation is 2.195×10^{-3}. This means that Eq. (29) provides a very accurate prediction.

TABLE 1.

COEFFICIENTS OF EQUATION (29)

i	j	a (i, j)	i	j	a (i, j)	i	j	a (i, j)	i	j	a (i, j)
0	0	9.9842×10^{-4}	0	1	3.5489×10^{-4}	0	2	-1.2006×10^{-4}	0	3	3.2426×10^{-4}
1	0	-7.8161×10^{-8}	1	1	5.2261×10^{-6}	1	2	-1.0567×10^{-5}	1	3	9.8890×10^{-6}
2	0	8.7601×10^{-9}	2	1	-8.4137×10^{-8}	2	2	2.4056×10^{-7}	2	3	-1.8715×10^{-7}
3	0	-3.9076×10^{-11}	3	1	6.4816×10^{-10}	3	2	-1.9851×10^{-9}	3	3	1.7727×10^{-9}

SIMULATION RESULTS

A computer simulation program has been developed based on the above analysis. Given the refrigerating load or refrigerant flow rate and operating temperatures at each component, the software gives prediction of the system performance, thermodynamic properties of inlet and exit fluids at each component. The thermodynamic properties were checked with published data and found that Eq. (20)-(29) gave accurate predictions.

All the simulation results were based on the following reference conditions: Y_1 = 99%, T_g = 100C, T_c = T_a = 28C, T_e = -5C, E_{ex} = 80% and E_{sc} = 80%. If the value of one parameter varies, all the other parameters remain unchanged.

Figure 2 shows the effect of generator temperature on system performance at different ammonia concentrations in refrigerant. Figure 2 shows that the performance characteristics of the aqua-ammonia absorption refrigeration system are totally different from those of a mechanical vapor compression system. This is caused by the vapor-liquid behavior of a binary mixture, with one as absorbent and the other as refrigerant. For a fixed ammonia concentration in refrigerant, which is controlled by the rectifying process, the system performance varies with generator temperature and there exists an optimum generator temperature. Since the weak solution concentration entering the generator is determined by the absorber conditions, and the condenser and generator are at a same pressure level, the optimum generator temperature determines the optimum strong solution concentration leaving the generator, and hence the optimum refrigerant vapor rate emerging from the generator. If the solution temperature at the generator is higher than its optimum value, some thermal energy is wasted, since the higher the temperature, the higher grade the thermal energy source is. However, if the solution temperature is lower than the optimum value, less refrigerant vapor is produced at the generator. Therefore, for absorption systems, the control of generator temperature is important.

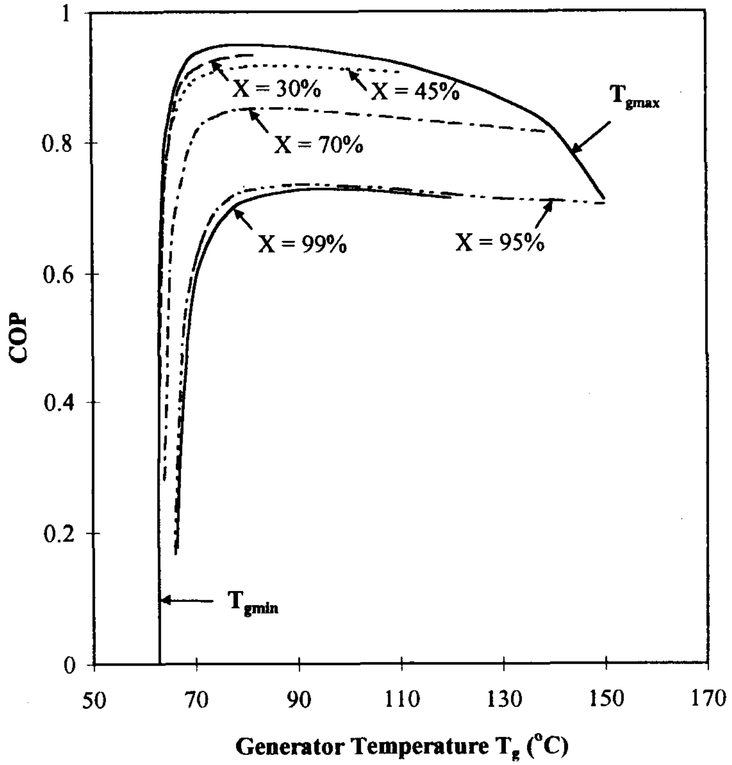

FIG. 2. THE EFFECT OF GENERATOR TEMPERATURE ON SYSTEM PERFORMANCE WITH VARIOUS AMMONIA CONCENTRATIONS IN THE REFRIGERANT

If the ammonia concentration in refrigerant is lowered by the proper control of the rectifying process, the system performance can be improved as shown in Fig. 2. This is caused by the increase in the refrigerant flow rate for the same flow rate of the weak solution entering the generator. However, this is not good practice. In contrast, refrigerant with as high as possible ammonia concentration is preferred since water vapor containing in the refrigerant may freeze along the pipes. Also when water finds its way to the evaporator, it elevates the evaporating temperature.

The generator possible operation region is enclosed by the generator temperature envelope as illustrated in Fig. 2. Operation with any generator temperature within the region can produce a refrigerating effect. The minimum and maximum permitted generator temperatures are also indicated in Fig. 2.

However, one needs to remember that the envelope curve may change if other running conditions vary.

Figure 3 illustrates the effect of generator temperature on circulation ratio given by Eq. (18). It is clearly shown that the refrigerant flow rate produced with lower ammonia concentration is higher than that with higher concentration since for the same weak solution flow rate, the higher the circulation ratio, the lower the refrigerant flow rate. When the generator temperature approaches its permitted minimum value shown in Fig. 2, the circulation ratio increases dramatically. This means that in order to maintain a constant refrigeration capacity, the sizes of the generator, absorber, solution heat exchanger, solution pump and expansion valve must be increased to cope with the extra solution flow load. Obviously, equipment capital cost will rise. For existing equipment,

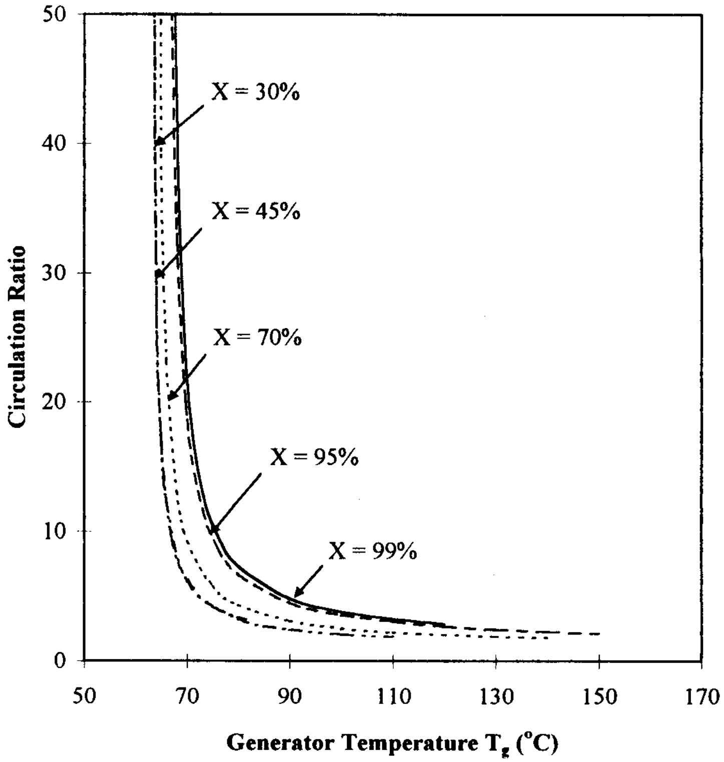

FIG. 3. THE EFFECT OF GENERATOR TEMPERATURE ON CIRCULATION RATIO WITH VARIOUS AMMONIA CONCENTRATIONS IN THE REFRIGERANT

if this is the case, the refrigeration capacity of the machine will drop accordingly. For generator temperature greater than its optimum value, the circulation ratio remains rather steady. Therefore it will be safer to operate the absorption machine with generator temperature greater, instead of smaller, than its optimum temperature if maintaining the optimum temperature is difficult to achieve.

Figure 4 shows the effect of condenser temperature on system performance. The effect is the same as that of a mechanical vapor compression machine, that is, with the increase in condenser temperature, the system performance decreases. Figure 4 also shows the effect of the solution heat exchanger and liquid subcooler. The solid line is the result of $E_{ex} = E_{sc} = 80\%$. The energy saving function of both components are clearly illustrated in Fig. 4. One of the major operating costs of the absorption system is the heat added in the generator and realistically there will be some cost associated with the removal of heat in

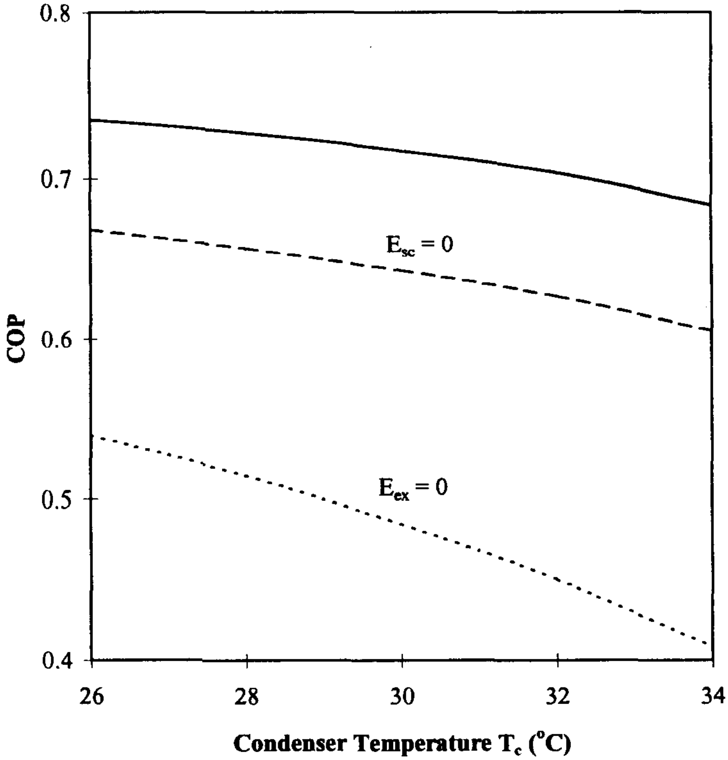

FIG. 4. THE EFFECT OF CONDENSER TEMPERATURE ON SYSTEM PERFORMANCE

the absorber. Obviously, the addition of a solution heat exchanger can reduce these costs. With the solution heat exchanger, the system performance can be improved by 36-67% for the conditions given in Fig. 4. In practice, a solution heat exchanger is always needed. The function of the liquid subcooler is similar to that of a solution heat exchanger, its improvement in system performance is not so dramatic as the solution heat exchanger, but still quite obvious. For the conditions in Fig. 4, adding a liquid subcooler can improve system performance by 10-13%. Of cause, there is cost for adding a liquid subcooler. Therefore, the adding of a liquid subcooler needs rational judgement between the actual cost and the improvement in system COP.

Figure 5 shows the effect of evaporator temperature on system performance. It is seen that the lower the refrigerating temperature, the lower the system COP. Generally speaking, the refrigerating temperature is determined by applications.

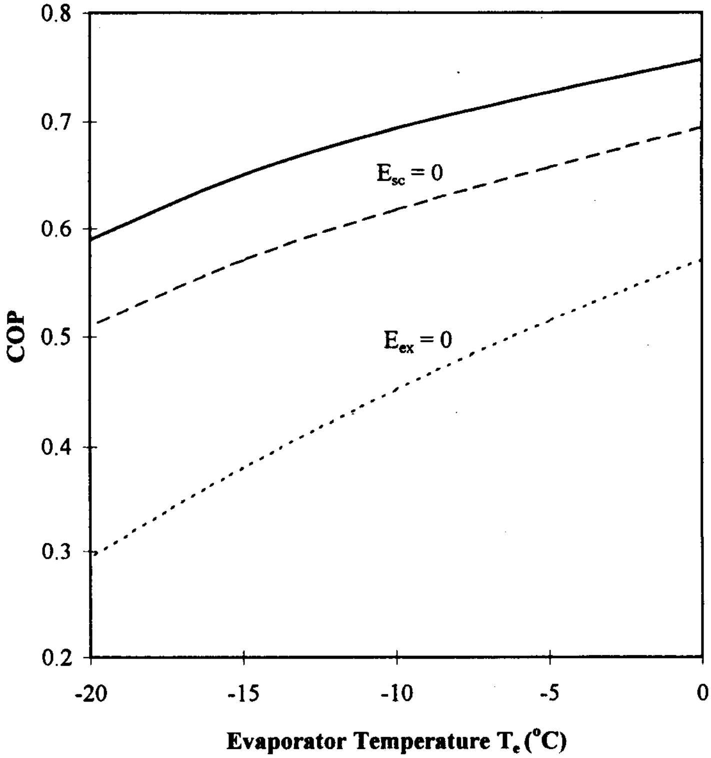

FIG. 5. THE EFFECT OF EVAPORATOR TEMPERATURE ON SYSTEM PERFORMANCE

Figure 2-5 show that COP for the aqua-ammonia absorption machine is in the region of 0.6-0.8. These values are much smaller than that of a mechanical vapor compression machine with typical value of 3. However, it is difficult to compare two systems with COP values since vapor compression cycle is a work-operated cycle because the elevation of pressure of the refrigerant is accomplished by a compressor that require work, and the absorption cycle is referred to as a heat-operated cycle as most of the operating cost is associated with providing the heat that drives off the vapor from the high pressure liquid. Work is a more efficient energy than heat, at the same time, it is much more expensive than heat. Bearing this in mind and considering using waste thermal energy as heat source, the overall costs for absorption machines should be low. This makes the aqua-ammonia absorption system a competitive option for food refrigeration.

CONCLUSIONS

In recent years, absorption refrigeration systems have attracted increasing interests since these systems can be powered by low-grade waste thermal energy. Aqua-ammonia absorption cycles can serve as alternative systems for mechanical vapor compression machines in food industry. In order to evaluate the performance of the water-ammonia absorption system, a simulation software was developed based on the thermodynamic analysis of the system. The simulation results show that operating conditions, especially generator temperature should be optimized to achieve optimum performance. The generator temperature envelope defines the possible permitted region for generator temperatures. The simulation results also show that a solution heat exchanger and liquid subcooler can greatly improve system COP. A solution heat exchanger is always needed for an absorption machine, the addition of a liquid subcooler may need to consider the cost of the subcooler since its improvement in system COP is much less pronounced as a solution heat exchanger. The optimum results presented in the paper can be used to select suitable operating conditions and the heat sources for water-ammonia absorption refrigerators.

NOMENCLATURE

E	effectiveness
h	enthalpy (kJ/kg)
m	mass flow rate (kg/s)
P	pressure (kPa)
Q	thermal energy (kW)
X	ammonia mass fraction in liquid phase

$\bar{X}$	ammonia mole fraction in liquid phase
T	temperature (K)
v	specific volume (m^3/kg)
Y	ammonia mass fraction in vapor phase
$\bar{Y}$	ammonia mole fraction in vapor phase

Subscripts

a	absorber
c	condenser
e	evaporator
ex	solution heat exchanger
g	generator
l	liquid phase of water-ammonia mixture
me	mechanical
sc	liquid subcooler
v	gas phase of water-ammonia mixture

REFERENCES

ALVARES, S.G. and TREPP, Ch. 1987. Simulation of a solar driven aqua-ammonia absorption refrigeration system. Part II: Viability for milk cooling at remote Brazilian dairy farms. Int. J. Refrig. *10*, 70–76.

ASHRAE 1993. *ASHRAE Handbook: Fundamentals*. Ch. 17, p17.81, ASHRAE, Atlanta, GA.

BOGART, M. 1981. *Ammonia Absorption Refrigeration in Industrial Processes*. Gulf Publ. Co., Houston, Texas.

BOURSEAU, P. and BUGAREL, R. 1986. Absorption-diffusion machines: comparison of the performances of NH_3-H_2O and NH_3-NaSCN. Int. J. Refrig. *9*, 206–214.

BULGAN, A.T. 1995. Optimisation of the theomodynamic model of aqua-ammonia absorption refrigeration systems. Energy Convers. Mgmt. *36*(2), 135–143.

BUTLER, D.J.G. 1994. Phase-out of CFCs and HCFCs: options for replacement and containment. In *CIBSE National Conference 1994*, pp. 181–186, CIBSE, London.

EAMES, I.W., APHORNRATANA, S. and SUN, D.-W. 1995. The jet pump cycle — A low cost refrigerator option powered by waste heat. Heat Recovery Systems & CHP. *15*, 711–721.

JAIN, P.C. and GABLE, G.K. 1971. Equilibrium property data equations for aqua-ammonia mixtures. ASHRAE Trans. *77*, 149.

KAUSHIK, S.C. and SHERIDAN, N.R. 1981. Computer modelling and theomodynamic assessment of an aqua-ammonia absorption cycle solar heat pump. Building and Environment *16*(3), 209–220.

MONERASINGHE, J. 1988. Biomass powered ice-maker for rural fishing village. *Proc. 2nd Asia Conf. on Technology for Rural Development* Dec. 4-7, 1985, Kuala Lumpur, Malaysia. pp. 567–581, World Scientific Pub. Co., Singapore.

NUMERICAL ALGORITHMS GROUP 1990. *Minimising or Maximising a Function*, NAG Fortran Library Manual Mark 14, Chap. E04, Vol. 3. NAG, Oxford OX2 8DR, U.K.

PATEK, J. and KLOMFAR, J. 1995. Simple functions for fast calculations of selected thermodynamic properties of the ammonia-water system. Int. J. Refrig. *18*(4), 228–234.

SHULZ, S.C.G. 1971. Equations of state for the system ammonia-water for use with computers. Commission II, I.I.R., Washington, D.C.

STAM, B.R. 1994. Heat pumps and absorption refrigeration equipment provide an alternative to groundwater. Voedingsmiddelen-Technologie *27*(24), 38–39.

STOECKER, W.F. 1975. Computer simulation of the performance of an aqua-ammonia absorption refrigeration system. *Proc. XIV International Congress of Refrigeration*, Moscow, *II*, 1060.

SUN, D.-W. 1996. Variable geometry ejectors and their applications in ejector refrigeration systems. Energy *21*(10), 919–929.

SUN, D.-W. 1997a. Solar powered combined ejector-vapour compression cycle for air conditioning and refrigeration. Energy Convers. Mgmt. *38*(5), 479–491.

SUN, D.-W. 1997b. Experimental investigation of the performance characteristics of a steam jet refrigeration system. Energy Sources *19*(4), 349–367.

SUN, D.-W. and EAMES, I.W. 1996. Performance characteristics of HCFC-123 ejector refrigeration cycles. Int. J. Energy Research *20*(10), 871–885.

SUN, D.-W., EAMES, I.W. and APHOMRATANA, S. 1996. Evaluation of a novel combined ejector-absorption refrigeration cycle. 1. Computer simulation. Int. J. Refrig. *19*(3), 172–180.

XU, S.Q. 1983. Utilisation of low value waste heat for an aqua-ammonia refrigeration pilot plant. Paper - American Society of Agricultural Engineers, No. 83-6542.

ZIEGLER, B. and TREPP, Ch. 1984. Equation of state for ammonia-water mixtures. Int. J. Refrig. *7*, 101–106.