MODELING AND PERFORMANCE ANALYSIS OF AMMONIA-WATER ABSORPTION REFRIGERATION SYSTEM FOR OCEAN-GOING FISHING VESSELS

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ABSTRACT
In this study, the modeling and performance analysis of ammonia-water absorption refrigeration system for ocean-going fishing vessels has been investigated. According to ammonia water solution saturated vapor and saturated liquid state equation, the thermodynamic properties of the refrigerant is studied. Based on the basic principles of the ammonia-water absorption refrigeration cycle, a mathematical model of the ammonia-water absorption refrigeration cycle is established so that the thermodynamic parameters of each state point and the heat load of each device can be calculated and then the coefficient of performance (COP), exeretic coefficient of performance (ECOP) for the system can also be calculated. Considering heat source temperature, cooing water temperature and refrigerating temperature, the effects of them on COP and ECOP are discussed by some simulation examples. After these calculations, some graphics indicating the change of the variables with the temperature parameters.

Keywords: ammonia-water absorption refrigeration system, ocean-going fishing vessels, modeling, exergy analysis

1. INTRODUCTION
For marine diesel engine, there is still about 50% of the fuel input energy not being put to productive use. The utilization of surplus heat rejected by the main propulsion engine provides substantial fuel savings, involving the reduction of pollutants. For the ocean-going fishing vessels, the catches' frozen preservation is very important, which related directly to the quality and economic value of the catches. Therefore, ammonia-water absorption refrigeration system has been paid an increased attention in which the surplus waste heat from the main engine is used to separate a refrigerant vapor from a binary solution. Sun[1] presented the thermodynamic design data and optimum design maps for absorption refrigeration systems. Ouadha[2] examined the feasibility of using waste heat from marine diesel engines to drive an ammonia-water absorption refrigeration system through a thermodynamic analysis. Sencan[3] analyzed thermodynamic evaluation of ammonia-water absorption refrigeration system using linear regression and M5' rules models within data mining process and artificial neural network model. Sozen [4] investigated the effect of heat exchangers on the system performance in an aqua-ammonia absorption refrigeration system. Wang [5]investigated of the feasibility of applying aqua-ammonia absorption refrigeration device to fishing vessels. Ni[6] established the thermodynamic mathematical model and analyzed the theoretical operation characteristics for fishing boat ammonia-water absorption refrigeration system. These researches are concerned with the thermodynamic analysis of ammonia-water absorption refrigeration system. Here, following the above mentioned studies, the paper investigates the modeling and performance analysis of ammonia-water absorption refrigeration system for the ocean-going fishing vessels.

2. THERMODYNAMIC ANALYSIS OF AMMONIA-WATER ABSORPTION REFRIGERATION SYSTEM
The ammonia-water absorption refrigeration system consists mainly of a generator, a condenser, an evaporator, an absorber, a solution heat exchanger and a circulation pump. The schematic illustration of the ammonia-water ARS is presented in Fig.1. The generator uses waste heat from the marine Diesel engine to separate ammonia vapour from the concentrated ammonia solution. When ammonia is evaporated off the generator, it contains some water vapour. To remove as much water vapour as possible, the vapour driven off at the generator first flows counter currently to the incoming solution in the rectification column. In the condenser, high pressure ammonia vapours are cooled and condensed to liquid state. Liquid ammonia leaves the condenser and flows to the evaporator through an expansion valve. The refrigerant then enters the evaporator, where it receives heat from the cold source. Then, ammonia vapour enters the absorber, where a weak solution of water and low concentration ammonia absorbs the refrigerant and, at the same time, releases heat to the neighbourhood. The ammonia-water solution

flows back to the generator through a circulation pump to undergo a new cycle.

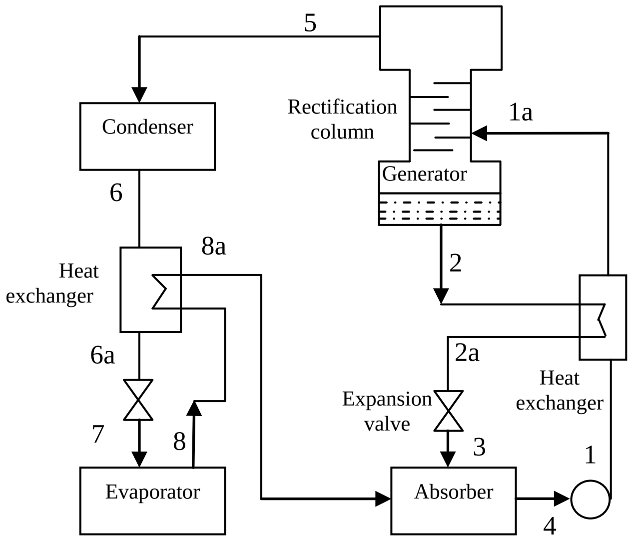

Figure 1: Schematic Diagram of an Ammonia-water Absorption Refrigeration System

2.1. Thermal-physical Properties of Ammonia-water Mixtures

For modeling and simulating an ammonia-water absorption refrigeration system, the thermal-physical properties of the ammonia-water have to be known as analytic function. Based on the state equation proposed by Schulz[7], using the thermodynamic relations, the expressions of thermodynamic parameters for temperature T, pressure P, enthalpy H, mass fraction x and y can be derived by mathematical transformation[8] .

The enthalpy H_R^{gM} in vapor phase of ammonia-water mixture can be expressed:

$$\begin{aligned} H_R^{gM} &= (1-y)H_R^{gH_2O} + yH_R^{gNH_3} \\ &= (1-y)[G_R^{gH_2O} - \theta(\partial G_R^{gH_2O}/\partial\theta)_\pi] + \\ &\quad y[G_R^{gNH_3} - \theta(\partial G_R^{gNH_3}/\partial\theta)_\pi] \end{aligned} \quad (1)$$

Where, $G_R = \dfrac{G_m}{R_m T_b}$ is contrast-state Gibbs free energy function, $\theta = \dfrac{T}{T_b}$ contrast-state temperature, $\pi = \dfrac{P}{P_b}$ contrast-state pressure. Superscript g is vapor phase, M mixture, e surplus. Subscript R is the contrast value.

The enthalpy H_R^{fM} in liquid phase of ammonia-water mixture can be defined as:

$$\begin{aligned} H_R^{fM} &= (1-x)H_R^{fH_2O} + xH_R^{fNH_3} + H_R^e \\ &= (1-x)[G_R^{fH_2O} - \theta(\partial G_R^{fH_2O}/\partial\theta)_\pi] + \\ &\quad x[G_R^{fNH_3} - \theta(\partial G_R^{fNH_3}/\partial\theta)_\pi] + G_R^e - \theta(\partial G_R^e/\partial\theta)_{\pi,x} \end{aligned} \quad (2)$$

Where, superscript f is liquid phase.

The relationship of T, P and x for vapor-liquid equilibrium is written as follows:

$$\begin{aligned} &\exp\left\{\frac{1}{\theta}\left[G_R^{fH_2O} + \theta\ln(1-x) + G_R^e - x(\partial G_R^e/\partial\theta)_{\pi,x} - G_R^{gH_2O}\right]\right\} + \\ &\exp\left\{\frac{1}{\theta}\left[G_R^{fNH_3} + \theta\ln x + G_R^e + (1-x)(\partial G_R^e/\partial\theta)_{\pi,x} - G_R^{gNH_3}\right]\right\} = 1 \end{aligned} \quad (3)$$

The relationship of T, P and y for vapor-liquid equilibrium is written as follows:

$$y = \exp\left\{\frac{1}{\theta}\left[G_R^{fNH_3} + \theta\ln x + G_R^e + (1-x)(\partial G_R^e/\partial\theta)_{\pi,x} - G_R^{gNH_3}\right]\right\} \quad (4)$$

The liquid equilibrium equation of pure ammonia is:

$$G_R^{fNH_3} - G_R^{gNH_3} = 1 \quad (5)$$

2.2. State Parameters Calculation of Ammonia-water Absorption Refrigeration System

Here, assuming the heat source temperature T_h, the cooling water temperature T_w and the refrigerating temperature T_e are known.

- point 6

The condensing temperature T_6 can be expressed:

$$T_6 = T_w + \Delta T_w + \Delta T_6 \quad (6)$$

Where, ΔT_w is the temperature rise of cooling water in the condenser, ΔT_6 the temperature difference between the hot side of the condenser.

The ammonia concentration $\xi_6 = 0.998$ and pressure P_6, enthalpy h_6 can be calculated by Eq.(3) and Eq.(2).

- point 2

The termination temperature of the generator T_2 can be expressed:

$$T_2 = T_h - \Delta T_2 \quad (7)$$

Where, ΔT_2 is the temperature difference between the hot side of the generator.

The pressure P_2 is given by the following expression:

$$P_2 = P_6 + \Delta P_2 \quad (8)$$

Where, ΔP_2 is the pipeline pressure drop from generator to condenser.

Then, the ammonia concentration ξ_2 and enthalpy h_2 can be calculated by Eq.(3) and Eq.(2).

- point 8

The termination temperature of the evaporator T_8 can be expressed:

$$T_8 = T_e - \Delta T_8 \tag{9}$$

Where, ΔT_8 is the temperature difference of heat transfer in the evaporator.

The pressure P_8 can be calculated by Eq.(5) based on the initial temperature of the evaporator T_8^* and temperature T_8^* is defined as:

$$T_8^* = T_8 - \Delta T_8^* \tag{10}$$

Where, ΔT_8^* is the temperature rise of the evaporator.

The enthalpy h_8 can be calculated by the following expression:

$$h_8 = h_8' + (h_8'' - h_8')\frac{0.998 - \xi_8'}{1 - \xi_8'} \tag{11}$$

Where, h_8' is the termination enthalpy in liquid phase of the evaporator, h_8'' the termination enthalpy in vapour phase of the evaporator, ξ_8' the termination ammonia concentration in liquid phase of evaporator.

- point 4

The termination temperature of the absorber T_4 can be expressed:

$$T_4 = T_w + \Delta T_4 \tag{12}$$

Where, ΔT_4 is the the temperature difference between the cold side of the absorber.

The pressure P_4 can be expressed:

$$P_4 = P_8 - \Delta P_4 \tag{13}$$

Where, ΔP_4 is the pipeline pressure drop from evaporator to absorber.

Then, the ammonia concentration ξ_4 and enthalpy h_4 can be calculated by Eq.(3) and Eq.(2).

- point 3

The outlet temperature of the solution heat exchanger T_3 can be expressed:

$$T_3 = T_4 + \Delta T_3 \tag{14}$$

Where, ΔT_3 is the the temperature difference between the cold side of the solution heat exchanger.

Based on the temperature T_3, ammonia concentration ξ_3 (= ξ_1), pressure P_3 and enthalpy h_3 can be calculated by Eq.(3) and Eq.(2).

- point 1

Based on the thermal equilibrium of solution heat exchanger, the outlet enthalpy of concentrated ammonia solution h_{1a} can be calculated:

$$h_{1a} = h_4 + (h_2 - h_3)\frac{f-1}{f}0.95 \tag{15}$$

Where, f is the solution circulation ratio.

The outlet temperature of concentrated ammonia solution T_{1a} can be calculated:

$$T_{1a} = T_4 + (T_2 - T_3)\frac{f-1}{f} \tag{16}$$

- point 5

The top ammonia concentration of rectification column ξ_5 should arrive 0.998. Based on pressure P_5 (= P_2) and ammonia concentration ξ_5, the temperature T_5 and enthalpy h_5 can be calculated by Eq.(3) and Eq.(1).

- point 8a, 6a

Based on the thermal equilibrium of supercooler, the outlet enthalpy in vapour phase of the supercooler h_{8a} can be calculated:

$$h_{8a} = h_8 + (h_8 - h_6)\upsilon \tag{17}$$

Where, υ is the load factor of supercooler.

the outlet temperature in vapour phase of the supercooler T_{8a} can be calculated:

$$T_{8a} = \frac{(h_{8a} - h_8)}{c''} + \mathrm{T}_8 \tag{18}$$

Where, c'' is the average specific heat of ammonia between T_8 and T_{8a}.

The outlet enthalpy h_{6a}, temperature T_{6a} in liquid phase of the supercooler can be expressed:

$$h_{6a} = h_6 + (h_{8a} - h_8)$$
$$T_{6a} = T_6 - \frac{(h_{8a} - h_8)}{c'} \tag{19}$$

Where, c' is the average specific heat of ammonia solution between T_6 and T_{6a}.

3. MODELING OF AMMONIA-WATER ABSORPTION REFRIGERATION SYSTEM

In the first law analysis, The cycle performance of ammonia-water absorption refrigeration system is measured by the coefficient of performance (COP). The larger COP is, the less energy will be consumed by the power unit in the same amount of refrigerating capacity.

In the second law analysis, the exergy is calculated for a system instead of energy and the difference between the energy qualities is also taken into account. The effectiveness of the ammonia-water absorption refrigeration system depends on the exeretic coefficient of performance (ECOP).

According to the mass conservation, the first and second laws of thermodynamics, the energy balance equations and exergy loss equations for some components of the ammonia-water absorption refrigeration system are expressed as follows:

Condenser:

$$q_C = h_5 - h_6$$
$$\Delta e_C = e_5 - e_6 \tag{20}$$

Evaporator:

$$q_E = h_{8a} - h_6$$
$$\Delta e_E = e_6 - e_{8a} + q_E(T_0 / T_e - 1) \tag{21}$$

Absorber:

$$q_A = h_{8a} - h_3 + f(h_3 - h_4)$$
$$\Delta e_A = e_{8a} - e_3 + f(e_3 - e_4) \tag{22}$$

Generator :

$$q_G = h_5 - h_2 + f(h_2 - h_{1a}) + q_R$$
$$\Delta e_G = f(e_{1a} - e_2) + e_2 - e_e + q_G(1 - T_0 / T_h) \tag{23}$$

Where, T_0 is the reference temperature. q is the unit thermal load, e the unit exergy. Subscripts A , C , E and G are absorber, condenser, evaporator and generator, respectively. q_R the unit thermal load of reflux condenser.

Then, ignoring the work required for the pumping process, COP is defined as:

$$COP = \frac{q_E}{q_G} = \frac{h_{8a} - h_6}{h_5 - h_2 + f(h_2 - h_{1a}) + q_R} \tag{24}$$

ECOP is defined as:

$$ECOP = \frac{e_E}{e_G} = COP \frac{\frac{T_0}{T_e} - 1}{1 - \frac{T_0}{T_g}} \tag{25}$$

4. PERFORMANCE ANALYSIS OF AMMONIA-WATER ABSORPTION REFRIGERATION SYSTEM

Based on the modeling of ammonia-water absorption refrigeration system, the effects of heat source temperature T_h , cooing water temperature T_w and refrigerating temperature T_e on the performance of it are discussed. The relevant temperature and pressure parameters are listed in Tab.1.

Table 1: Relevant temperature and pressure parameters

ΔT_w ($°C$)	5.00	ΔT_4 ($°C$)	7.00
ΔT_6 ($°C$)	3.00	ΔT_3 ($°C$)	8.00
ΔT_2 ($°C$)	13.00	ΔP_2 (MPa)	0.01
ΔT_8 ($°C$)	5.00	ΔP_4 (MPa)	0.03

4.1. Effect of Heat Source Temperature

The heat source temperature T_h are taken as 125-170 $°C$. The cooing water temperature T_w is taken as 30 $°C$ and the refrigerating temperature T_e is $5°C$. The variation of circulation ratio f with different heat source temperatures is shown in Fig.2. The circulation ratio f decreases with T_h increasing.

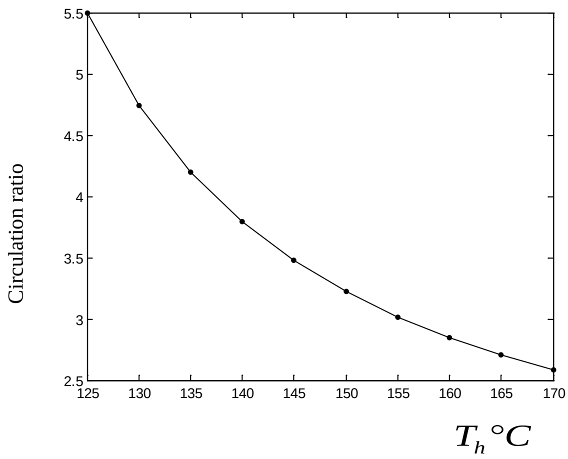

Figure 2: Variation of the circulation ratio with different heat source temperatures

The variations of COP and ECOP with different heat source temperatures are shown in Fig.3 and Fig.4. With the heat source temperature increasing, the COP value increases firstly and then decrease. The high COP value is obtained about at $145\,^{\circ}C$. However, compared with COP, ECOP value decrease and the variation of ECOP value is different greatly.

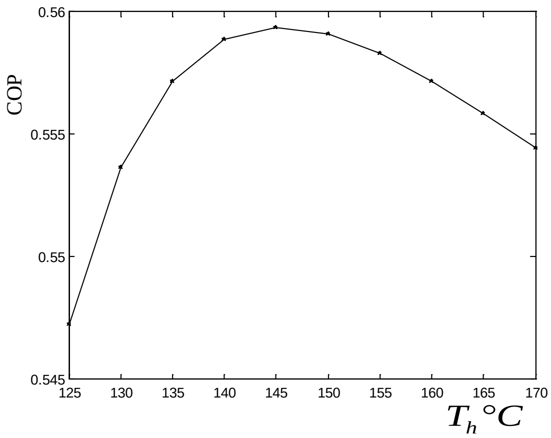

Figure 3: Variation of COP with different heat source temperatures

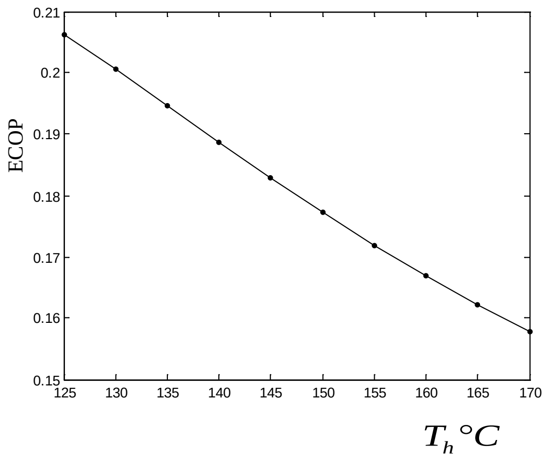

Figure 4: Variation of ECOP with different heat source temperatures

4.2. Effect of Cooling Water Temperature

The heat source temperature T_h is taken as $150\,^{\circ}C$ and the refrigerating temperature T_e is $5\,^{\circ}C$. The cooing water temperatures T_w are taken as 24-42 $^{\circ}C$ The variation of circulation ratio f with different cooing water temperatures is shown in Fig.5. The circulation ratio f increases with the cooing water temperature T_w increasing.

The variations of COP and ECOP with different cooling water temperatures are shown in Fig.6 and Fig.7. With the cooling water temperature increasing, the COP value increases and the ECOP value decreases.

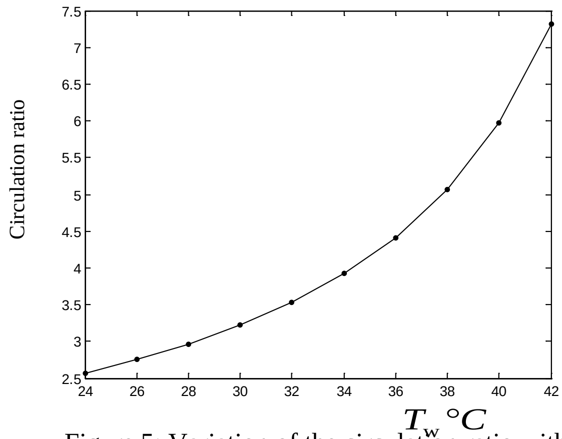

Figure 5: Variation of the circulation ratio with different cooling water temperatures

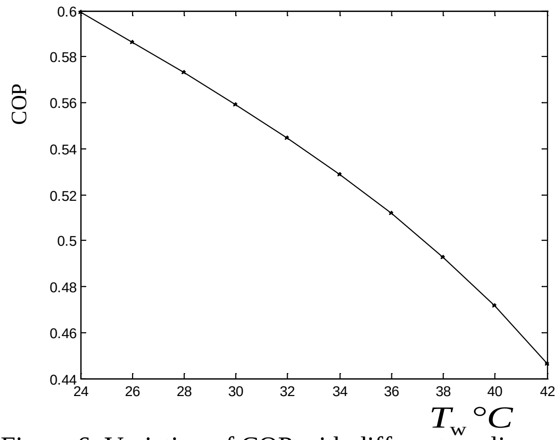

Figure 6: Variation of COP with different cooling water temperatures

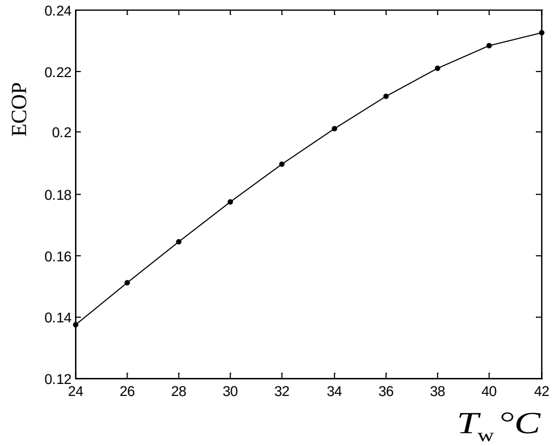

Figure 7: Variation of ECOP with different cooling water temperatures

4.3. Effect of Refrigerating Temperature

The heat source temperature T_h is taken as $150\,^{\circ}C$ and the cooing water temperature T_w is $30\,^{\circ}C$. The refrigerating temperatures T_e are taken as -15-15 $^{\circ}C$.

The variation of circulation ratio f with different refrigerating temperatures is shown in Fig.8. The circulation ratio f decreases with the refrigerating temperature T_e increasing.

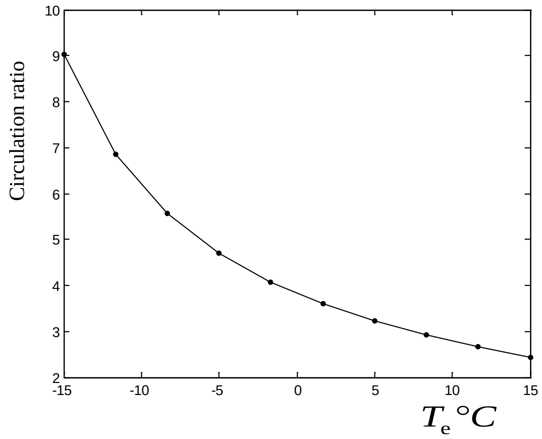

Figure 8: Variation of the circulation ratio with different refrigerating temperatures

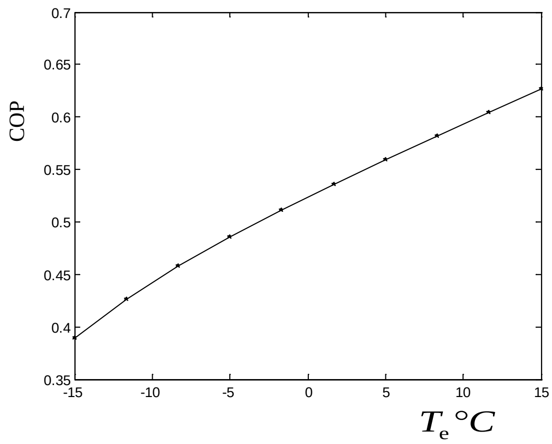

Figure 9: Variation of COP with different refrigerating temperatures

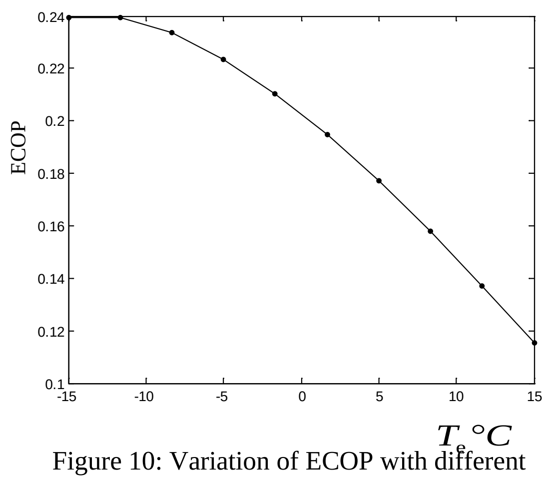

Figure 10: Variation of ECOP with different refrigerating temperatures

The variations of COP and ECOP with different cooling water temperatures are shown in Fig.9 and Fig.10. With the refrigerating temperature increasing, the COP value increases, however, the ECOP value decreases.

5. CONCLUSIONS

An modeling and performance analysis of ammonia-water absorption refrigeration system for ocean-going fishing vessels has been investigated in this work. Based on the Schulz state equation, the thermodynamic parameters analysis is carried on. Considering heat source temperature, cooing water temperature and refrigerating temperature, a mathmatics model is constructed in which COP and ECOP are taken as the evaluation parameter for ammonia-water absorption refrigeration system. The simulation results showed that the effects of heat source temperature, cooing water temperature and refrigerating temperature on COP and ECOP are different greatly, for COP is built on the first law and ECOP is on the second law of thermodynamics. Thus can be used for the optimization design of ammonia-water absorption refrigeration system.

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In the post-graduate stage, the applicant participated in several scientific researches supported by national and provincial funds. Based on traditional topology optimization and element free Galerkin method (EFGM), the topology optimization design for multi-stiff structure, multi-compliant mechanism and heterogeneous heat conduction structure with specific heat characteristics, are investigated, respectively. On this basis, the concurrent design method for structure with material was proposed. And the digital design framework of Ideal Functional Material Components (IFMC) was constructed. Involved in these projects, the applicant has accumulated rich experience in various aspects, such as numerical modeling and simulation, structural topology optimization, optimization algorithm design and so on.

Since staring work at School of Mechanical Engineering, Dalian Ocean University in April 2008, the applicant has launched a series of exploratory researches with the support of DOU’s Startup Fund for Doctor and the National Natural Science Foundation of China under grant No. 51109023, which mainly include numerical simulation for fishing vessels power plant, fishing vessels’ structure and material optimization, fishing vessels’ energy structure optimization, and processing complex data.