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# Development of a Compact Absorption Chiller Using Hydrophobic Flat Sheet Membrane

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## ABSTRACT

Automotive absorption refrigeration systems have enormous potential to conserve energy using exhaust gas from the car engine. The absorption system, however, has not yet been used for automobiles owing to its large volume, weight, and the weaknesses of unexpected vibrations and slope while driving. The LiBr solution may be scattered by the vibration on the cooling tube in the absorber, leading to malfunction of the absorber. Further, because of the sloping road, the free surface of the solution in the generator and absorber may cause mixing of the refrigerant and the LiBr solution. To solve these problems, attempts were made to use a hydrophobic flat sheet membrane. The membrane is a porous medium that has microscale pores and is used for filtration to separate specific substances. By the hydrophobicity of the membrane, the LiBr solution cannot penetrate the pores of the membrane. Only vapor can penetrate the pores. This means the membrane can separate the LiBr solution and vapor refrigerant. By using the membrane for all the components of the absorption refrigeration system, the free surface of the working fluid is eliminated, and the mixing of fluids does not occur because of the sloping road. To reduce the volume of components, a combined generator–condenser unit and absorber–evaporator unit were designed, and these performance were measured.

## 1. INTRODUCTION

Compression-type refrigerating machines are used for automobile air conditioners. There is no energy consumption in heating in winter, because they use exhaust heat from the engine, but in summer, the compressor operates using the shaft power of the engine, so it consumes energy for cooling. Therefore, fuel consumption increases when the air conditioner is operating. In addition, although R134a is mainly used as a refrigerant for a compression-type refrigerator for automobiles, it affects global warming, because it has high greenhouse potential. If a compact absorption refrigerating machine that can utilize exhaust heat from the engine as a driving force can be developed, it can greatly contribute to reducing fuel consumption in summer. However, owing to its large size and weight, the existing absorption refrigerating machine is not suitable for vehicles that require lightweight and small-size refrigerating machines. Furthermore, in a container such as an absorber, a regenerator, an evaporator, a condenser, the absorbing solution or the refrigerant liquid flows down on the heat transfer tubes, and the heat and mass transfer is performed through the free interface. Therefore, the refrigerant liquid or the absorbing solution could drift or mix owing to vibrations peculiar to the car or the inclination of the vehicle body.

Thus, in developing an automobile absorption chiller that utilizes exhaust heat from an engine, weight reduction, miniaturization, and heat and mass transfer devices that do not use a falling liquid film are required. In this study, we developed an absorption refrigeration system that transports vapor refrigerant through a hydrophobic flat membrane that does not allow liquid to pass through.

## 2. MEMBRANE-TYPE ABSORPTION REFRIGERATING MACHINE

### 2.1 Use of Hydrophobic Membrane

In this section, a method of applying a hydrophobic flat membrane in the absorber of an absorption refrigerating machine is described. As shown in Fig. 1, the absorption solution (aqueous solution of lithium bromide) flows between a hydrophobic flat membrane and a cooling water pipe. The flat membrane has fine pores and is hydrophobic. The absorption solution cannot penetrate the pores, and only the gas can pass through the membrane driven by the pressure difference. As seen in the enlarged view on the right side of Fig. 1, the gas–liquid interface is formed in the pore. Refrigerant vapor (water vapor) generated in the evaporator reaches the opposite side of the flat membrane, passes through the pores, and is absorbed into the absorption solution. The driving force by which the refrigerant vapor is absorbed is the difference of the vapor pressure between the refrigerant and the solution on the surface. The solution at the entrance of the pores absorbs water vapor, resulting in a decrease in concentration and an increase in temperature. To absorb the refrigerant vapor continuously, it is necessary to keep the vapor pressure of the solution low. Therefore, the gas–liquid interface should be updated quickly.

To improve the performance of the absorption chiller utilizing the hydrophobic flat membrane, it is necessary to select a suitable membrane and to form a shallow flow channel for increasing the flow velocity of the solution. The requirements for the membrane are high hydrophobicity (large contact angle) and high refrigerant vapor permeability (large pores and as thin as possible). When an appropriate membrane is selected, the solution does not permeate the membrane, and the vapor refrigerant is absorbed into the solution through the membrane pores. With this structure, because the vapor refrigerant and solution can be completely separated by a membrane, there is no possibility of mixing owing to vibration and inclination of the vehicle.

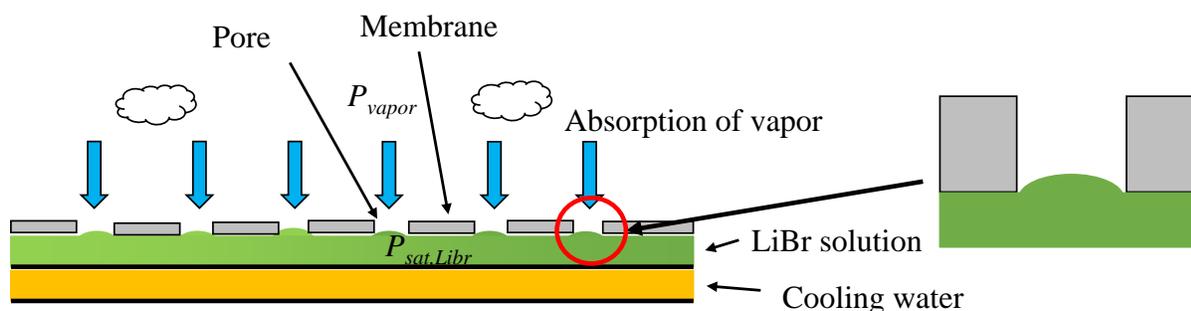


Fig. 1. Absorption through a flat membrane

### 2.2 Absorber–Evaporator Integrated Unit

Thus far, research on absorption phenomena using hydrophobic films has been conducted (Nasr-Isfahani et al., 2014), but the miniaturization of the apparatus is not sufficient. The purpose of this research is to make the absorber and the evaporator an integrated unit and make it smaller. A sectional view of the integral unit of the absorber–evaporator is shown in Fig. 2. Note that the regenerator–condenser integrated unit also has the same structure. The unit is mainly composed of five plates. The evaporator is on the left side, and the absorber is on the right side. The flow path for chilled water is engraved on the leftmost plate, the flow path for the refrigerant liquid is engraved on the next left plate, a spacer is centrally located, the flow path for the solution is engraved on the next left plate, and the flow path for the cooling water is engraved on the rightmost plate. The spacer is used for thermal insulation to reduce the heat transmission from the absorber to the evaporator and is made of polycarbonate. Two hydrophobic flat membranes are placed on both sides of the spacer. The refrigerant vapor evaporated through the left membrane moves to the spacer, passes through the right membrane, and is absorbed by the solution. The three central plates are kept in a vacuum state using O-rings.

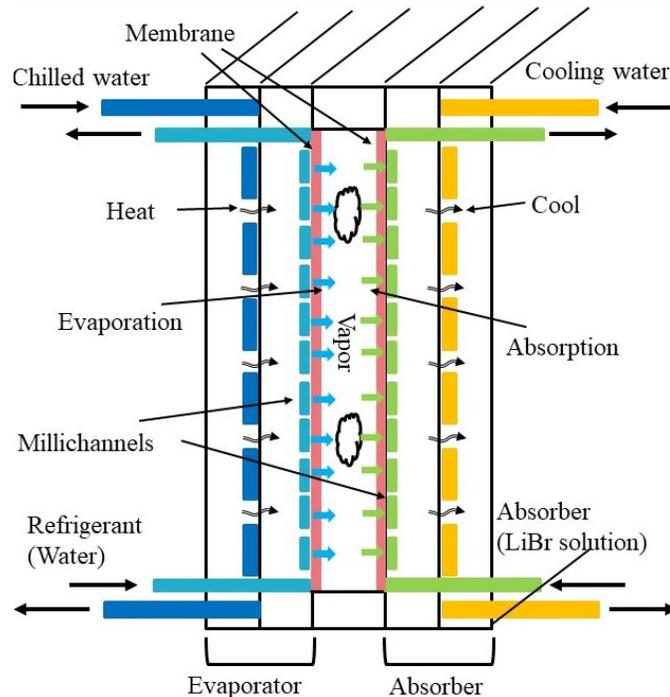


Fig. 2. Schematic of an absorber–evaporator combined unit

### 2.3 Distillation Performance of Hydrophobic Membrane

The refrigerant vapor moves through the pores of the hydrophobic membrane. Therefore, the permeation performance of vapor is an important factor. The vapor passes through the membrane with a driving force of the difference in vapor pressure on both sides of the membrane. According to the Darcy law, the permeation flux of the vapor is expressed in Equation (1) (Pena et al., 1998):

$$J_v = B_m \Delta p \quad (1)$$

$J_v$	permeation flux [kg/m <sup>2</sup> s]
$B_m$	permeation coefficient [kg/m <sup>2</sup> sPa]
$\Delta p$	vapor pressure difference [Pa]

Because the absorption refrigerating machine using an aqueous solution of lithium bromide operates in a vacuum state, the permeability coefficient in the pores on a micron order is often explained by the Knudsen diffusion model (Fan and Peng, 2012). The Knudsen diffusion model is established when the Knudsen number is sufficiently larger than 1. When the pore size is much smaller than the mean free path of gas molecules, collisions between molecules can be neglected, and the molecules move while repeating collision with the wall surface of the pores. Using the Knudsen diffusion model, the permeation coefficient can be expressed by Equation (2). This concept is widely used for calculation of the permeability of the membrane (Pena et al., 1998).

$$B_m = \frac{2\varepsilon r_p}{3\tau\delta} \sqrt{\frac{8M_v}{\pi RT_m}} \quad (2)$$

$\varepsilon$	porosity
$r_p$	pore diameter [m]
$\tau$	tortuosity
$\delta$	thickness of membrane [m]
$M_v$	molecular weight of gas [kg/mol]
$R$	gas constant [J/mol K]

$T_m$  temperature [K]

In calculating the permeation coefficient, it is necessary to know the tortuosity. The tortuosity is a value obtained by dividing the length from the entrance to the exit of the pore by the thickness of the membrane. The membrane used in this study has a structure in which the pores are not linearly opened but are complicated. Therefore, the tortuosity becomes larger than 1. However, it is difficult to estimate the tortuosity from the scanning electron microscopy image of the membrane. For this reason, we decided to adopt the value of 2.8 used in the research by Fan et al. [2] in this study.

### 3. RESULTS OF THEORETICAL SIMULATION

Table 1. Simulation conditions

Temperature of generator	90°C
Temperature of condenser	40°C
Temperature of evaporator	15°C
Temperature of absorber	35°C
Area of membrane used in absorber and evaporator	0.035 m <sup>2</sup>
Pore size of membrane	0.2 μm
Thickness of membrane	100 μm

The target cooling capacity was set to 200 W. The permeation performance of the membrane was theoretically analyzed to decide the area of membrane. The simulation conditions are shown in Table 1. The area of membrane used in the absorber and evaporator was fixed to 0.035 m<sup>2</sup>, the flow rate of the weak solution was fixed to 0.0008 kg/s, and the size of the membrane used in the generator and condenser was changed from 0.007 m<sup>2</sup> to 0.034 m<sup>2</sup>. We assumed that the temperature of the solution in each component was constant. The mass flux of the water vapor penetrating the membrane was determined according to Darcy's law and the Knudsen diffusion model, which were used. The result is shown in Fig. 3. The result shows that the size of the membrane enhances the cooling capacity and the coefficient of

performance (COP), but the rate of increase in the cooling capacity and COP becomes small. The membrane permeation performance is proportional to the difference between the saturation pressure of the solution and the vapor-side pressure. Because the vapor pressure difference of the generator is larger than that of the absorber, the membrane area of the generator is smaller than that of the absorber. Another simulation was conducted to decide the flow rate of the weak solution. The simulation conditions were the same as those in the previous simulation. The area of the membrane used in the generator and condenser was 0.022 m<sup>2</sup>. The flow rate of the weak solution was changed from 0.0005 kg/s to 0.0015 kg/s. The result is shown in Fig. 4. The result shows that the flow rate of the weak solution enhances the cooling capacity but deteriorates the COP. The increase in the solution flow rate causes the increase in the sensible heat loss in the solution generation process, and the COP decreases.

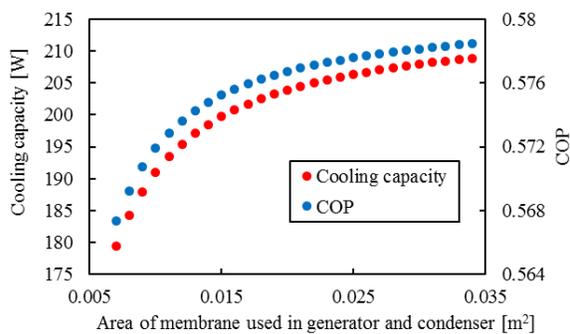


Fig. 3. Cooling capacity and COP with respect to area of membrane

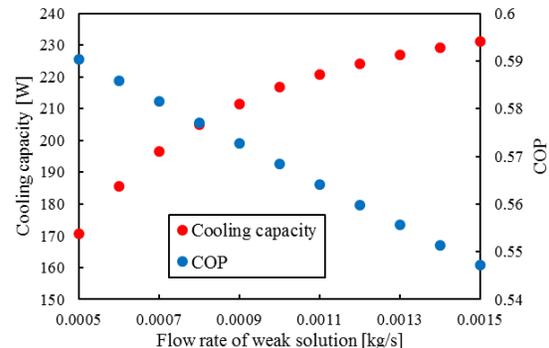


Fig. 4. Cooling capacity and COP with respect to flow rate of weak solution

## 4. EXPERIMENTS

### 4.1 Experimental Setup

In this study, an absorber–evaporator integrated unit was fabricated, as shown in Fig. 2, and vapor transfer characteristics were evaluated. By adjusting the solution temperature and the refrigerant temperature, it can also be

operated as a regenerator–condenser integrated unit. Figures 5 and 6 are system diagrams for testing the performances of an absorber–evaporator integrated unit and a regenerator–condenser integrated unit, respectively. The size of each unit was designed to demonstrate the cooling capacity of 200 W under normal cooling-temperature conditions.

Before starting the experiment, the temperature and concentration of the lithium bromide aqueous solution were adjusted using a heater equipped inside the lithium bromide aqueous solution tank. Two Coriolis flow meters were installed (flow meter 1, flow meter 2), and the flow rate, temperature, and density of the lithium bromide aqueous solution were measured. The solution concentration was calculated from these values. In the evaporator, the water that could not be evaporated was sent to the liquid tank and sent again to the evaporator using a pump. Regarding heated water, cooling water, and chilled water, the temperature was adjusted using a constant temperature bath and sent to the integrated unit. The space of the spacer part of the integral unit was connected to the vacuum gauge, and the pressure inside the spacer could be measured. In this study, a flat membrane made of hydrophobic polytetrafluoroethylene (PTFE) was used. The contact angle was approximately  $120^\circ$  for water and approximately  $130^\circ$  for a 55% aqueous solution of lithium bromide. Table 2 lists the main specifications of the membrane.

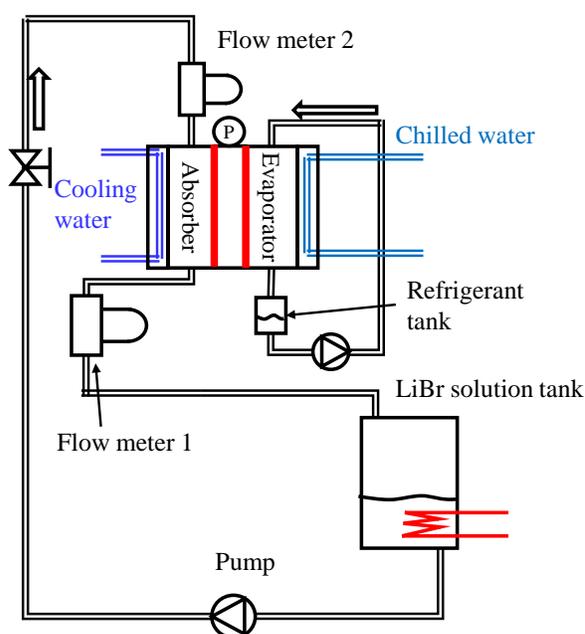


Fig. 5. Test loop for an absorber–evaporator combined unit

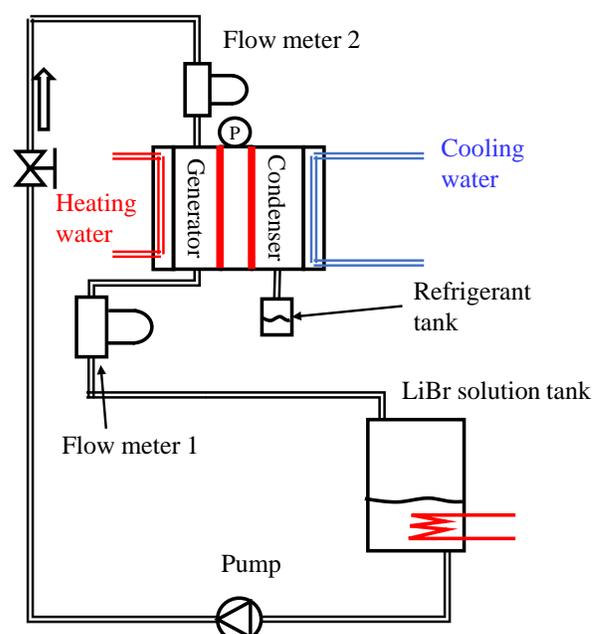


Fig. 6. Test loop for a generator–condenser combined unit

Table 2. Specification of flat membrane

Material	PTFE
Pore diameter	0.22 $\mu\text{m}$
Thickness	200 $\mu\text{m}$
Porosity	80%
Tortuosity	2.8 (assumed)

## 4.2 Experimental Conditions

Experiments were conducted under the absorber–evaporator integrated unit condition and the regenerator–condenser integrated unit condition; however, because of structural problems in the condenser, adequate results were not obtained. This time, the results for the absorber–evaporator integral unit condition were reported. Experimental conditions of the absorber–evaporator integrated unit condition are listed in Table 3.

Table 3. Experimental conditions for absorber–evaporator unit

Temperature of chilled water [°C]	15
Temperature of cooling water [°C]	30, 35
Pore diameter of membrane [ $\mu\text{m}$ ]	0.22
Flow rate of strong LiBr solution [kg/s]	0.003

### 4.3 Experimental Results

Figure 7 shows the time variation in the solution concentrations at the inlet and outlet at the cooling water temperature of 30°C. Ignoring the fluctuation at the start of the experiment and looking at the data when the concentration was stable, the concentration difference between the inlet and the outlet was approximately 0.7%. During the experiment, because the vapor did not condense in the spacer section, all the vapor evaporated in the evaporator moved to the absorber. The driving force for vapor transport in the absorber was the vapor pressure difference between the solution and the vapor in the spacer; in the evaporator, it is the vapor pressure difference between the liquid refrigerant and the vapor in the spacer. Figure 8 shows the experimental results of the permeability coefficient calculated using Equation (1) from the vapor pressure difference and the vapor transfer amount of the absorber and evaporator, and the theoretical value of the permeability coefficient was calculated using Equation (2). The vapor pressures in the absorber and the evaporator were obtained by averaging the saturated vapor pressures at the respective inlet and outlet.

The experimental values of the vapor pressure difference and the permeability coefficient and the theoretical permeability coefficients calculated by the Knudsen diffusion model are summarized in Table 4. The permeability coefficient on the evaporator side was about four times larger than that on the absorber side. The concentration gradient was generated because the solution was not sufficiently mixed in the flow channel, and the concentration of the LiBr solution in the pores of the membrane was low. As a result, the permeability coefficient of the absorber is estimated to be small. There is a possibility that the permeation performance may be improved by increasing the flow velocity with a reduction of the thickness of the solution flow path and suppressing the rise in vapor pressure. The tortuosity used for theoretical calculations cannot be ascertained with a microscope and has some uncertainty. In addition, it is also necessary to consider the problem of whether the permeability coefficient of water vapor in the hydrophobic membrane can be calculated by the Knudsen diffusion model.

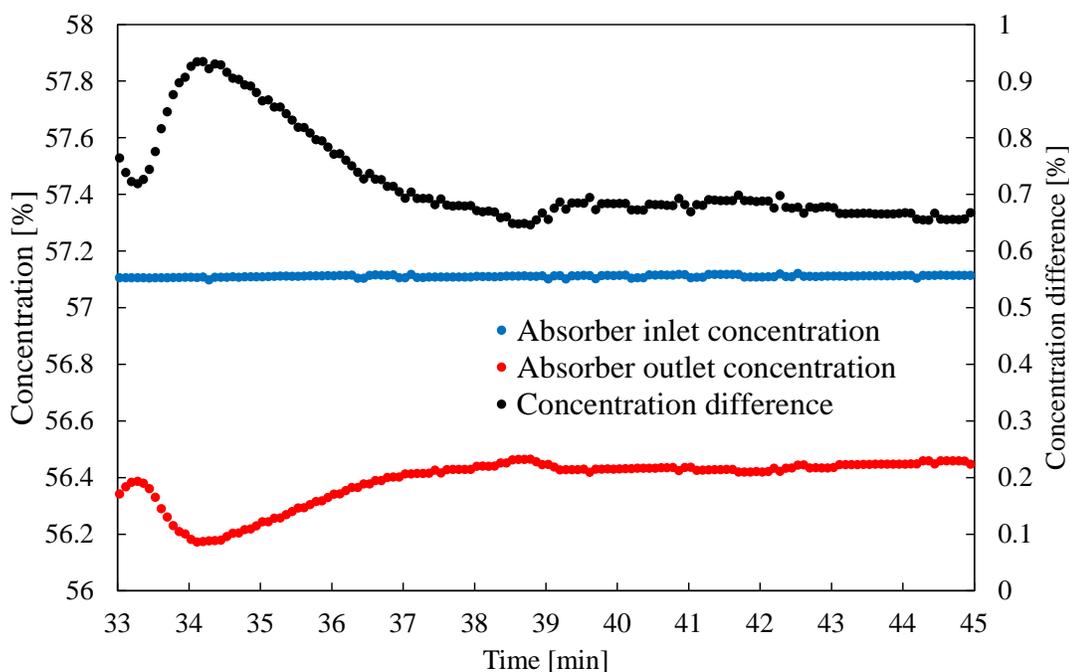


Fig. 7. Changes in concentrations of absorbent solution at the inlet and outlet in absorber

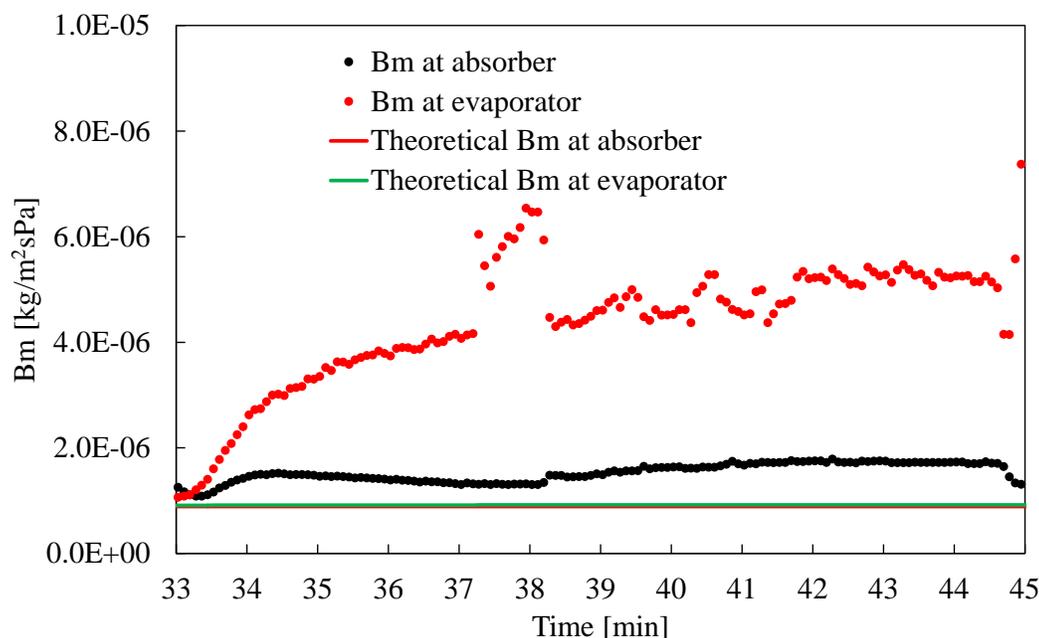


Fig. 8. Distillation coefficients at absorber and evaporator

Table 4. Summary of distillation coefficients of membrane

Temperature of cooling water [°C]	30	35
Temperature of chilled water [°C]	15	
Vapor pressure difference in absorber [Pa]	668	678
Vapor pressure difference in evaporator [Pa]	227	176
Permeation coefficient in absorber [kg/m <sup>2</sup> sPa]	$1.7 \times 10^{-6}$	$1.1 \times 10^{-6}$
Permeation coefficient in evaporator [kg/m <sup>2</sup> sPa]	$5.1 \times 10^{-6}$	$4.3 \times 10^{-6}$
Theoretical permeation coefficient in absorber [kg/m <sup>2</sup> sPa]	$8.9 \times 10^{-7}$	
Theoretical permeation coefficient in evaporator [kg/m <sup>2</sup> sPa]	$9.1 \times 10^{-7}$	

## 5. CONCLUSION

In this study, an experimental setup was constructed based on the simulation and the validity study of an absorptive refrigerating machine for automobiles, and the performance of the absorber–evaporator integrated unit was measured. From the experimental results, the permeability coefficient of the membrane was determined and compared with the theoretical value of the Knudsen diffusion model. There was a gap between the experimental and theoretical values, and it was found that a method for measuring the physical properties, such as pore size distribution, porosity, and tortuosity of the membrane, should be established. It was also suggested that the permeation performance could be improved by increasing the solution flow rate. In the future, we plan to proceed with the experiment on the regenerator–condenser integrated unit, as well as measure the physical properties of membrane.

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