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Comparison on Vacuum Membrane Dehumidification Systems with Moisture Selective Dense Membrane

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Abstract

Membrane heat pump using vacuum membrane dehumidification (VMD) systems have been investigated recently due to the potential of significant enhancement of energy efficiency. The principal of VMD systems lie in the nature of selective transmission of moisture through dense membrane. The driving force of separation is the vapor pressure gradient between feed and permeate side of a membrane, and the vapor is discharged to the atmosphere through a vacuum compressor. In this paper, performance analysis of three different type of VMD systems – a bypass air mixed type, a water vapor discharge type, and a condenser combined type – are compared to evaluate the performance under same temperature and humidity condition. From the simulation, water vapor discharge VMD shows the best dehumidification COP than others. Further, some conditions like ambient air temperature and relative humidity conditions or minimum vapor pressure difference which are assumed in the basic simulation are varied to simulate in depth. Simulation results are discussed at a practical application point of view.

Keywords: Vacuum membrane dehumidification; Dense membrane; Vacuum compressor; Dehumidification COP;

1. Introduction

These days, cooling demand is rapidly increasing due to global warming and globally increased cooling demand [1]. However, conventional vapor compression cycle which has been widely used for cooling leads acceleration of global warming and ozone depletion because of refrigerants it uses. It is necessary to find new technologies from the different point of traditional ones. Most widely used traditional technology for cooling was vapor compression technology. Vapor compression cycles can supply space cooling to maintain a comfortable environment within buildings. Since it was started to use for artificial active cooling, many researchers tried to improve its efficiency and it contributed greatly to energy savings so far. However, it is not enough to satisfy increasing cooling demand due to the improvement of a standard of living. Heating, ventilation and air conditioning (HVAC) system is regarded as an important technology and it accounts for almost 40% of global energy consumption [2]. Moreover, conventional refrigerants used in vapor compression cycles cause global climate change and the ozone layer depletion when they are released to the atmosphere. Some alternative refrigerants have low global warming potential, but they still have issues of toxicity, low efficiency, flammability or high equipment cost. To approach this problem by the new way, alternative systems are considered to replace classical vapor compression cycle.

According to Building Technologies Office in US Department of Energy, non-traditional technologies were evaluated by energy saving potential, installing and operational cost, complexity and realization possibility, etc [3]. From the report, moisture selective membrane was introduced as a key material to reduce energy

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consumption during cooling and dehumidification. So far, its potential of cooling for residential and commercial buildings in all climate regions was studied but yet to be realized as a commercial system.

As a successive study of the technical progress, this study investigated three types of vacuum membrane dehumidification systems by comparing their dehumidification COP. Simulation results were analyzed and compared under several reference conditions.

2. Description of Vacuum Membrane Dehumidification (VMD) System

A membrane is a selective barrier which is used to separate specific species from a mixture. Two sides of the membrane are referred as the feed side and the permeate side. Depending on existence of pores, a membrane can be classified into porous membrane or dense membrane. Porous membrane consists of physical pores sized from micro- to nanoscale. However, dense membrane consists of nanochannels without physical pores, where only a selected gas can pass through with a theoretical background of a solution-diffusion model [4]. When material of the dense membrane is hydrophilic ions, only water vapor can be separated from the air (Fig. 1). In this case, the vapor pressure difference between both sides of the membrane acts as the driving force [5-7]. Therefore, the water vapor can move to the other side by the water vapor pressure gradient. It means that separation can occur even if the absolute pressure of the feed side is lower than the permeate side.

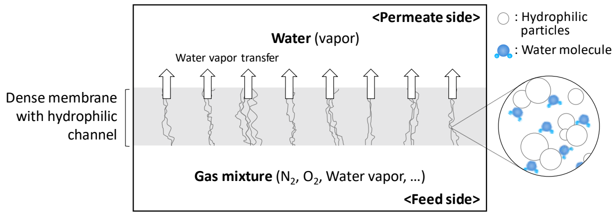

Fig. 1. Structure and vapor separation process of a dense membrane

VMD systems introduced from previous studies consist of a water selective dense membrane and a vacuum compressor [8]. Suction of the vacuum compressor is located at the permeate side of the membrane. Since the vapor pressure at the permeate side is lower than that of humid air, water vapor can be separated through the membrane, and the dehumidified air is supplied to conditioned space. As a simple configuration, direct compression vacuum membrane dehumidification system (D-VMD) is presented in Fig. 2.

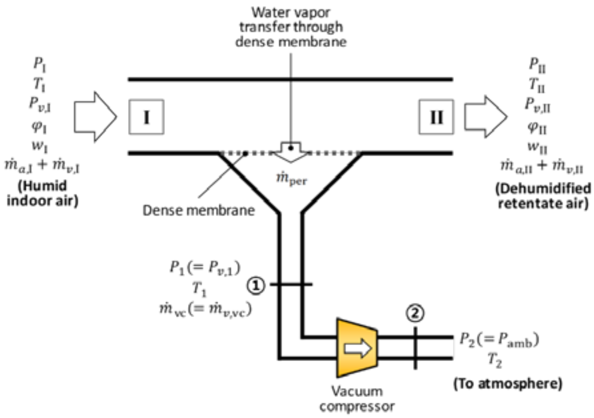

Fig. 2. Schematic diagram of direct compression vacuum membrane dehumidification system

As only water vapor is separated from the humid indoor air by vapor pressure difference, the permeate side pressure of the vacuum compressor inlet can be set below.

$$P_1 = P_{v,II} - \Delta P_{v,min} \tag{1}$$

where $\Delta P_{v,\text{min}}$ is required minimum vapor pressure difference for separation. Since humid air is dehumidified continuously while passing the membrane surface, $\Delta P_{v,\text{min}}$ is obtained based on the exit vapor pressure $P_{v,\text{II}}$ where the difference of vapor pressure is the minimized. Thus, $\Delta P_{v,\text{min}}$ can be called pinch point vapor pressure difference which is conceptually the same as pinch point temperature difference in the general heat exchanger.

Since air temperature is kept constant and only absolute humidity (w_I) is decreased during the separation process, the mass flow rate of permeated water vapor can be calculated as below. Assuming that water vapor selectivity of the membrane is infinite, the only water vapor will pass the vacuum compressor.

$$\dot{m}_{vc} = \dot{m}_{per} = (w_I - w_{II})\dot{m}_{a,I} \tag{2}$$

Then the permeated water vapor is compressed to atmospheric pressure (P_amb) by the compressor and discharged outwards. However, since vapor pressures ($P_{v,\text{I}}$, $P_{v,\text{II}}$) are typically very low in a few kilopascals, very high compression ratio (r_vc) of 60~100 is required to eject the compressed water vapor, which makes D-VMD system unrealistic. To resolve the high compression ratio issue, several other modified VMD systems like bypass air mixed type, water vapor discharge type, and condenser combined type were proposed.

2.1. Bypass air mixed vacuum membrane dehumidification system (B-VMD)

Figure 3 presents a bypass air mixed VMD (B-VMD) system [9]. As shown in this figure, a portion of dehumidified air is bypassed to the permeate side through a throttling valve.

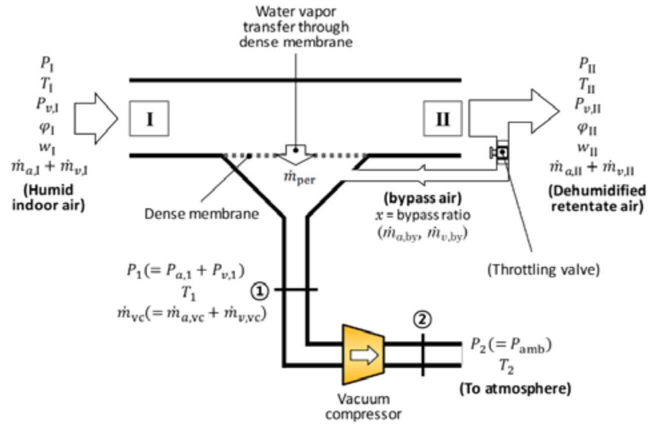

Fig. 3. Schematic diagram of bypass air mixed vacuum membrane dehumidification system

When bypass ratio x is defined by the fraction of bypassed retentate air to the permeate side, the dry air pressure of compressor inlet side ($P_{a,1}$) is multiplication of the bypass ratio and pressure of retentate air ($P_{a,\text{II}}$). Therefore, the total pressure of the vacuum compressor inlet side (P_1) becomes higher than the case of D-VMD. Since the vapor pressure of the permeate side is still lower than the feed side like Eq. (1), the vapor can be transferred from the feed side to the permeate side.

$$P_1 = P_{a,1} + P_{v,1} = xP_{a,II} + P_{v,II} - \Delta P_{v,min} \tag{3}$$

When two species are mixed at the same volume, volume flow rates of each material are calculated based on ideal gas law. At the compressor inlet side, the ratio of water vapor volume flow rate to the total flow rate is equal to the ratio of corresponding partial pressure as in Eq. (4).

$$\frac{\dot{V}_{v,1}}{\dot{V}_{a,1}+\dot{V}_{v,1}} = \frac{P_{v,1}}{P_{a,1}+P_{v,1}} \tag{4}$$

By combining Eq. (6) and ideal gas relation, bypass ratio (x) is derived as below.

$$x = \frac{P_{v,II}\dot{n}_{a,II} - P_{a,II}\dot{n}_{per} - \Delta P_{v,min}\dot{n}_{a,II}}{P_{a,II}\dot{n}_{v,II}} \quad (5)$$

Thus, the total mass flow rate through the vacuum compressor is presented as below.

$$\dot{m}_{vc} = \dot{m}_{a,vc} + \dot{m}_{v,vc} = \dot{m}_{a,by} + \dot{m}_{v,by} + \dot{m}_{per} = (x\dot{n}_{a,vc})M_a + (x\dot{n}_{v,vc})M_v + (w_I - w_{II})\dot{m}_{a,I} \quad (6)$$

In B-VMD system, since the mixture of permeated vapor and bypassed air is compressed by the vacuum compressor, compression ratio can be reduced in a reasonable level. However, as the flow rate through the vacuum compressor ($\dot{m}_{vc}$) increases, the vacuum compressor will consume more power.

2.2. Water vapor discharge vacuum membrane dehumidification system (W-VMD)

Figure 8 shows is water vapor discharge VMD (W-VMD) [10]. As seen in the figure, two membrane exchangers are located both at inlet and exit of the vacuum compressor. As in the previous configuration, the water vapor is permeated from the humid air through dense membrane 1. And the vacuum compressor pressurizes the water vapor higher than the vapor pressure of ambient side ($P_{v,III}$, $P_{v,IV}$). Then, the pressurized water vapor passes to the ambient side through dense membrane 2, which enables water vapor separation out of indoor air. As the water vapor is transferred through dense membrane 2, the state IV becomes the highest vapor pressure part of the ambient side. The compressor exit pressure is given as in Eq. (7), where the $P_{v,IV}$ is set slightly higher than $P_{v,III}$ because some permeated water vapor from the vacuum compressor is added to the ambient air stream.

$$P_2 = P_{v,IV} + \Delta P_{v,min} \quad (7)$$

From this concept, compression ratio of the vacuum compressor is lower by 3~5 because water vapor both at the compressor inlet and exit transfers based on partial vapor pressure. Due to the reduced compression ratio, compressor exit temperature is lowered in an acceptable range. Therefore, the flow rate and power consumption is getting much smaller since only water vapor passes the vacuum compressor.

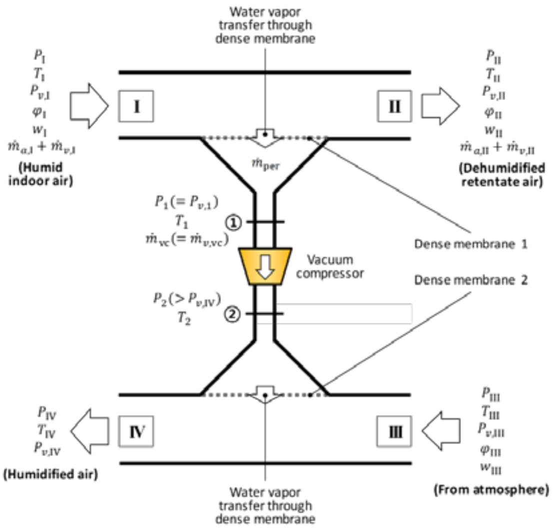

Fig. 4. Schematic diagram of water vapor discharge vacuum membrane dehumidification system

2.3. Condenser combined vacuum membrane dehumidification system (C-VMD)

The modified VMD model is a condenser combined VMD (C-VMD) system which is represented in Fig. 5 [11]. In this system, a condenser cooled by the ambient air is installed at the vacuum compressor exit, where the compressed water vapor condenses to the liquid state. The condensed water is then rejected to the ambient by a condensed water pump.

In this system, the compressor elevates the vapor pressure to the saturation pressure (P_{cond}) where the condensation temperature (T_{cond}) becomes higher than the ambient air temperature (T_{amb}). Therefore, the compressor exit side pressure is defined as below.

$$P_2 = P_{cond} \tag{8}$$

Since compression ratio is lowered in a practical range for the same reason as the W-VMD system, the C-VMD system can be a potential candidate of the direct VMD systems.

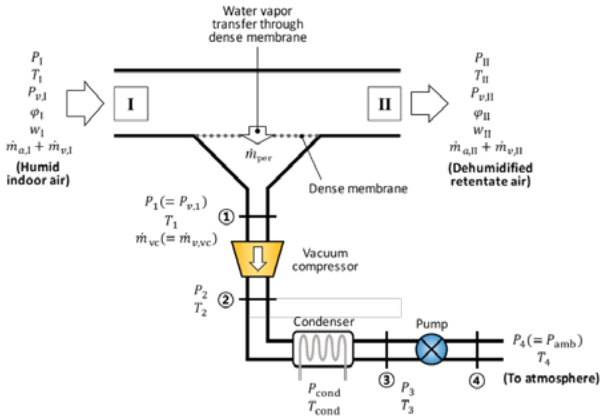

Fig. 5. Schematic diagram of condenser combined vacuum membrane dehumidification system

3. Simulation Methodology

Simulation conditions are set as in Table 1. In the VMD systems, dehumidification process is isothermal so no sensible cooling is considered, where dry bulb temperature of the retentate air and the feed air is the same.

Table 1 Simulation conditions for thermodynamic performance analysis of vacuum membrane dehumidification system

Location	Parameters	Values
Indoor air (Feed air) (I)[1)]	Dry bulb temperature (T_I) [°C]	27
	Relative humidity (φ_I) [%]	46.94
	Absolute humidity (w_I) [kgv/kga]	0.01045
	Mass flow rate ($\dot{m}_I$) [kg/s]	0.05
	Vapor pressure ($P_{v,I}$) [kPa]	1.6749
Retentate air (II)	Dry bulb temperature (T_{II}) [°C]	27
	Relative humidity (φ_{II}) [%]	32.59
	Absolute humidity (w_{II}) [kgv/kga]	0.00722
	Vapor pressure ($P_{v,II}$) [kPa]	1.1632
Ambient air (III)[2)]	Dry bulb temperature (T_{III}) [°C]	27 ~ 39
	Relative humidity (φ_{III}) [%]	40 ~ 90
	Absolute humidity (w_{III}) [kgv/kga]	0.00888 ~ 0.04123
	Vapor pressure ($P_{v,III}$) [kPa]	1.4272 ~ 6.3002
Membrane exchanger	Minimum vapor pressure difference ($\Delta P_{v,min}$) [kPa]	0.2 ~ 1.1
Vacuum compressor/Pump	Isentropic efficiency (η_{pump}, η_{vc}) [-]	0.8
Condenser	Temperature difference (ΔT) [°C]	3 ~ 7

1) The condition of indoor air are set based on ISO 5151 standard [12].
2) The specific condition of ambient air (35°C, 40.28% RH) is selected from ISO 5151 standard.

From the given total flow rate of feed air, $\dot{m}_I$= 0.05 kg/s, a set of absolute humidity (w) and vapor pressure (P_v) of indoor, retentate, and ambient air are calculated from each dry bulb temperature (T) and relative humidity (φ). The minimum vapor pressure difference ($\Delta P_{v,min}$) between the feed side and the permeate side is varied from 0.2 to 1.1 kPa. Isentropic efficiencies (η) of the compressor and the water pump are set as 0.8. For the C-VMD model, temperature difference in a condenser (ΔT) is set as 5°C. The permeance of a dense membrane in the VMD systems is referred to σ. The selectivity of air is not considered, which means only water vapor but no air can pass the membrane during the separation. For the simulation, MATLAB software is used for the calculation by combining thermodynamic property database REFPROP 9.0 [13].

To simulate the W-VMD and C-VMD systems, the pressure of membrane permeate side is set equal to P_1 and calculated by Eq. (1) with the assumed minimum vapor pressure difference. Since there is only water vapor at the compressor inlet, it can be said that the total pressure is equal to the vapor pressure ($P_1 = P_{v,1}$). The vacuum compressor exit pressure (P_2) is different for each model. In the B-VMD, the mixture of dry air and water vapor is discharged to the ambient directly, so it is compressed to the atmospheric pressure. On the other hand, the exit pressure of W-VMD is calculated by Eq. (7) based on the vapor pressure of ambient air because there is one more membrane at the compressor exit side. In this case, the vapor pressure difference condition at this additional membrane mass exchanger must be satisfied as well. For C-VMD, P_2 is determined by the condenser temperature which is higher than the ambient air as in Eq. (8). Since the compressed water vapor is condensed by the ambient air, the discharge pressure should be at the saturated pressure of the condenser. Thus, the compression ratio (r_{vc}) of the vacuum compressor in each VMD systems is provided as below.

$$r_{vc} = \frac{P_2}{P_1} = \frac{P_2}{P_{v,II} - \Delta P_{v,min}} \tag{9}$$

Mass flow rates of feed air, retentate air and permeated water vapor are calculated by Eq. (2) and conditions in Table 1. Vacuum compressor mass flow rates for W-VMD and C-VMD are same as that of permeated water vapor, but it is different in B-VMD because the bypassed air is mixed with the permeated water vapor before it enters to the compressor. Therefore, mass flow rate through the vacuum compressor in B-VMD system is determined by Eq. (6). When isentropic efficiency of the vacuum compressor is given by η_{vc}, exit enthalpy is calculated as below.

$$h_2 = \frac{h_{2s} - h_1}{\eta_{vc}} + h_1 \tag{10}$$

From the compressor exit pressure and enthalpy, the exit temperature can be obtained. For the total work of C-VMD, work by the water pump in Eq. (12) is added to that of the vacuum compressor.

$$\dot{W}_{vc} = \dot{m}_{vc}(h_2 - h_1) \tag{11}$$

$$\dot{W}_{pump} = \dot{m}_{vc} \frac{v_{cond}(P_{amb} - P_{cond})}{\eta_{pump}} \tag{12}$$

The latent heat removed by the VMD systems can be calculated as below. For B-VMD system, $\dot{m}_{a,by} h_{a,\mathrm{II}}$ is subtracted from the Eq. (13) due to feed air reduction by the bypassed air.

$$\dot{Q}_{lh} = (\dot{m}_a h_a + \dot{m}_v h_v)_I - (\dot{m}_a h_a + \dot{m}_v h_v)_{II} \tag{13}$$

COP_{DH} can be calculated by the total work and the total latent heat removed from the humid feed air passing the VMD system as in Eq. (14).

$$COP_{DH} = \frac{\dot{Q}_{total}}{\dot{W}_{total}} = \frac{\dot{Q}_{lh}}{\dot{W}_{vc}} \ or \ \frac{\dot{Q}_{lh}}{\dot{W}_{vc} + \dot{W}_{pump}} \tag{14}$$

4. Performance variation by ambient condition

From the above simulation models, the VDM systems are rated from W-VMD, C-VMD, to B-VMD in order of dehumidification performance. In this section, two VMDs of W-VMD and C-VMD which shows applicable performance among the three VMDs are compared as the variation of ambient air condition. For W-VMD, the ambient air is flowing at the permeate side of the membrane 2 at the compressor exit side. The discharged air from the compressor must have higher vapor pressure than that of the permeate side to pass through the membrane. In C-VMD, the vacuum compressor exit pressure (P_2) should be the saturation pressure at the condenser where saturation temperature is set higher by ΔT than ambient temperature. Thus, P_2 is closely related with T_{III}.

Figure 6 shows dehumidification performances simulated under various ambient temperatures. Ambient temperature is changed from 27°C to 39°C under relative humidity (φ_{III}) of 50%, 70%, and 90%. Since compressor exit pressure of the two systems are increased when both temperature and relative humidity increase, the vacuum compressor consumes more work to discharge the permeated vapor to the atmosphere, which results in degradation of COP_{DH}.

Additional simulation of C-VMD is conducted by varying the temperature difference at the condenser (ΔT). When changing the ΔT, the dehumidification performance of C-VMD is also changed because it affects the calculation of saturated vapor pressure. As shown in Fig. 7, when ΔT increases, COP_{DH} is decreased due to the rise of saturated vapor pressure. When we reduce ΔT by 3°C, C-VMD overtakes the performance of W-VMD in some cases of high humid condition in Fig. 7. This is because vacuum compressor exit pressure of W-VMD should be increased due to high vapor pressure of ambient air at the humid conditions. Surely, the larger the condenser size is, the better COP_{DH} becomes by reduced ΔT.

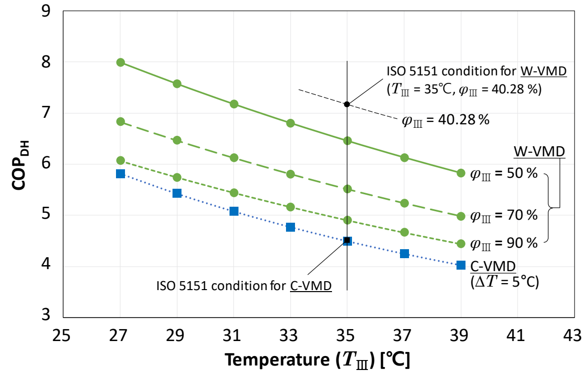

Fig. 6. Simulation results of W-VMD and C-VMD with various ambient temperature (TIII) and relative humidity (φIII) conditions

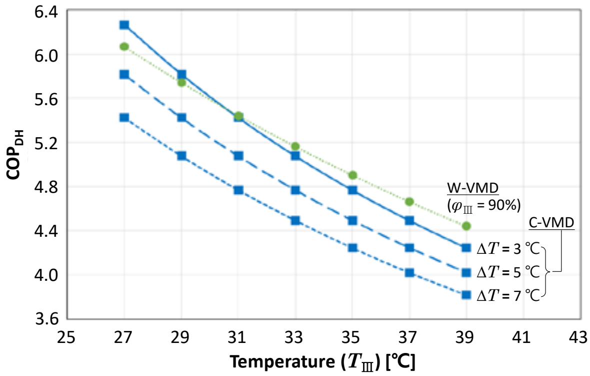

Fig.7. Simulation results of C-VMD with various ambient temperature (TIII) and ΔT conditions

Figure 8 presents a bar chart to compare dehumidification performance of the three VMD systems under selected combinations of ambient temperature and relative humidity. The minimum vapor pressure difference ($\Delta P_{v,min}$), the condenser temperature difference (ΔT), and bypass ratio (x) for B-VMD are fixed as 0.5 kPa, 5°C and 0.1266, respectively. As seen in the chart, the B-VMD system shows the lowest uniform COP_{DH} at all conditions since the performance of B-VMD is affected not by ambient condition but by bypass ratio. COP_{DH} of W-VMD shows a trend to decrease overall as increase of ambient temperature and humidity. As seen in the chart, COP_{DH} shows better performance at low ambient temperature. However, when comparing the humid case (T_{III}=29°C, φ_{III}=90%) and the hot dry condition (T_{III}=35°C, φ_{III}=40.28%), COP_{DH} of W-VMD shows higher value at the hot dry condition due to lower vapor pressure ($P_{v,III}$). Since vacuum compressor work in W-VMD depends on the exit side pressure which is a function of ambient vapor pressure, low ambient vapor pressure will reduce vacuum compressor work and enhance dehumidification performance. COP_{DH} of C-VMD is continuously decreased as increase of ambient temperature because high ambient temperature requires high condenser temperature. But the performance of C-VMD is only a function of ambient temperature, and not influenced by ambient humidity. The difference between COP_{DH} of W-VMD and C-VMD is reduced in humid condition.

From the mathematical simulation in this study, W-VMD shows the best dehumidification performance among three VMD systems. In terms of COP_{DH}, W-VMD should be used for air conditioning, but there is some problem that must be overcome to use this system in practice. It is known that the vapor passes through the membrane by absolute vapor pressure gradient. However, since the total pressure of membrane 2 permeate side in W-VMD is higher about 30 times than the feed side, the total pressure difference may act as resistance of vapor transportation. Therefore, the membrane should have permeability high enough to transfer the desired mass flow of water vapor and also have durability to withstand the high-pressure difference.

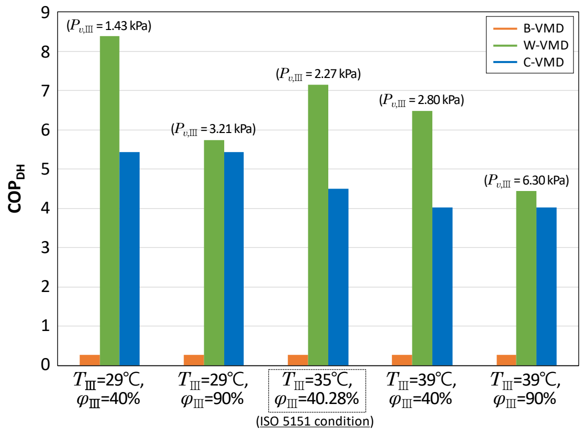

Fig. 8. Dehumidification performance of B-VMD, W-VMD, and C-VMD at selected temperature-humidity conditions of ambient air

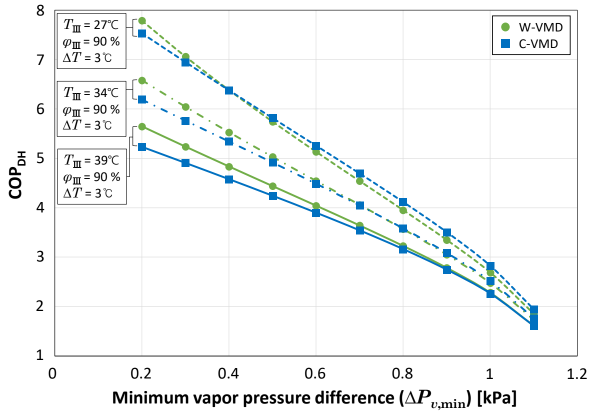

Fig. 9. Dehumidification COP variation by minimum vapor pressure difference

In contrast, C-VMD is practically simpler to implement than W-VMD from a technical point of view since only one membrane exchanger is required for C-VMD. With regard to the performance, C-VMD can compete COP_{DH} with W-VMD in a humid climate once the temperature difference of condenser (ΔT) is designed low. Figure 9 shows COP_{DH} of C-VMD and W-VMD about $\Delta P_{v,\min}$ variation at 90% of relative humidity when ΔT is 3℃. In the figure, COP_{DH} of both systems tend to decrease as $\Delta P_{v,\min}$ increases, but decline of C-VMD is smaller than that of W-VMD. Especially under low temperature condition, COP_{DH} of C-VMD exceeds W-

VMD partly. Therefore, considering the suitability of implementation, C-VMD can be an option to compete with the W-VMD system in humid climate.

5. Conclusion

Three VMD systems of bypass air mixed type (B-VMD), water vapor discharge type (W-VMD), and condenser combined type (C-VMD) are compared mathematically. In B-VMD system, a part of dehumidified air is bypassed to the vacuum compressor inlet, where the mixture of dry air and water vapor enters to the vacuum compressor. In contrast, since only water vapor exists at the vacuum compressor inlet in W-VMD and C-VMD, mass flow rate through the compressor is much smaller than the B-VMD case. In this manner, B-VMD shows much lower COP_{DH} than W-VMD and C-VMD by the simulation due to a large air flow through the compressor. W-VMD shows the highest dehumidification performance, but requires two membrane mass exchangers. To observe the performance under various climate condition, additional simulation was conducted for W-VMD and C-VMD.

As a recommendation of VMD system, W-VMD shows better performance overall than C-VMD. However, C-VMD shows comparable or better performance than W-VMD under hot and humid climate condition. Considering W-VMD requires two membrane mass exchanger, C-VMD can compete depending on the climate condition. Under the given minimum vapor pressure differences, $\Delta P_{v,\min}$ and compression ratio can be reduced for larger membrane size, which results in COP_{DH} enhancement. From the simulation, we can verify the performance of VMD systems at the design stage. For the experimental validation, the dense membrane having good performance is needed for small membrane size and high durability.

Nomenclature

COP	coefficient of performance [-]	φ	relative humidity [%]
h	enthalpy [kJ/kg]		
$\dot{m}$	mass flow rate [kg/s]	*Subscripts*	
$\dot{n}$	molar flow rate [kmol/s]	a	air
P_a	dry air partial pressure [kPa]	by	bypassed
P_v	vapor partial pressure [kPa]	cond	condenser
r	compression ratio [-]	DH	dehumidification
$\dot{V}$	volume flow rate [m^3/s]	I~IV	location of air flow
v	specific volume [m^3/kg]	lh	latent heat
$\dot{W}$	work [kW]	per	permeated
w	absolute humidity [kgv/kga]	s	isentropic
x	bypass ratio [mol/mol]	v	vapor
η	isentropic efficiency [-]	vc	vacuum compressor

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