14th IEA Heat Pump Conference
15-18 May 2023, Chicago, Illinois

Ideal performance analysis of membrane-based vacuum dehumidification systems

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Abstract

Non-vapor compression air-conditioning systems were proposed to overcome the limitation of the traditional air-conditioning systems. It was reported that the membrane heat pump(MHP) has a high energy saving potential. The sensible heat and latent heat removal process can be separated by the system. A membrane-based vacuum dehumidification system(MVD) is the latent heat removal systems, consisting of a membrane mass exchanger and a vacuum compressor. The water vapor discharge type membrane-based vacuum dehumidification system(W-MVD) and condenser combined membrane-based vacuum dehumidification system(C-MVD) show high performance among the several structure of MVDs. One system can show a higher performance than another system according to temperature and humidity conditions, but there is lack of discussion about the ideal COP values for actual system design, analysis, and optimization. In this study, the ideal dehumidification COP of the two systems were defined by reversible cycle analysis. It was achieved by removing all irreversibility of system consisting of membrane mass exchangers, vacuum compressor and heat exchanger, and the ideal COP formulas were derived. Exergy analysis and compressor consumption modeling methods were used to verify the formulas, and the COP values calculated by them show a high consistency. when comparing the ideal COP of the MVDs based on the results, the W-MVD shows higher performance than the C-MVD in unsaturation outdoor air condition.

Keywords: Membrane-based vacuum dehumidification system, Membrane mass exchanger, Vacuum compressor, Irreversibility, Ideal dehumidification COP;

1. Introduction

As the demand for building air conditioning increases worldwide and regulations on fluorine-based refrigerants are strengthened, the problem of using the existing vapor compression air conditioning system continues to be raised. At the same time, with the development of sensible load management technologies such as passive house technology and high-efficiency sensible heat control air conditioning system technology, the importance of the humidity control system is increasing to maintain thermal comfort for residents. Accordingly, a membrane heat pump (MHP) that can overcome the shortcomings of the existing dehumidification system has been proposed as one of the promising technologies [1].

The membrane heat pump uses the principle that water vapor can be separated from indoor humid air by utilizing a dense membrane with selective permeance to water vapor. Fig. 1 shows the change in the state of the theoretical air conditioned by each dehumidification system on the psychrometric chart. Unlike a condensation dehumidification system that simultaneously removes sensible heat and latent heat, the membrane heat pump can separate the sensible heat removal and latent heat removal process. Since it is not necessary to overcool the air and then reheat it, it is possible to take a thermal gain by the cooling path in the psychrometric chart. Since dehumidification using a desiccant is an isenthalpic process, thermal gains by the cooling path cannot be expected, but the disadvantages of desiccant that consume a large amount of heat during the regeneration process can be compensated. However, the coefficient of performance(COP) should be considered along with the state change of humid air to evaluate how efficient the system operates under certain cooling load conditions.

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Chun et al. constructed a system that combines the membrane-based vacuum dehumidification system(MVD) and the air carrying energy radiant air conditioning system to calculate the COP [2]. Since the dehumidification performance of the MVD varies depending on the vapor separated from humid air discharge structure, various structures have been studied to improve performance. Scovazzo & Scovazzo proposed a system that reduced the compression ratio of the vacuum compressor by bypassing a portion of the dehumidified air [3]. Dais Analytic proposed a system in which two membrane material exchangers were installed at the inlet and outlet of the vacuum compressor to reduce the compression ratio and vapor flow rate at the same time [4], and Bui et al. proposed a system in which a condenser was installed at the outlet of the compressor [5]. Lim et al. compared the dehumidification COP of each system and analyzed the effect of outside air conditions [6].

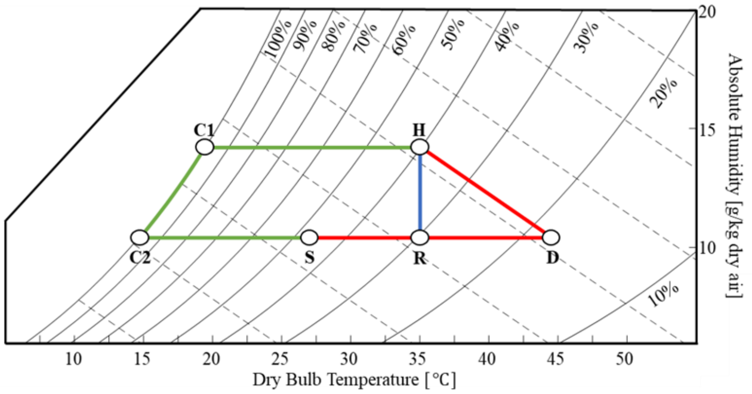

Fig. 1 Cooling path of each system on psychrometric chart; (a) condensation (H-C1-C2-S) (b) desiccant (H-D-S) (c) MHP (H-R-S)

Certain substances can be selectively permeated and separated from the mixture by the membrane, and permeation occurs due to chemical potential differences between two sides of the membrane. In the gas-gas separation membrane, it is divided into a porous membrane and a dense membrane according to the presence or absence of a physical pore through which gas molecules can pass, as shown in Fig. 2, the difference in partial pressure of each gas acts as a chemical potential difference. The permeation performance of the membrane is expressed in terms of permeance, which means the transmittance of a specific material per unit potential difference, and selectivity, which means the transmittance ratio of the two materials. The permeance and selectivity of the dense membranes used in the MVD mainly affect the dehumidification capacity and COP of the system, but they are in trade-off relation as shown in Fig. 3, so it is important to select the appropriate membrane according to the situation. Accordingly, experimental studies are being conducted to accurately measure the permeation characteristic values of the membrane under the operation conditions of the system to be actually used. Lee et al. measured the water vapor permeance and selectivity of the water vapor selective dense membrane that can be used in the MVD according to the pressure difference before and after the membrane and the transmission direction of water vapor and air [7].

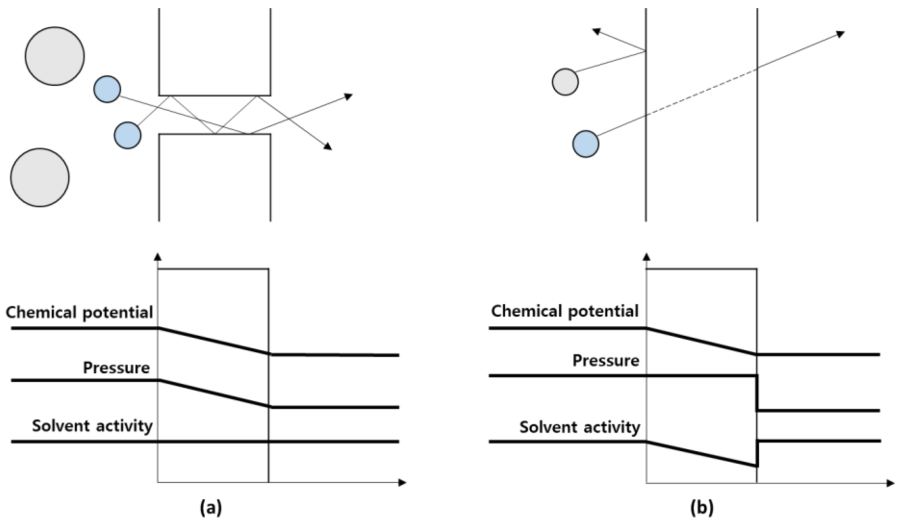

Fig. 2 Gas separation by partial pressure difference between two sides of membrane; (a) porous membrane (b) dense membrane

In previous studies on the performance of the MVD, the analysis was conducted assuming a membrane that completely blocks dry air and transmits only water vapor, that is, a membrane with infinite selectivity. However, since even a selectivity of the membrane is infinite, there is irreversibility by finite pressure difference as driving force of permeation process. So, this study aims to calculate the ideal dehumidification COP of MVDs by removing all irreversibility for systems design and optimization.

In other previous studies on the ideal performance of air-conditioning system, ideal COP of liquid-desiccant dehumidification and condensing dehumidification system were proposed by Zhang et al. as shown follows [8]. The ideal COP of isothermal dehumidification by perfect gas separation without considering the water vapor discharge process was proposed by Bui et al. as shown Eq.(4) [5]. And Labban et al. proposed ideal COP of generalized air-conditioning system discharging water vapor to ambient using exergy analysis as shown Eq.(5) [9]. This exergy model was used as an evaluation criterion of the ideal COP of MVDs.

$$COP_{des.Zhang} = \frac{1}{T_R}\frac{T_{iso.O}T_{iso.I}}{T_{iso.O}-T_{iso.I}} \tag{1}$$

$$COP_{con.Zhang} = \frac{T_{DP.I}}{T_R-T_{DP.I}} \tag{2}$$

$$W_{sep} = -nRT(X_a lnX_a + X_w lnX_w) \tag{3}$$

$$COP_{MVD.Bui} = \frac{m_w h_{fg}}{\Delta W_{sep}} \tag{4}$$

$$COP_{DH.Labban} = \frac{\dot{m}_w(h_{wet}-h_{dry})}{\dot{m}_w\xi_w+\dot{m}_{dry}\xi_{dry}-\dot{m}_{wet}\xi_{wet}} \tag{5}$$

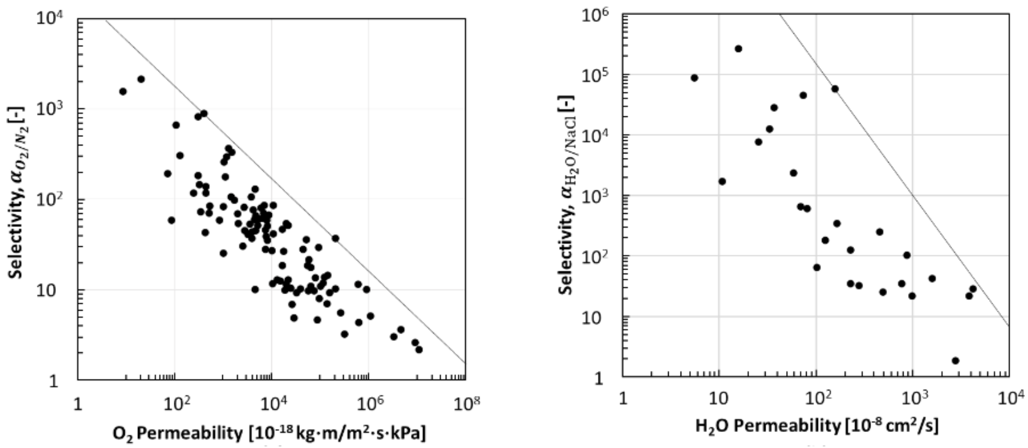

Fig. 3 Membrane permeance and selectivity trade-off; (a, left) O_2/N_2 [10] (b, right) H_2O/Nacl [11]

2. Simulation methodology

2.1. Idealized reversible MVDs

Fig. 4 shows the structure of water vapor discharge type MVD(W-MVD) and condenser-combined type MVD(C-MVD), which are analysis target systems of this study. W-MVD has two membrane mass exchangers installed at the inlet and outlet of the vacuum compressor, and C-MVD has a membrane mass exchanger installed at the inlet and heat exchanger at the outlet of the vacuum compressor. Assumptions to define the ideal dehumidification COP of the two systems were shown below.

- ✓ Both permeance and selectivity, the permeation characteristics of the membrane, are infinite.
- ✓ The effect of concentration polarization near the membrane surface is negligible.
- ✓ The water vapor adiabatically compressed to outdoor dry-bulb temperature and isothermally compressed to outdoor vapor partial pressure within non-condensation conditions.
- ✓ There is no irreversibility by temperature difference in the condenser of the C-MVD.
- ✓ The mass flow rate of indoor and outdoor air passing through the membrane mass exchangers is sufficiently large compared to the amount of water vapor permeation.

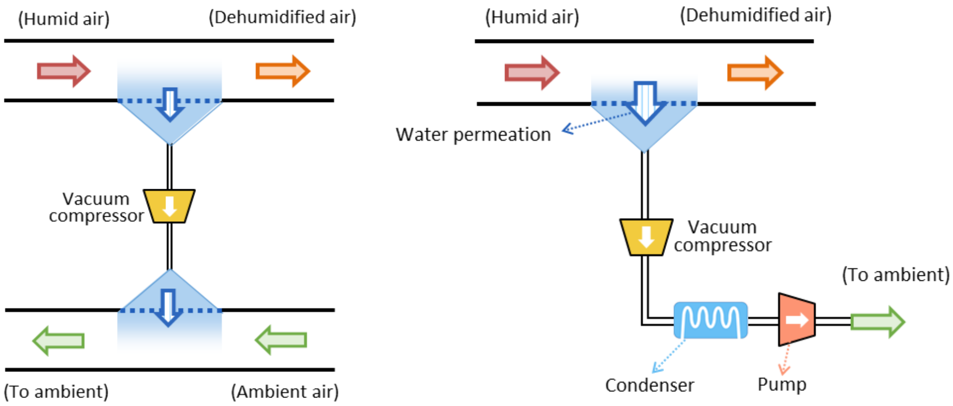

Fig. 4 Structure of the (a, left) W-MVD and (b, right) C-MVD

2.2. Mass transfer and compression model

The dense membrane separates water vapor from humid air according to vapor permeance and selectivity defined by Eqs. (6) and (7), respectively. Permeance is a property of a membrane determined by the water vapor transmission amount through the membrane under the partial pressure difference between the feed and the permeate side. The higher the water vapor permeance, the greater the amount of water vapor permeated through the membrane per unit area, per unit pressure difference, and per unit time. Selectivity is another property of a membrane determined by the transmission ratio between water vapor and dry air. The higher the selectivity, the more water vapor can transfer. Most previous studies have defined a membrane with infinite selectivity that can only permeate pure water vapor from humid air as an ideal membrane.

$$\beta = \frac{\dot{m}}{A \cdot \Delta P} \tag{6}$$

$$\alpha = \frac{\beta_w}{\beta_a} \tag{7}$$

Eq. (6) can be rewritten in the explicit form for mass flow rate of water vapor $\dot{m} = \beta A \cdot \Delta P$ where the formulation is similar to the heat transfer process. When the cross-sectional area is finite, infinite permeance is assumed for a reversible isothermal mass transfer process, such as assuming an infinite overall heat transfer coefficient for a reversible isothermal heat transfer process. Therefore, in this study, a membrane with infinite vapor permeance and selectivity was defined as an ideal membrane for dehumidification, and mass transfer through such a membrane is reversible.

Since the pressure difference between the feed side and the permeate side is zero in the ideal membrane, the inlet pressure of the vacuum compressor is the same as the water vapor partial pressure of the indoor air in both MVDs. The vacuum compressor outlet pressure is the same as the water vapor partial pressure of outdoor air in the case of W-MVD, and the water vapor saturation pressure at the outside temperature in the case of C-MVD. At this time, the suction and discharge pressure of the vacuum compressor must not exceed the saturation pressure at each temperature so that water vapor does not condense. If the indoor water vapor partial pressure is higher than the outdoor water vapor partial pressure, indoor water vapor can be discharged to the outside with only the membrane mass exchanger. This is a passive ventilation process, and MVD is unnecessary. Therefore, this case is out of analysis range. Only the case where the outdoor vapor partial pressure was higher than that of the indoor was considered. Fig. 5 is a T-s diagram showing the compression process of MVDs under the condition that the indoor and outdoor relative humidity is 100%. Since the water vapor partial pressure of outdoor air is the saturation pressure, the two MVDs are expressed in the same diagram. Due to the feasibility of isothermal compression, the water vapor would be adiabatically compressed from point 1 to 2'. Then the temperature difference between the vacuum compressor outlet and the outdoor air cause heat exchange. Since this irreversibility should be removed for ideal dehumidification of MVD, it is assumed that adiabatic compression from point 1 to point 2 and isothermal compression from point 2 to point 3 are performed. Assuming that water vapor is an ideal gas, the dehumidification COP are as follows.

$$COP_{DH} = \frac{\dot{m}_w h_{fg}}{\dot{W}_{isen} + \dot{W}_{isoth}} = h_{fg}\left[\frac{kRT_1}{k-1}\left(r_{p1}^{\frac{k-1}{k}} - 1\right) + RT_2 \ln r_{p2}\right]^{-1} \tag{8}$$

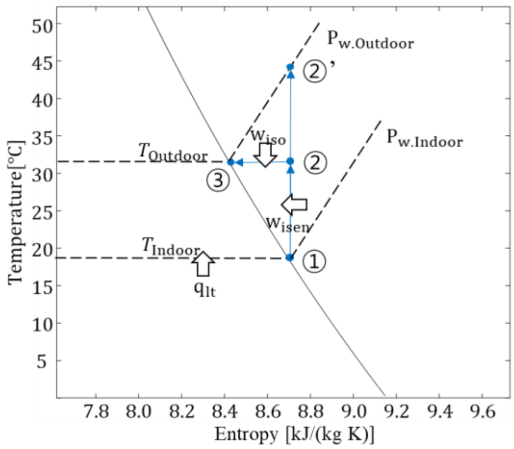

Fig. 5 Compression process on T-s diagram of W-MVD and C-MVD under indoor and outdoor saturation conditions

2.3. Theoretical dehumidification performance

Fig. 6 is a T-s diagram showing the approximate compression process of MVDs under conditions where indoor and outdoor relative humidity is less than 100%. Water vapor separated from indoor humid air is compressed to outdoor water vapor partial pressure in W-MVD and saturation pressure in C-MVD. That is, the outlet pressure of the vacuum compressor depends on both the condenser temperature and humidity in the W-MVD, but on only the condenser temperature in the C-MVD. Therefore, in order to remove irreversibility by heat transfer under unsaturated conditions, water vapor must be isothermally compressed at the dew point temperature of the outdoor air in W-MVD and at the dry-bulb temperature of the outdoor air in C-MVD. The permeation driving force of the membrane mass exchanger is the partial pressure difference of each gas, so the point I and O of W-MVD and point I of C-MVD can be approximated by saturated vapor points on the same pressure line, i.e., points 1 and 3 of W-MVD and 1 of C-MVD. Accordingly, the two MVDs approximately is a rectangular-shape open cycle on T-s diagram, such as a Carnot refrigeration cycle. That is, the ideal dehumidification COP of W-MVD can be expressed as a Carnot refrigeration COP, where the high temperature is the outdoor air dew-point temperature and the low temperature is the indoor air dew-point temperature, as shown in Eq. (9). Similarly, the ideal dehumidification COP of C-MVD can be expressed as a Carnot refrigeration COP, where the high temperature is the dry-bulb temperature of outdoor air and the low temperature is the indoor air dew-point temperature, as shown in Eq. (10).

$$COP_{W-MVD.ideal} = \frac{f(P_{w,I})}{f(P_{w,O}) - f(P_{w,I})} = \frac{T_{DP.I}}{T_{DP.O} - T_{DP.I}} \tag{9}$$

$$COP_{C-MVD.ideal} = \frac{T_{DP.I}}{T_O - T_{DP.I}} \tag{10}$$

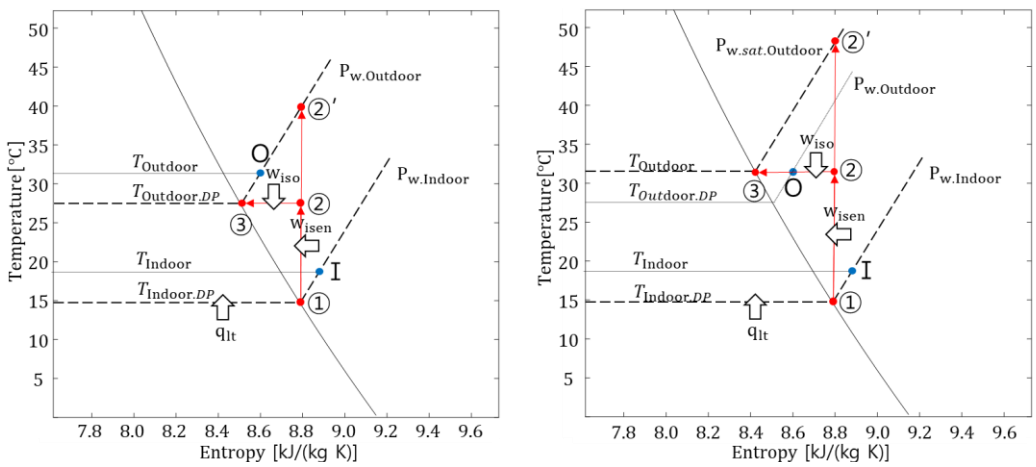

Fig. 6 Approximate compression process of (a, left) W-MVD and (b, right) C-MVD under indoor and outdoor unsaturated conditions

The dehumidification COP of both MVDs was calculated by Eq. (8). In the calculation process, the explicit method using MATLAB was applied based on MVD component modeling, and the thermodynamic properties are based on REFPROP 9.0. The results were mutually verified by comparing the ideal dehumidification COP calculated by Eq. (8) and Eqs. (9), (10). And the values and the theoretical maximum isothermal dehumidification COP calculated by Eq. (5) were compared. The detailed conditions applied to the performance analysis are shown in Table 1. The constraints conditions to analyze the effects of indoor and outdoor air conditions were represented as values in brackets. Both the isentropic efficiency and isothermal efficiency of the vacuum compressor were assumed to be 1.0, and the pump consumption of the C-MVD was ignored. Changes in indoor and outdoor air due to dehumidification and water vapor discharge were ignored, and the atmospheric pressure was assumed to be 101.325 kPa. Since the water permeance and selectivity of the membrane are assumed to be infinite, the compressor inlet and outlet conditions are the same as indoor and outdoor vapor conditions each. The temperature and humidity conditions were based on the ASHRAE 55 standard and KS C 9306 standard.

Table 1. Analysis conditions for ideal COP of MVD calculation

Location	Parameters	Values
Indoor air (Compressor inlet)	Dry-bulb temperature (T_I) [℃]	10~34 (24)
	Relative humidity (RH_I) [%]	50 ~ 100 (62.66)
	Absolute humidity (AH_I) [kgv/kga]	0.0093 ~ 0.0189 (0.0117)
	Dewpoint temperature ($T_{DP.I}$) [℃]	12.98 ~ 24 (16.44)
	Vapor partial pressure ($P_{w,I}$) [kPa]	1.493 ~ 2.985 (1.871)
Outdoor air (Compressor outlet)	Dry-bulb temperature (T_o) [℃]	16 ~ 40 (24)
	Relative humidity (RH_o) [%]	50 ~ 100 (84.22)
	Absolute humidity (AH_o) [kgv/kga]	0.0093 ~ 0.0489 (0.0158)
	Dewpoint temperature ($T_{DP.o}$) [℃]	12.98 ~ 40 (21.18)
	Vapor partial pressure ($P_{w.o}$) [kPa]	1.493 ~ 7.383 (2.513)
Vacuum compressor	Isentropic efficiency ($\eta_{isen.vc}$) [-]	1.0
	Isothermal efficiency ($\eta_{iso.vc}$) [-]	1.0

3. Results and discussion

Fig. 7 shows the ideal dehumidification COP ratio of the two MVDs according to outdoor air relative humidity calculated by Eqs. (9) and (10). By the absence of irreversibility, the ideal dehumidification COP of C-MVD is consistent with the dehumidification COP of W-MVD with 100% outdoor relative humidity. However, under unsaturated conditions of outdoor air, the ideal dehumidification COP of W-MVD is always higher than that of C-MVD, because the dry-bulb temperature is always higher than the dew-point temperature. Under the dehumidification conditions of KS C 9306 standard (T_I=24°C, RH_I=62.66%, T_O=24°C, RH_I=84.22%), C-MVD shows about 63.21% of W-VMD. Since the result of Eq. (9) is consistent with the result of Equation (8) under the saturation condition of outside air, the performance of W-MVD and C-MVD was analyzed with Eq. (8) in the subsequent result analysis.

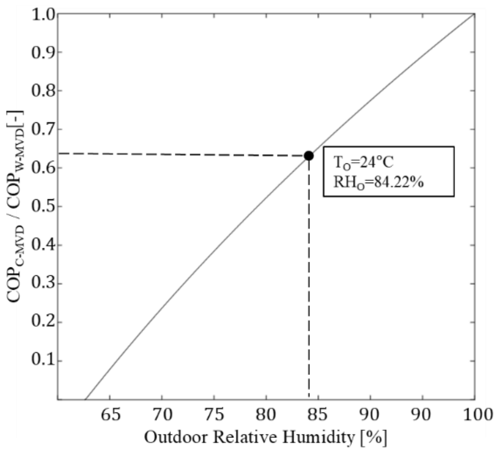

Fig. 7 Ratio of ideal dehumidification performance of W-MVD and C-MVD according to outdoor air relative humidity (T_I=24°C, RH_I=62.66%, T_O=24°C)

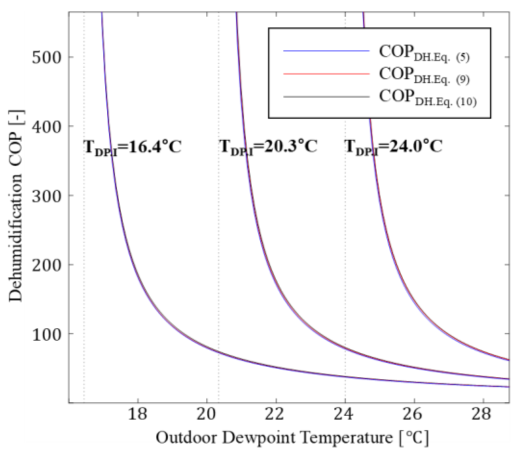

Fig. 8 Theoretical dehumidification COP of MVD according to the outdoor air dew-point temperature by Eqs. (5), (9) and (10) (T_I=24°C, RHo=100%)

Fig. 8 shows the theoretical dehumidification COP according to the outdoor air dew-point temperature for each indoor dew-point $_{\text{temperature}}$ calculated by Eqs. (5), (8), and (9). The indoor temperature is 24.0°C and the relative humidity is 62.66% to 100%, and the outdoor temperature varies from 16°C to 30°C and the relative humidity is 100%. The values obtained by the three methods are almost identical, and the ideal dehumidification COP of W-MVD and C-MVD under outdoor air saturation conditions is equivalent to the theoretical maximum COP of any isothermal system that discharges water vapor into the outdoor air.

3.1. Influence of temperature and humidity

In order to consider the effects of climate conditions on dehumidification COP of MVDs, dehumidification COP variation according to outdoor temperature and relative humidity were analyzed. Fig. 9 shows variation by climate conditions. It is assumed that the temperature of the indoor air is 24°C, and the relative humidity is constant at (a) 100% or (b) 50%. The outdoor temperature changed from 23°C to 40°C, and the relative humidity changed from 50% to 100%. As the temperature or relative humidity of outdoor air increased, the dehumidification COP decreased under all conditions. This is because the water vapor separated from the indoor air and passed through the membrane must be compressed to a higher pressure as the partial pressure of outdoor water vapor increases, resulting in an increase in vacuum compressor power consumption. This can also be explained by the fact that under the $T_{DP.I}$ condition set in Eq. (9), the larger the $T_{DP.O}$, the smaller the dehumidification COP, and is consistent with the intuition that the dehumidification system has higher COP in dryer climates.

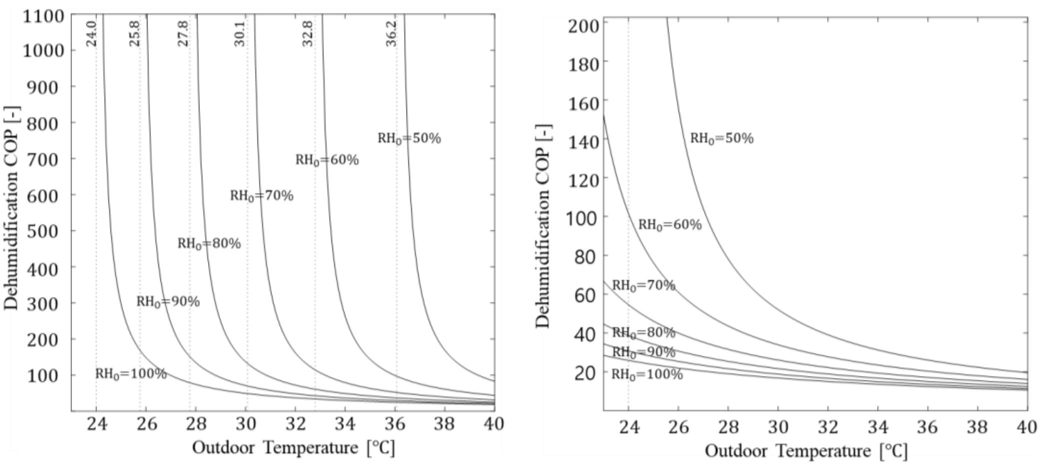

Fig. 9 Ideal dehumidification COP of MVD variation as outdoor air conditions of T_I=24°C; (a, left) RH_I=100%, (b, right) RH_I=50%

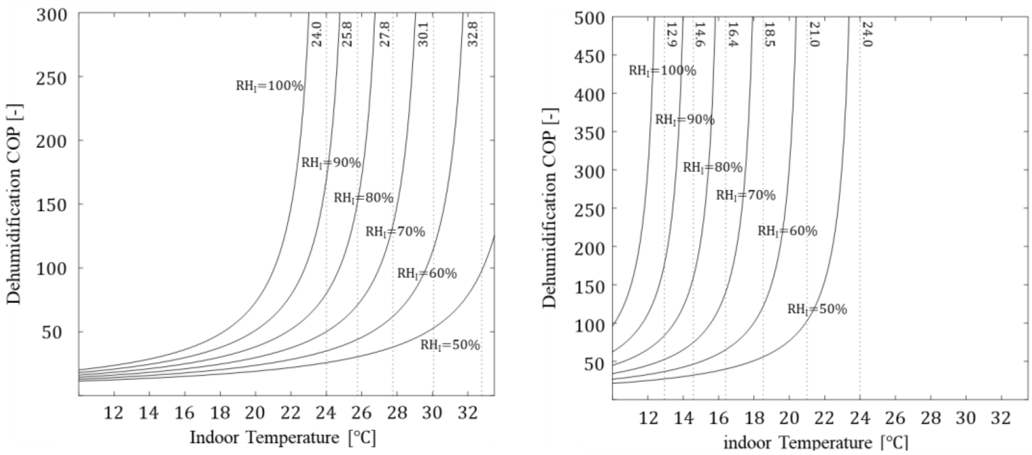

Fig. 10 Ideal dehumidification COP according to indoor air conditions of T_O=24°C; (a, left) RH_O=100%, (b, right) RH_O=50%

Since indoor conditions change when an air-conditioning system is operated, the effects of changes in indoor conditions on dehumidification COP under specific climate conditions was analyzed. Fig. 10 shows the ideal dehumidification COP variation according to the indoor air temperature and relative humidity. It is assumed that the outdoor air temperature is 24°C and the relative humidity is constant at (a) 100% or (b) 50%. The indoor temperature changed from 10°C to 34°C, and the relative humidity changed from 100% to 50%. As the indoor temperature or relative humidity increased, the ideal dehumidification COP increased. This is because the compression ratio of vacuum compressor decreases as the partial pressure of indoor water vapor increases. This can also be explained by the fact that under the $T_{DP.O}$ condition set in Eq. (9), the larger the $T_{DP.I}$, the larger the dehumidification COP, and is consistent with the intuition that the dehumidification system has lower COP in the humid state at the beginning of dehumidification.

3.2. Irreversibility analysis

The assumptions of the MVD are not valid in a real system due to the inevitable irreversibility, so the actual dehumidification COP of MVD cannot reach the ideal values. Considering the finite permeance of the membrane, the finite pressure difference on both sides of the membrane mass exchanger is required as the driving force of mass transfer, which acts as irreversible. The inlet pressure of the vacuum compressor should be lower than the indoor vapor partial pressure, and the outlet pressure of the vacuum compressor in W-MVD should be higher than the outdoor vapor partial pressure, so the compression ratio increases and the energy consumption of the vacuum compressor increases. Fig. 11 and Fig. 12 (a) show the dehumidification COP according to the outdoor air temperature for each finite pressure difference between both sides of the membrane with finite water vapor permeance and infinite selectivity of W-MVD and C-MVD, respectively. It was assumed that the indoor temperature was 24°C, the relative humidity was 62.66%, and the outdoor relative humidity was 84.22%. The lower the water vapor permeance of the membrane, the greater the required water vapor partial pressure difference, the higher the irreversibility of the system, so the dehumidification COP is lower than the ideal value. If the pressure difference is not required due to infinite water vapor permeance, the dehumidification COP is consistent with the ideal dehumidification COP.

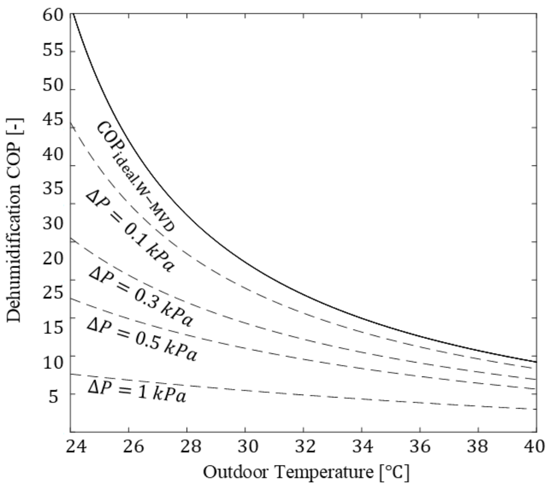

Fig. 11 Dehumidification COP reduction in W-MVD due to finite pressure difference (T_I=24°C, RH_I=62.66% and RH_O=84.22%)

Fig. 10 (b) shows the dehumidification COP of C-MVD with infinite membrane permeance and selectivity for each temperature difference in the condenser. Like the pressure difference between both sides of the membrane, the dehumidification COP approaches an ideal value as the temperature difference decreases, because water vapor must be compressed more at a temperature higher than the outdoor temperature dew to the finite temperature difference. In all results, the finite pressure difference and temperature difference were assumed to be uniform.

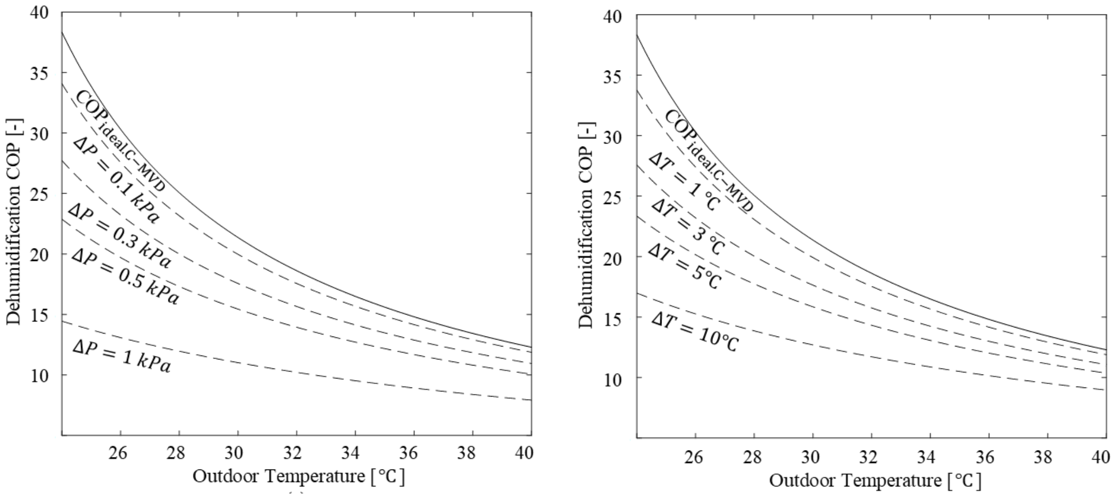

Fig. 12 Dehumidification COP reduction in C-MVD due to (a, left) finite pressure difference and (b, right) finite temperature difference (T_I=24°C, RH_I=62.66% and RH_O=84.22%)

In the compression processes of the ideal MVD, the water vapor separated from the indoor humid air was isothermally compressed at the outdoor temperature. However, in practice, it is difficult to implement isothermal compression as discussed in section 2.2, and a single adiabatic compression causes MVD to consume more energy as it is compressed to a temperature higher than the outdoor air temperature. Fig. 13 compares the ideal dehumidification COP and the dehumidification COP in this case. Systems that have a single adiabatic compression show lower performance under all temperature and humidity conditions than systems that have an ideal compression process.

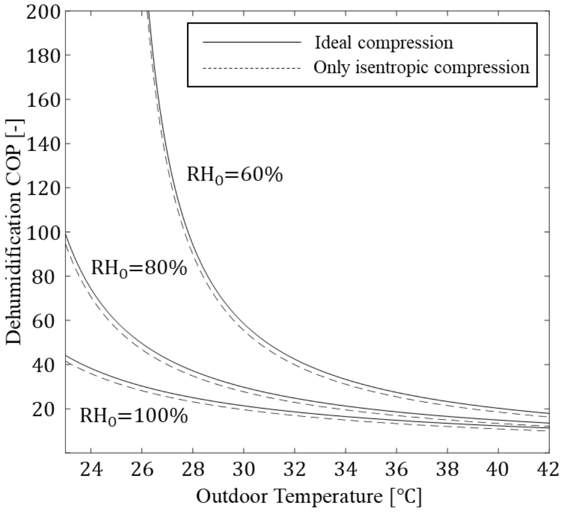

Fig. 13 COP reduction of MVD by single adiabatic compression

4. Conclusion

The ideal dehumidification COP of MVDs was analyzed in this study. Assuming that there is no irreversibility of the membrane mass exchanger, vacuum compressor, and heat exchanger, the ideal dehumidification COP of W-MVD and C-MVD was defined by Carnot cycle analysis. This was compared with the thermodynamical maximum dehumidification COP values calculated by exergy analysis and components modeling. Based on the newly defined ideal dehumidification COP, the influence of operation conditions and irreversibility of actual systems were analyzed. The results of this study were as followed.

- ✓ The dehumidification COP values calculated by the three methods matched. These results show dehumidification COP of MVDs can be predicted by only the temperature and humidity conditions of the indoor and outdoor air where MVDs are operated.

- ✓ The ideal W-MVD has a thermodynamical maximum dehumidification COP, which is 58.2% higher than the ideal dehumidification COP of C-MVD under reference condition.

- ✓ The dehumidification COP of the systems is better in cold, dry climates. The values near the asymptote are very high because the water vapor partial pressure of indoor air is higher than that of outdoor air or the difference is too low. The MVD is not useful in these conditions. however, it is possible to identify temperature and humidity conditions in which the dehumidification can be performed with low energy consumption or only simple passive ventilation without operation of the systems.

- ✓ Under the desirable temperature and humidity requirements, the ideal dehumidification COP of W-MVD is approximately 50-100 and considering the irreversibility inevitable in real systems such as finite pressure difference in mass transfer process or finite temperature difference in heat transfer process, the actual COP is lowered according to the degree of the irreversibility. The presented equations for the ideal dehumidification COP of MVDs can be used as a basis for research on performance of MVDs and for system design and optimization.

Nomenclature

A	Area [m^2]	α	Selectivity [-]
AH	Absolute humidity [kgv/kga]	β	Permeance [kg/m2·s·kPa]
h_{fg}	Specific enthalpy of vaporization [kJ/kg]	ξ	Specific exergy [kJ/kg]
k	Specific heat ratio [-]		
n	Number of moles [mol]	***Subscript***	
m	Mass [kg]	a	Air
$\dot{m}$	Mass flow rate [kg/s]	amb	Ambient air
P	Pressure [kPa]	con	Condensing dehumidification
P_w	Water vapor partial pressure [kPa]	des	Liquid desiccant dehumidification
R	Gas constant [8.314472 J/mol·K]	DH	Dehumidification
RH	Relative humidity [%]	dry	humidified dry air
r_p	Compression ratio [-]	iso	Isothermal compression
T	Temperature [K]	$isen$	Isentropic compression
T_{DP}	Dewpoint temperature [K]	I	Indoor air
T_{iso}	Iso-relative humidity line Temperature [K]	O	Outdoor air
T_R	Temperature of reference point of exergy [K]	sep	Complete gas separation
W	Work [kJ]	w	Water vapor
$\dot{W}$	Power [kW]	wet	Not dehumidified wet air
X	Mole fraction [-]	1,2,3	State point number

Acknowledgements

This research was jointly supported by National Research Foundation of Korea (NRF No. 2019R1A2C108869414) and by Korea Institute of Energy Technology Evaluation and Planning (KETEP) (20202020900290, 20214000000280, 20212050100010) funded by Ministry of Trade, Industry and Energy. This research was also supported by the Korea Environmental Industry & Technology Institute (KEITI No. 2020003060005) Authors sincerely appreciate their supports.

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