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

Design and energy performance of the heat pump-driven liquid desiccant system with an ultrasonic atomization

Soo-Jin Lee[a], Seong-Yong Cheon[a], Su-Young Jo[a], Jae-Won Joung[a], Jae-Weon Jeong[a,*]

[a]Department of Architectural Engineering, College of Engineering, Hanyang University, Seoul, 04763, Republic of Korea

Abstract

The purpose of this study was to propose and design the heat pump-driven liquid desiccant ventilation system with an ultrasonic atomization. The energy consumption was also compared with the conventional heat pump-driven liquid desiccant system. Because the proposed system could be operated at low solution flow rate than the reference system, the proposed system was suggested for energy benefits. The results indicated that the two systems showed similar heat pump cycle such as temperature of the evaporator and condenser. However, heat pump size could be reduced in the proposed system because of the low solution loads compared with the reference system. Finally, the proposed system could be operated with 30% reduced energy consumption during the summer season than the reference system.

Keywords: Liquid desiccant; heat pump; ultrasonic atomization; energy performance;

1. Introduction

The liquid desiccant (LD) has been suggested as an energy efficiency technology for air dehumidification because of its thermal characteristics [1,2]. The LD technology has based on the mass transfer of the air and solution. The driving force is water vapor pressure difference in the LD system, while the conventional dehumidification method using condensation of the water vapor in the air. LD system has energy saving potential and high efficiency compared with the conventional condensation dehumidification because it does not use unnecessary energy to cool the air below the dew point temperature.

The LD technology was based on the heat and mass transfer between the air and solution by the difference of the water vapor pressure. To make the vapor pressure difference, the solution is cooled and heated before the absorber and regenerator intake. The various heat sources to provide the cooling and heating have been investigated. Badami and Portoraro suggested [3] a trigeneration plant which combined LD system and cooling tower for absorber, natural gas combined heat and power cogeneration system for regenerator. Especially, the heat pump was considered as an energy conservative heat source because it provides cooling and heating concurrently. Shin et al. [4] investigated the energy saving potential of the heat pump-driven LD system compared with conventional LD system which used cooling tower and boiler. The results indicated that the proposed LD system could reduce about 33% primary energy consumption per year. Niu et al. [5] performed the capacity matching between the LD and heat pump hybrid air-conditioning system via novel matching indices to system energy stability. Although many heat pump-driven LD systems have been proposed to reduce the energy consumption, some limitations have been observed; the solution cooling and heating loads are kept constant. Thus, the innovative design that can reduce the heat pump size is required to save the heat pump compressor power.

* Corresponding author. Tel.: +82-2-2220-2370; fax: +82-2-1945.
E-mail address: jjwarc@hanyang.ac.kr.

Meanwhile, LD system applied the solution atomization with ultrasound generator have been studied [6,7]. Yang et al. [6] investigated the mass transfer performance of the ultrasonic atomization LD dehumidification system experimentally. They also investigated the regeneration performance of the ultrasonic atomization LD regeneration system [7]. The results indicated that the proposed system could save about 35 – 60% according to the required regeneration rate. Lee and Jeong [8] evaluated the dehumidification and energy performance of the solution atomization-based LD dehumidifier. The solution atomization-based LD dehumidifier could save 17% solution consumption compared with the conventional LD dehumidifier.

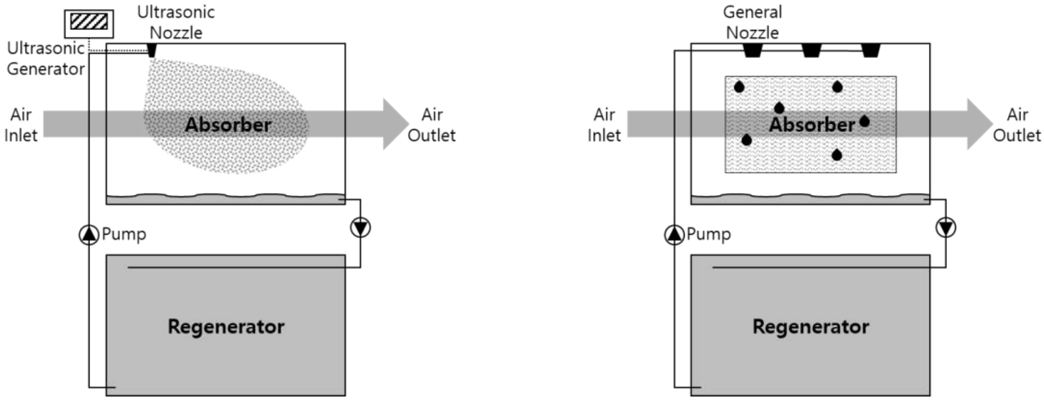
In many researches, the solution atomization-based LD system could reduce the solution cooling and heating load because of the low liquid-to-gas ratio (L/G ratio). However, there are few cases where the solution atomization-based LD system and heat pump are combined and applied as a ventilation or air-conditioning system for buildings. Thus, the heat pump-driven LD system with an ultrasonic atomization was designed and evaluated. The heat pump sizing was performed to apply as a ventilation system via detailed simulation and energy saving potential was also investigated.

2. System Overview

2.1. Liquid desiccant unit

Figure 1 shows the liquid desiccant unit according to the solution atomization methods. Figure 1 (a) is the ultrasonic atomization-based LD unit. The solution was atomized as a fine droplet through the ultrasonic generator and nozzle to increase the heat and mass transfer area between the air and solution by increasing the surface area of the solution droplets. The air and solution contacted directly in the absorber and regenerator, and the dehumidification and regeneration occurred.

Figure 1 (b) is the packbed-based LD units. The solution and air contacted by packing materials such as cellulose type paper. The solution was atomized through the general nozzle onto the packing material. The air was passed through the packing material and dehumidified by contacting with the solution. The packing material was composed of a honeycomb structure to ensure the heat and mass transfer area in limited volume.



(a) Ultrasonic atomized-based (b) Packbed-based

Fig. 1. Schematic of the liquid desiccant unit

2.2. Heat pump-driven liquid desiccant ventilation system

The heat pump-driven liquid desiccant ventilation system consisted of absorber, regenerator, heat pump, indirect evaporative cooler, two fans, and two pump as shown in Figure 2. The outdoor air was introduced to the absorber and dehumidified by the absorber. The air after the absorber passed the primary side of the indirect evaporative cooler and precooled. If the evaporative capacity is remained, the air was additionally cooled by a second evaporator.

Meanwhile, the liquid desiccant solution was circulated from absorber to regenerator. To make the water vapor pressure difference between the air and solution, the solution was cooled and heated before spraying the absorber and regenerator. The heat pump was used for solution cooling and heating concurrently.

The indoor latent load was treated by the heat pump-driven liquid desiccant ventilation system, while the parallel system was required to remove the indoor sensible load. The air-conditioning system based on heat pump cycle was selected as a parallel system.

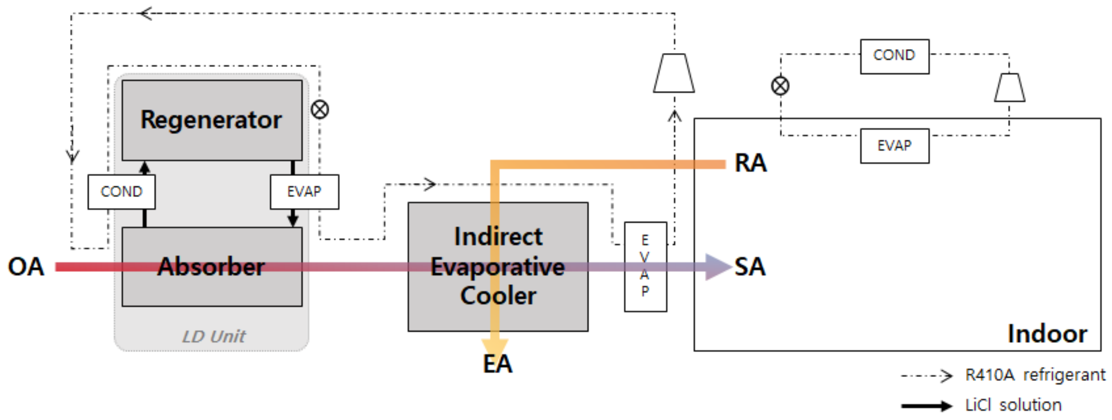


Fig. 2. Schematic of the heat pump-driven liquid desiccant ventilation system

3. Simulation Overview

3.1. Analysis of the liquid desiccant unit

The heat and mass transfer between the air and solution in the counter-flow absorber/regenerator is shown as a one-dimensional problem. Accordingly, the absorber /regenerator model based on the heat and mass conservation relations were expressed using Equations (1) – (5) [9,10]. All the following equations for each differential element are computed numerically using a forward-finite difference algorithm. In the following equations, 101 nodes are segmented along the length of the absorber and regenerator; the unit length (ΔL) is $L/100$. The mass transfer rate was calculated by the number of mass transfer unit (NTU), and the heat transfer rate was also calculated by the NTU and Lewis number (Le). The Lewis number was assumed to be a 0.92 by the heat and mass diffusivity of the air and solution [10].

$$\frac{\omega_a^i-\omega_a^{i+1}}{\Delta L}=\frac{NTU}{L}\times(\omega_{a,in}-\omega_{eq,in}) \tag{1}$$

$$\dot{m}_{a,in}\times(\omega_a^i-\omega_a^{i+1})+\dot{m}_{s,in}\times x_{s,in}\times\left(\frac{1}{x_s^{i+1}}-\frac{1}{x_s^i}\right)=0 \tag{2}$$

$$\dot{m}_s^{i+1}\times x_s^{i+1}=\dot{m}_s^i\times x_s^i \tag{3}$$

$$\frac{h_a^i-h_a^{i+1}}{\Delta L}=\frac{NTU\times Le}{L}\times\left(\left(h_a^{i+1}-h_{eq}^i\right)+\left(\left(\frac{h_{vap,ts}}{Le}-h_{vap,0^\circ\mathrm{C}}\right)\times\left(\omega_{a,in}-\omega_{eq,in}\right)\right)\right) \tag{4}$$

$$\dot{m}_{a,in}\times(h_a^i-h_a^{i+1})+\dot{m}_{s,in}\times x_{s,in}\times\left(\frac{h_s^{i+1}}{x_s^{i+1}}-\frac{h_s^i}{x_s^i}\right)=0 \tag{5}$$

The number of transfer unit (NTU) was estimated by the open literature [11,12] through the mass transfer coefficient, heat and mass transfer area per system unit volume, system volume, and flow rate of the air (Equation (6)). The mass transfer coefficient (k_m) was assumed to be 0.01 kg/m^{2}s, the heat and mass transfer area per system unit volume can be calculated by Equations (7) and (8).

The heat and mass transfer area per system unit volume is difference with the solution atomization and contact methods between the air and solution. The solution was atomized into the absorber and the air and solution contacted directly without the packing materials in the ultrasonic atomized-based LD unit. Thus, the contact area between the air and solution is related to the surface area of the atomized solution with fine droplets, and calculated by solution flow rate and density, droplet size, cross-sectional area of the absorber and regenerator, and relative velocity between the air and solution (Equation (7)). While the solution wetted the packing materials and the air was passing through the packing materials in the packbed-based LD unit. In the packbed-based LD unit, the area was affected by the dimension of the packing material, and converted by channel sizes of the packing material, efficiency of the wetting area, and solution flow rate (Equation (8)).

$$NTU = \frac{k_m \times S_v \times V}{\dot{m}_a} \quad (6)$$

$$S_{v,UADS} = \frac{6 \times \dot{m}_{s,in}}{\rho_{s,in} \times ds \times S_{cross} \times u_{fa}} \quad (7)$$

$$S_{v,packbed} = a \times (1 - 1.203 \left(\frac{u_s^2}{s \times g}\right)^{0.111}) \quad (8)$$

3.2. Heat pump sizing process

The heat pump was used for solution cooling and heating, thus the heat pump should be design with capacity to provide the sufficient cooling and heating for the LD unit. Because the solution heating load is bigger than the cooling load in the LD unit, the heat pump was designed with heating priority. The Figure 3 shows the sizing process of the heat pump. The first step is determining the temperature of the evaporator and condenser to satisfy the target solution temperature for absorber and regenerator. The second step is drawing the pressure-enthalpy diagram of the heat pump cycle through the determined temperature of the evaporator and condenser. The subcooled and superheated degrees were neglected at condenser and evaporator, respectively. The compressor outlet enthalpy of the refrigerant was assumed by the isentropic efficiency of the compressor, and it is assumed to be a 0.75 [13]. The third step is estimating the mass flow rate of the refrigerant to satisfy the solution heating load. Finally, the extra solution cooling or extra evaporating load is determined by comparing the real solution cooling load and ideal evaporator capacity. As presented in Figure 3, if the real solution cooling load is greater than the ideal evaporator capacity, the extra solution cooling is needed. The extra solution cooling load was can be obtained by subtracting ideal evaporator capacity from real solution cooling load. Whereas the real solution cooling load is less than the ideal evaporator capacity, the extra evaporating is required to maintain a stable heat pump cycle. The extra evaporating load can be estimated by subtracting real solution cooling load from the ideal evaporator capacity, and it is used for indoor cooling. The R410A was selected as a refrigerant of the heat pump system.

3.3. Energy simulation

The energy was used for the heat pump-driven liquid desiccant system and indoor parallel system. The LD system consumed the energy in the compressor, fan, and pump. The air-conditioning system with heat pump cycle was selected as an indoor parallel system, and the indoor parallel system consumed the energy in the compressor.

The compressor power for LD system could be calculated by the refrigerant mass flow rate, and enthalpy at the in and out point of the compressor (Equation (9)). The remained indoor parallel system load could be estimated using Equation (10) through the extra evaporating load and indoor sensible load. The energy consumption of the parallel system could be calculated using the system COP by Equation (11). The fan is a variable speed, thus, it was estimated by Equations (12). The pump is a constant speed and the energy consumption was estimated by Equations (13) [14].

$$P_{comp} = \dot{m}_{R410A} \times \left(h_{comp,out} - h_{comp,in}\right) \quad (9)$$

$$\dot{Q}_{parallel,remain} = \dot{Q}_{sen} - \dot{Q}_{extra,evap,load} \quad (10)$$

$$\dot{E}_{parallel} = \frac{\dot{Q}_{parallel,remain}}{COP_{indoor,system}} \quad (11)$$

$$P_{fan} = \frac{\dot{V}_a \times \Delta P}{\eta_{fan}} \times (0.0013 + 0.147PLR + 0.9506PLR^2 - 0.0998PLR^3) \quad (12)$$

$$P_{pump} = \frac{\rho \times g \times \dot{V}_s \times H}{1000 \times \eta_{pump}} \quad (13)$$

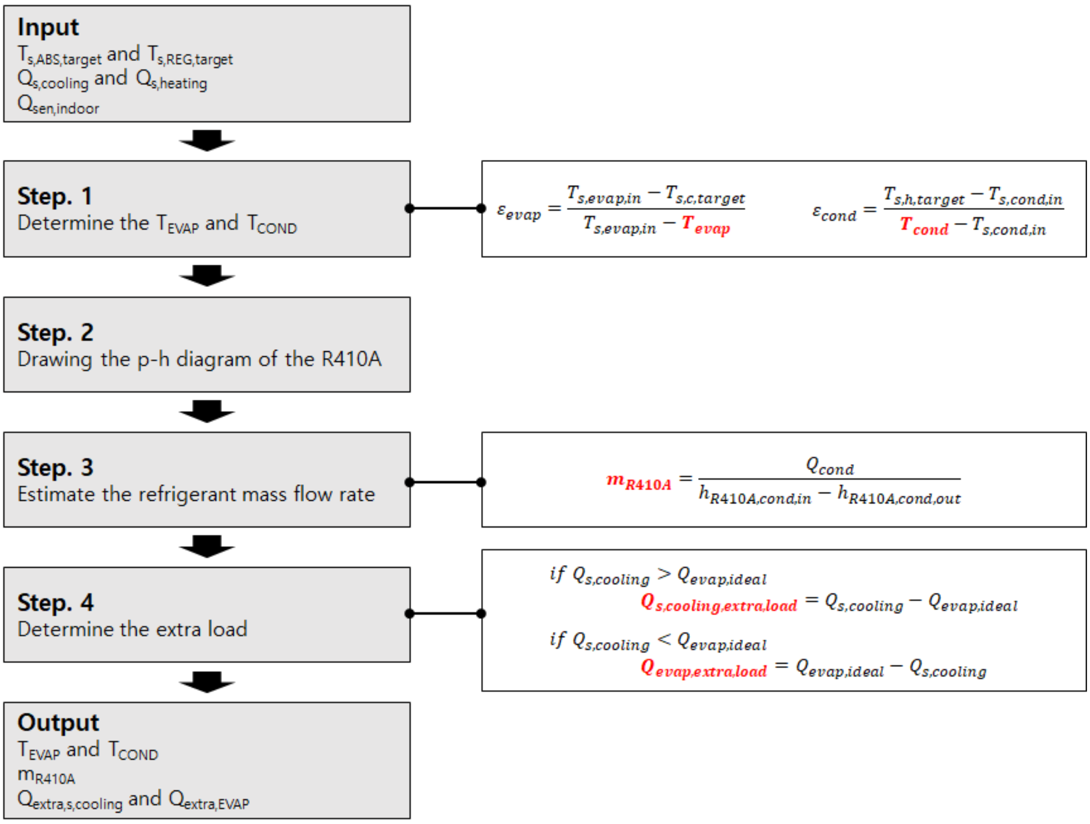


Fig. 3. Flowchart of the heat pump sizing.

4. Results

4.1. Solution cooling and heating load

The solution cooling and heating load was estimated by the solution temperatures at each state of LD unit. Figure 4 shows the maximum solution cooling and heating load comparison of the ultrasonic atomized-based LD unit (i.e., the proposed system) and the packbed-based LD unit (i.e., the reference system).

The proposed system required low solution load than the reference system because the mass flow rate of the solution is different in each system. The proposed system operated 0.4 L/G ratio in the absorber and 0.6 L/G ratio in the regenerator, while the reference system operated 1.0 L/G ratio in the absorber and regenerator under the design condition. Thus, the proposed system needed 0.61 kW and 1.41 kW for solution cooling and heating, whereas the reference system required 1.54 kW and 2.16 kW for cooling and heating, respectively.

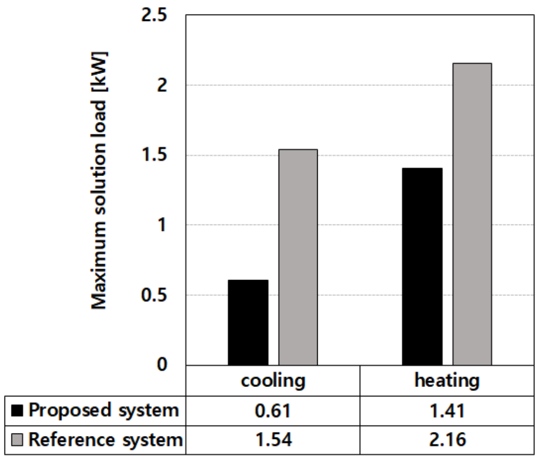


Fig. 4. Maximum solution cooling and heating load in each system.

4.2. Heat pump design

The Figure 5 shows the heat pump cycle for solution cooling and heating. To satisfy the target solution temperature for absorber and regenerator (i.e., absorber target solution temperature = 20℃ and regenerator target solution temperature = 50℃), the evaporator was designed the lower temperature (i.e., about 17℃) than the absorber target and the regenerator was designed the higher temperature (i.e., about 54℃) than the regenerator target. Because the two systems targeted the same temperature for absorber and regenerator, the evaporator and regenerator temperature were almost the same. On the other hand, each system has different in solution cooling and heating load for same target humidity ratio of the supply air because of the L/G ratio. Thus, the mass flow rate of the refrigerant is different. The heating load is greater than cooling load, the extra evaporator load occurred in all summer season. The detailed heat pump sizing results are indicated in Table 1.

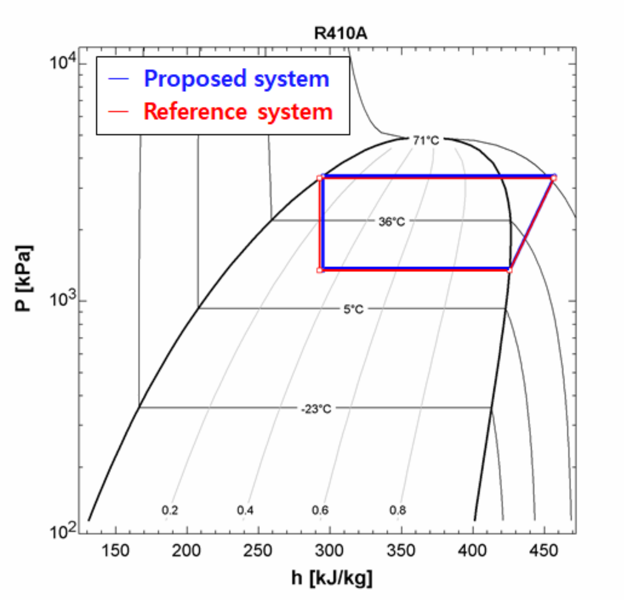


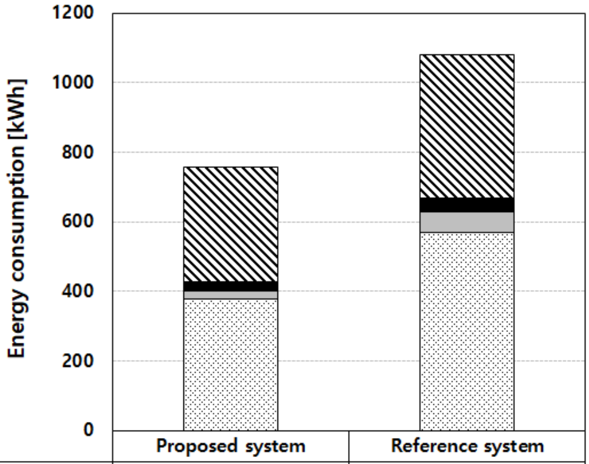
Fig. 5. Pressure-enthalpy diagram of the designed heat pump.

Table 1. Detailed size of the heat pump

	Proposed system	Reference system
Evaporator temperature [℃]	17.8	17.6
Condenser temperature [℃]	54.2	53.2
Refrigerant mass flow rate [kg/s]	0.0087	0.0132
Extra solution cooling load [kW]	0	0
Extra evaporating load [kW]	0.634	0.398

4.3. Energy consumption

The energy consumption of each system during the summer season was shown in Figure 6. The compressor for LD unit power was consumed about 377.1 and 570.3 kWh in the proposed and reference system, respectively. The proposed system used 0.0087 kg/s refrigerant and the reference system used 0.0132 kg/s as indicated in Table 1, thus the compressor power could be reduced in the proposed system because of the low mass flow rate of the refrigerant. Also, because the extra evaporating load in the proposed system is bigger than the reference system, the remained indoor parallel system load is lower in the proposed system. Thus, the proposed system consumed about 330.2 kWh and the reference system consumed 412.3 kWh for indoor parallel system. The fan and pump energy could be reduced in the proposed system even considering PLR (part load ratio) because of the low pressure drop and low solution flow rate in LD unit than the reference system. Finally, the proposed system consumed 757.5 kWh and reference system consumed 1079.7 kWh during the summer season, and the proposed system could save about 30% energy consumption.



	Proposed system	Reference system
Parallel system	330.2	412.3
Pump	26.9	37.8
Fan	23.3	59.3
Ventilation system	377.1	570.3

Fig. 6. Comparison of the total energy consumption during the summer season.

5. Conclusion

This study proposed a heat pump-driven LD ventilation system with a solution atomization technology. The detailed heat pump sizing was conducted to evaluate the energy benefits of the proposed system compared with the conventional LD ventilation system. The total energy consumption during the summer season was predicted.

The results show that the proposed and the reference system had similar heat pump cycle such as temperature of the evaporator and condenser because the target conditions of the LD unit are same. However, the proposed system could reduce the solution cooling and heating load than the reference system. Thus, the mass flow rate of the refrigerant of the heat pump combined with LD unit is difference and the proposed system required less refrigerant. Finally, the total energy consumption during the summer season could be saved about 30% in the heat pump-driven LD ventilation system with ultrasonic atomized compared with the conventional packbed-based LD ventilation system. In the future work, the system will be manufactured and evaluated experimentally.

Acknowledgements

This work was supported by the National Research Foundation of Korea(NRF) grant funded by the Korea government(MSIT) (No. 2022R1A2B5B02001975 and No. 2022R1A4A1026503) and Korea Environment Industry & Technology Institute (KEITI) through Prospective green technology innovation project, funded by Korea Ministry of Environment (MOE) (RE202103243).

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