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Performance analysis of three-stage liquid desiccant deep dehumidification processor driven by heat pump

Zhang Haiqiang Liu Xiaohua Jiang Yi

(School of Architecture, Tsinghua University, Beijing 100084, China)

Abstract: A new type of a heat pump driven three-stage lithium bromide liquid desiccant deep dehumidification processor is presented, which can dehumidify the outdoor humid air to a rather dry state, even when there is no available indoor exhaust air. The test results show that with an outdoor air temperature of 28 to 31 ℃ and an outdoor air humidity ratio of 11 to 14 g/kg, the supply air temperature and the supply air humidity ratio are 1.6 to 2.6 ℃ and 2.6 to 3.0 g/kg, respectively, and the coefficient of performance (COP) of the processor is 1.8. During the test, a liquid pipeline link problem leading to mixture losses of hot and cold liquid desiccants is found. These pipelines are modified. Then, the performance of the modified processor is investigated. And the experimental results show that with an outdoor air temperature of 25 to 32 ℃ and an outdoor air humidity ratio of 18 to 21 g/kg, the supply air temperature and the supply air humidity ratio are 3.2 to 4.0 ℃ and 3.4 to 3.6 g/kg, respectively, and the COP is 2.8. Finally, a mathematical model of the processor is established. The comparison of the simulation results and the test results of the processor exhibits that the pipeline modification improves the performance by about 20%.

Key words: deep dehumidification; liquid desiccant; performance test; air-conditioning

Liquid desiccant air-conditioning systems have been drawing more and more attention due to the advantages of the energy saving potential and benefits for indoor air quality[1-2]. And the systems driven by heat pumps have quickly developed, where the exhaust heat from the condenser is used to regenerate the liquid desiccant and the cooling capacity from the evaporator is used to cool the liquid desiccant. Lazzarin and Castellotti[3] introduced a heat pump liquid desiccant dehumidification processor, manufactured by DryKor Ltd. They found that when the outdoor air temperature was 30 ℃ and the outdoor humidity ratio was 18.7 g/kg, the supply air humidity ratio and the supply air temperature were 8.7 to 11.3 g/kg and 23 to 24 ℃, respectively, and the COP of the processor was 3.2 to 4.2. Liu et al.[4] studied a liquid desiccant outdoor air processor powered by a heat pump with two stages of total heat recovery and one stage of dehumidification and regeneration. The results show that when the outdoor air temperature was 28 to 30 ℃ and the outdoor humidity ratio was 20 to 22 g/kg, the supply air humidity ratio was 11.0 to 11.5 g/kg and the COP of the processor was 6.3 to 7.3. Moreover, they made tests of a heat pump liquid desiccant outdoor air processor with two stages of total heat recovery and two stages of dehumidification and regeneration[5]. With an outdoor air temperature of 36 ℃ and an outdoor humidity ratio of 25.8 g/kg, the supply air temperature and the supply air humidity ratio were 17.4 ℃ and 9.1 g/kg, respectively, and the COP of the processor was 5.0. These researches indicate that the lower the required humidity ratio of the supply air is, the lower the COP of the processor is. And the recovery of the indoor exhaust air can greatly improve the processor's performance.

The previous studies mainly focus on the indoor comfortable air-conditioning environment, with the humidity ratio of the supply air usually over 8 g/kg. Present studies are concentrated on the utilization of industrial air-conditioning, where the required humidity ratio of the supply air is commonly as low as 5 g/kg and the indoor exhaust air can be hardly utilized under many conditions. The principle and the operating mode of a new three-stage liquid desiccant deep dehumidification processor using lithium bromide are proposed in this paper. This processor can dehumidify the outdoor humid air to a rather dry state (lower than 5 g/kg), and it also adopts the outdoor air to regenerate a diluted liquid desiccant.

1 Principle of Deep Dehumidification Processor

The deep dehumidification processor powered by a heat pump includes three heat pump systems and three stages of dehumidification and regeneration, as shown in Fig. 1. In this processor, a stream of outdoor air is dehumidified and cooled stage by stage in three dehumidification modules, and another stream of outdoor air is adopted to regenerate the diluted liquid desiccant in three regeneration modules. The liquid desiccant is LiBr. The regeneration modules A, C, E cooperate with the dehumidification modules B, D, F, respectively. Take the group composed of modules A and B as an example. In module A, the air transfers heat and moisture with the hot and diluted liquid desiccant, and the water in the liquid desiccant evaporates into the air stream; hence, the desiccant is regenerated. In module B, the moisture from the processed air is absorbed by the cool and strong liquid desiccant, and the air is cooled and dehumidified during the process. The solution from the groove of module A is heated by the condenser and then supplied to the top of module A. The regenerated solution exchanges heat in the exchanger with the cool and diluted solution coming from module B, and then it is supplied to module B. Subsequently, the solution from the groove of module B is cooled by the evaporator and it supplies to the top of module B. The sprayed solution absorbs the moisture and becomes diluted. Then, the diluted solution exchanges heat with the hot and strong solution from module A. Thus, a circulation is completed. If the three heat pump systems run

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Biographies: Zhang Haiqiang (1987—), male, graduate; Liu Xiaohua (corresponding author), female, doctor, associate professor, lxh@mail.tsinghua.edu.cn

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together, the evaporative temperatures in modules B, D, F will decline and the condensing temperatures in modules A, C, E will decrease. According to the load ratio, the processor can adjust the operating number of the heat pump systems. Under the full load condition, the three heat pump systems run together, called as “three-stage”. Under the partial load condition, two or one heat pump systems can operate, called as “two-stage” or “one-stage”, respectively. The handling processes of the processed outdoor air as well as the regeneration outdoor air are shown in Fig. 2.

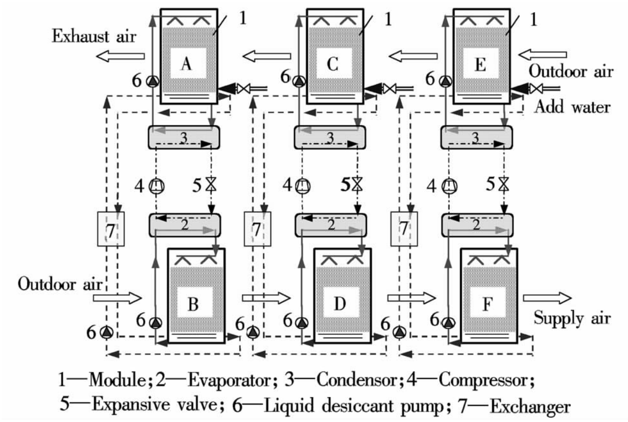

Fig. 1 Operating principle of deep dehumidification processor

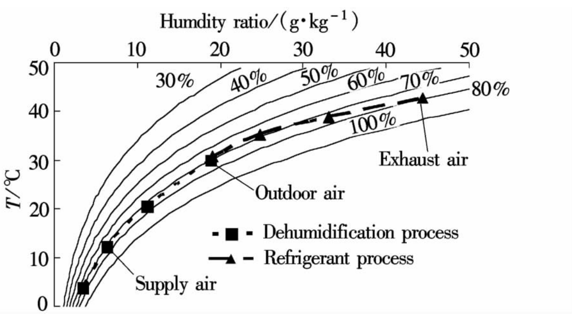

Fig. 2 Air handling process in air psychrometric chart

2 Performance Test

This deep dehumidification processor manufactured by Beijing Sinorefine Co., Ltd. is installed in an explosives factory in Anhui province of China, as shown in Fig. 3. The design temperature and the humidity ratio of the supply air are 14 ℃ and 6.0 g/kg, respectively[6]. Two groups of tests are made. In the first group of tests, the processor is called processor Ⅰ. The performance of processor Ⅰ operating at various stages is tested. The error of the energy balance φ can be calculated by

$$\varphi = \frac{Q_e - (Q + N_p)}{Q_e} \times 100\% \tag{1}$$

where Q_e, Q and N_p are the exhaust energy, the cooling capacity and the power consumption of the processor, respectively. As the detailed data shown in Tab. 1, φ is less than 15%. The mean relative bias difference of the energy balance ψ can be calculated by

$$\psi = \frac{\sum_{i=1}^{n} |\theta_i|}{n}, \quad \theta_i = \frac{D_i - D_i'}{D_i} \times 100\% \tag{2}$$

where θ_i is the relative bias difference of the i-th couple of data. Here, the mean relative bias difference of the energy balance is 3.4%, showing that the test data is valid.

Fig. 3 Photo of liquid desiccant deep dehumidification processor

Tab. 1 Air parameters and COP of processor Ⅰ

State	No.	G_s/(m³·h⁻¹)	G_e/(m³·h⁻¹)	Outdoor air T/℃	Outdoor air d/(g·kg⁻¹)	Supply air T/℃	Supply air d/(g·kg⁻¹)	Exhaust air T/℃	Exhaust air d/(g·kg⁻¹)	Q/kW	N_p/kW	Q_e/kW	N_s/kW	COP
One-stage	1	3 742	3 427	26.2	5.1	12.0	3.5	33.9	14.2	23.0	10.4	32.8	8.6	2.7
	2	3 742	2 801	26.2	5.0	12.4	4.3	35.5	14.9	19.2	9.9	29.7	8.1	2.4
	3	3 742	2 801	26.6	4.9	12.8	4.5	36.6	15.4	18.0	10.0	31.4	8.2	2.2
	4	3 742	3 122	21.0	6.6	12.5	3.7	34.6	13.4	19.7	10.5	30.9	8.7	2.3
	5	3 742	3 122	22.1	6.5	11.9	4.4	35.6	14.1	19.8	9.7	32.1	7.9	2.5
	6	3 742	3 122	21.5	6.2	11.4	4.0	34.1	13.9	19.7	9.8	31.0	8.0	2.5
	7	3 742	3 179	20.1	6.3	11.1	3.8	33.1	13.6	19.2	10.0	31.3	8.2	2.3
Two-stage	1	3 162	2 574	27.4	10.8	7.2	3.7	42.5	32.7	40.7	20.2	61.5	18.6	2.2
	2	3 162	2 574	27.4	10.9	7.4	3.7	42.2	32.2	40.7	19.9	60.2	18.3	2.2
	3	3 162	2 574	27.9	10.7	7.4	3.8	42.4	33.1	40.6	20.0	61.6	18.4	2.2
	4	3 162	2 574	28.0	10.7	7.2	3.8	42.2	32.9	40.9	20.1	60.8	18.5	2.2
Three-stage	1	2 706	2 347	30.5	12.0	1.6	2.8	45.0	42.3	47.7	28.3	73.7	27.1	1.8
	2	2 706	2 347	30.6	11.9	1.8	2.8	45.1	42.4	47.1	28.8	73.5	27.3	1.7
	3	2 706	2 347	30.2	12.3	2.1	2.9	45.0	43.5	47.5	29.4	74.1	27.9	1.7
	4	2 706	2 347	30.0	13.4	2.3	2.9	45.4	44.6	49.6	29.4	76.1	27.9	1.8
	5	2 706	2 347	29.7	13.7	2.6	3.0	45.1	45.6	49.4	29.0	77.9	27.8	1.8

Note: Q is the cooling capacity of the processor and it is equal to the product of the supply air flow rate and the enthalpy difference between the outdoor air and the supply air; Q_e is the exhaust energy of the processor and it is equal to the product of the exhaust air flow rate and the enthalpy difference between the exhaust air and the outdoor air; N_s is the power consumption of the system; G_s is the supply air flow rate; G_e is the exhaust air flow rate; t is the air temperature; d is the air humidity ratio.

In the first group of tests, the exhaust air flow is as much as 80% to 90% of the supply air flow. With the increase in the running stages, the resistance of processor Ⅰ increases and the air flow decreases, as shown in Tab. 1. The COP can be calculated by

$$\mathrm{COP} = \frac{Q}{N_s} \tag{3}$$

The COP of processor Ⅰ calculated by Eq. (3) is 1.8. The temperature and the humidity ratio of the supply air are 1.6 to 2.6 ℃ and 2.6 to 3.0 g/kg, respectively, under the condition that the outdoor air temperature is 28 to 31 ℃ and the outdoor humidity ratio is 11 to 14 g/kg. Under the running condition of the multiple-stage, the stage which is close to the outlet of the supply air has a lower evaporative temperature and energy efficiency in its refrigerant system than the stage which is near the inlet of the outdoor air. So the COP of processor Ⅰ decreases with the increase in the running stages.

A liquid pipeline link problem is found in processor Ⅰ, as shown in Fig. 4(a). The regenerated liquid desiccant flowing out of the exchanger converges with the cool solution from module B and is cooled by the evaporator. At the same time, the diluted solution flowing out of the exchanger converges with the hot solution from module A and is heated by the condenser. The linkage problem can lead to the mixture losses of hot and cold liquid desiccants, which is illustrated in the next section. So some pipelines are modified to make the converged points at the inlets of evaporators and condensers in the new processor which is called processor Ⅱ, as shown in Fig. 4(b), while the group composed of modules C and D and the group composed of modules E and F are the same as shown in Fig. 1.

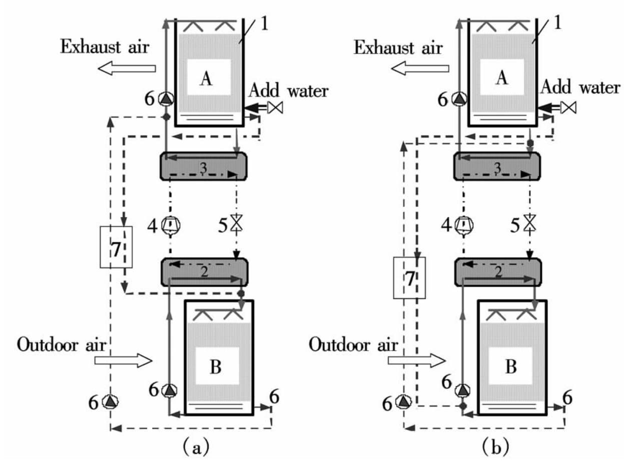

Fig. 4 Liquid pipeline link types of processors. (a) Processor Ⅰ; (b) Processor Ⅱ

The second group of tests are processed after the modification of the pipelines. The air parameters and performance result of processor Ⅱ are shown in Tab. 2. These data show that the error of the energy balance φ is less than +10% and the mean relative bias difference ψ is 6.9%, proving the validation of the test data. With an outdoor air temperature of 25 to 32 ℃ and an outdoor air humidity ratio of 18 to 21 g/kg, the COP of processor Ⅱ is 2.8 and the supply air temperature and the supply air humidity ratio are 3.4 to 3.6 g/kg and 3.2 to 4.0 ℃, respectively.

Tab. 2 Air parameters and COP of processor Ⅱ

State	No.	G_s/ ($m^3 \cdot h^{-1}$)	G_e/ ($m^3 \cdot h^{-1}$)	Outdoor air		Supply air		Exhaust air		Q/kW	N_p/kW	Q_e/kW	N_s/kW	COP
				T/℃	d/ ($g \cdot kg^{-1}$)	T/℃	d/ ($g \cdot kg^{-1}$)	T/℃	d/ ($g \cdot kg^{-1}$)					
One-stage	1	4 209	4 018	30.8	18.3	21.6	11.2	41.4	29.6	38.5	11.8	52.9	9.2	4.2
	2	4 209	4 018	31.4	18.4	22.1	11.5	41.8	30.1	38.3	12.2	53.7	9.7	4.0
	3	4 209	4 018	31.6	18.6	22.3	11.5	42.2	30.6	38.9	12.2	54.5	9.6	4.0
	4	4 209	4 018	31.4	18.9	22.4	11.5	42.2	30.1	39.4	12.3	53.1	9.7	4.0
Two-stage	1	3 952	3 991	30.8	18.6	12.7	6.5	42.3	38.3	65.1	20.5	82.7	18.1	3.6
	2	3 952	3 991	31.3	18.4	12.9	6.6	42.7	38.3	64.3	21.0	82.3	18.5	3.5
	3	3 867	3 755	31.3	18.4	13.2	6.6	43.2	38.2	62.5	21.1	78.0	18.6	3.4
	4	3 867	3 755	30.7	18.7	13.3	6.7	43.9	38.8	62.4	21.7	81.4	19.2	3.2
Three-stage	1	3 325	3 487	30.2	19.1	3.2	3.5	45.5	43.7	74.7	27.9	93.8	26.3	2.8
	2	3 325	3 487	30.5	18.9	3.7	3.5	44.3	42.4	73.5	28.1	92.2	25.9	2.8
	3	3 361	3 523	30.2	18.8	4.0	3.6	44.4	45.2	73.2	27.9	95.8	25.7	2.8
	4	3 361	3 523	31.3	18.5	3.2	3.4	44.9	43.4	75.1	28.8	93.5	26.6	2.8

3 Modelling and Discussion

To analyze the difference in power consumption due to the mixture loss caused by the pipeline link problem, a mathematical model of processor Ⅰ is built according to the method proposed by Liu et al[7]. The results of the first test shown in Tab. 1 are used to validate the model of processor Ⅰ.

As shown in Figs. 5 and 6, the simulation results of the temperature, the humidity ratio, the power of the compressor and the COP of processor Ⅰ are almost consistent with the test results, because their mean relative bias differences are 8.0%, 7.7%, 2.3% and 5.7%, respectively. So the model can be used to analyze the performance of processor Ⅰ.

With the validated model of processor Ⅰ, the performance before and after the modification are compared and the results are listed in Tab. 3. The liquid pipeline link modification has a great effect on the performance of the deep dehumidification processor Ⅰ, improving about 10% to 15% of the cooling capacity and 20% to 25% of the COP whether or not the power of the solution pumps is considered. Take the group composed of modules A and B as an example toexplain the reason. After the pipeline link modification, the liquid desiccant supplied to the top of module B becomes cool and

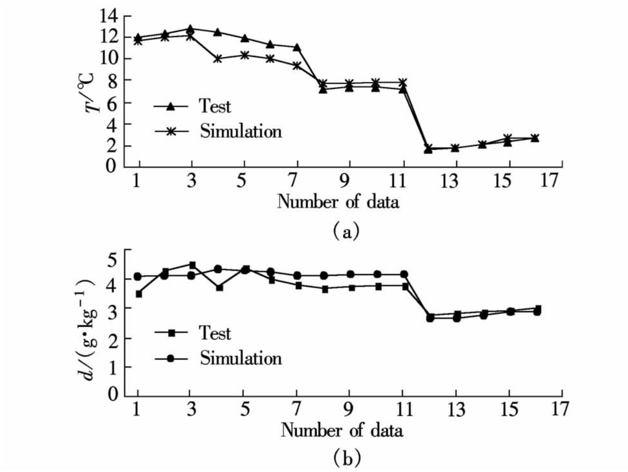

Fig. 5 Parameters of supply air. (a) Temperature; (b) Humidity ratio

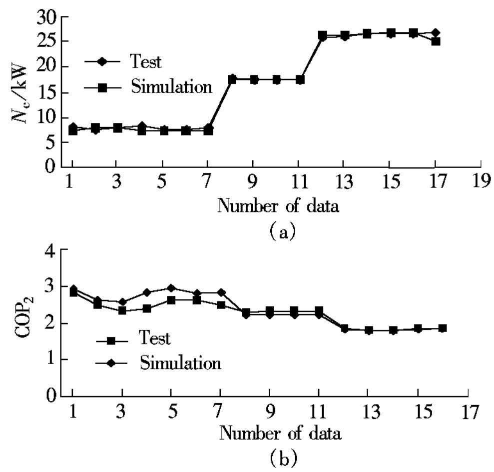

Fig. 6 Parameters of model of processor Ⅰ. (a) Power of compressor; (b) COP_2

Tab. 3 Comparison of processors Ⅰ and Ⅱ

State	No.	Test data of processor Ⅱ: Supply air T/℃	Test data of processor Ⅱ: Supply air d/(g·kg^{-1})	Test data of processor Ⅱ: Q/kW	Test data of processor Ⅱ: N_c/kW	Test data of processor Ⅱ: COP_2	Simulation data of processor Ⅰ: Supply air T/℃	Simulation data of processor Ⅰ: Supply air d/(g·kg^{-1})	Simulation data of processor Ⅰ: Q/kW	Simulation data of processor Ⅰ: N_c/kW	Simulation data of processor Ⅰ: COP_2	P_1/%	P_2/%
One-stage	1	21.6	11.2	38.5	8.81	4.4	23.2	12.0	33.4	9.7	3.5	15.4	26.5
	2	22.1	11.5	38.3	9.18	4.2	23.6	12.2	33.6	9.7	3.5	13.9	20.5
	3	22.3	11.5	38.9	9.17	4.2	23.8	12.4	33.8	9.7	3.5	14.8	22.0
	4	22.4	11.5	39.4	9.25	4.3	23.8	12.5	33.9	9.8	3.5	16.0	22.5
Two-stage	1	12.7	6.5	65.1	17.23	3.8	16.1	7.7	56.6	19.2	2.9	14.9	28.3
	2	12.9	6.6	64.3	17.61	3.7	16.1	7.6	56.5	19.2	2.9	13.8	24.2
	3	13.2	6.6	62.5	17.73	3.5	16.0	7.6	55.8	19.4	2.9	12.0	22.5
	4	13.3	6.7	62.4	18.32	3.4	16.0	7.6	55.8	19.4	2.9	11.7	18.3
Three-stage	1	3.2	3.5	74.7	24.99	3.0	7.9	4.3	66.9	28.1	2.4	11.5	25.2
	2	3.7	3.5	73.5	24.68	3.0	7.8	4.3	66.7	28.1	2.4	10.3	25.5
	3	4.0	3.6	73.2	24.66	3.0	7.9	4.3	66.9	28.0	2.4	9.5	24.5
	4	3.2	3.4	75.1	25.53	2.9	8.0	4.3	67.2	28.1	2.4	11.8	22.9

Note: P_1 and P_2 are the increased percentages of the cooling capacity and the COP, respectively.

the absorptive capacity of moisture is improved. At the same time, the desiccant supplied to the top of module A becomes hot and the regeneration ability is also improved. So the cooling capacity of the processor is enhanced. In addition, the liquid desiccant into the evaporator becomes hotter with the increase in the evaporative temperature, and the liquid desiccant in the condenser becomes cooler with the decrease in the condensing temperature. As a result, the efficiency of the heat pump is improved and the power of the compressor is reduced. The COP of the processor is improved with the increase in the cooling capacity; that is,

$$COP_2 = \frac{Q}{N_c} \tag{4}$$

where N_c is the power consumption of the compressor.

4 Conclusion

This paper presents a new three-stage liquid desiccant deep dehumidification processor, which can dehumidify the outdoor air to a dry state (the humidity ratio is lower than 5 g/kg). In this deep dehumidification processor, three heat pump systems are applied to drive three stages of dehumidification and regeneration, respectively.

A liquid desiccant pipeline link problem leading to the mixture losses of hot and cold liquid desiccants is found in the first test for processor Ⅰ. Some pipelines are modified to make the converged points locate before the inlets of the evaporators and condensers. For processor Ⅱ, with an outdoor air temperature of 25 to 32 ℃ and an outdoor air humidity ratio of 18 to 21 g/kg, the supply air humidity ratio and the supply air temperature are 3.4 to 3.6 g/kg and 3.2 to 4.0 ℃, respectively, and the COP is 2.8. According to the mathematical model, the modification of the pipeline can improve the performance by about 20%.

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热泵驱动的三级溶液深度除湿机组的性能分析

张海强　刘晓华　江　亿

(清华大学建筑学院,北京 100084)

摘要:设计了一种热泵驱动的溴化锂溶液深度除湿机组,该机组适用于无回风可利用、低湿度需求的场合.机组的性能测试结果显示,当室外温度为 28 ~31 ℃,含湿量为 11 ~14 g/kg 时,机组的送风温度为 1.6 ~2.6 ℃,含湿量为 2.6 ~3.0 g/kg,系统 COP 为 1.8.测试时发现了一个造成冷热溶液混合损失的管路链接问题,并对其进行修改.然后,对修改后的新机组进行了性能测试,结果显示,在室外温度为 25 ~32℃,含湿量为 18 ~21 g/kg 时,机组的送风温度为 3.2 ~4.0 ℃,含湿量为 3.4 ~3.6 g/kg,系统 COP 为 2.8.最后,对机组建立数学模型,并将模拟结果与实测数据进行比较,结果表明管路改动使机组性能提升约 20%.

关键词:深度除湿;溶液除湿剂;性能测试;空调
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