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Procedia Engineering 121 (2015) 2224 – 2231

Procedia Engineering

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9th International Symposium on Heating, Ventilation and Air Conditioning (ISHVAC) and the 3rd International Conference on Building Energy and Environment (COBEE)

Optimization of a novel solar heat pipe heat recovery fan

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Abstract

A novel heat pipe recovery fan has been proposed in order to improve the indoor environment, which can be either a perfect supplement to the traditional desiccant technology or an alternative. Heat pipe can be applied to recycle heat and enhance solution dehumidification . This paper presents a new technological process that the heat can be derived from waste water and solar energy can be reserved to use when necessary. By applying the correlated coefficient ,the performance of the system is presented in terms of dehumidification capacity and heat recovery efficiency by simulation.The effect of supplying air flow rate on the airborne pollutant transports are also illustrated based on the new fan with the indoor heat source.The results demonstrate that the best indoor rate for the present system is limited to 0.3ms − 1.In addition,the the liquid to gas flow rate ratios(L/G) has a great influence on dehumidification effectiveness and heat recovery effectiveness.The conclusion can be used as reference information for the optimization of new fan design and application.

Peer-review under responsibility of the organizing committee of ISHVAC-COBEE 2015

Keywords: Optimization; Pollutant; Solar energy; Desiccant heat recovery

1. INTRODUCTION

With the influence of the air quality in recent years,people gradually realize the importance of improving the indoor air, and pay more attention on how to improve the indoor air environment.Associated with the use of air conditioning, however, the energy consumption is increasing. The International Energy Agency has gathered

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Peer-review under responsibility of the organizing committee of ISHVAC-COBEE 2015
doi:10.1016/j.proeng.2015.09.096

Nomenclature

η	efficiency (%)	h	the enthalpy
t	temperature(℃)	w	moisture of air
d	moisture(%)	Q	flux

frightening data on energy consumption trends. The consequences can be obtained from the analysis of the trend of main world energy data from IEA(IEA 2012). Figure 1 shows the industry consumption on the final energy consumption and the rate reaches approximate 40% for some years. One of the consumers of final energy consumption is building(main HVAC) accounts for 27.3%. Compared with other countries, the energy consumption of building in china shows a significant increase with the use of HVAC system.Resolving the contradiction between energy conservation and comfortable indoor environment is an urgent task for researchers.With the development of the heat pipe,the heat pipe is widely used in the industry and energy efficiency sectors. Leonard L. V(2005) studied heat pipes in modern heat exchangers.M.S. Soylemez(2003) presented the optimization of heat pipe heat exchange for waste heat recovery. Liu and Tang (2006) investigated looped separate heat pipe as waste heat recovery facility.Liu and Li (1997) examined application of heat pipe heat exchangers for humidity control in air conditioning systems.The reducing of the energy consumption not only depends on improving the heat transfer efficiency but using sustainable energy, especially solar energy and waste heat energy. It is specifically surprised that the solar thermal has increased so quickly since 2004 to 2014 as shown in Fig2. Many studies have been carried out using renewable energy sources to decrease consumption.

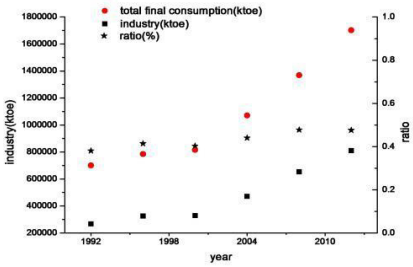


Fig1 Industry energy consumption

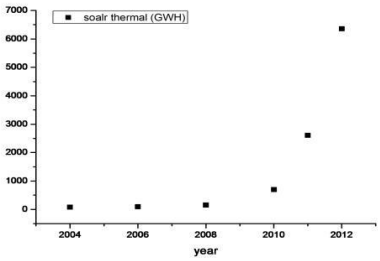


Fig2 Solar thermal

Liquid desiccant systems for cooling and dehumidification have been conducted using solar energy for regeneration (Oberg and Goswami(1998).Gommed and G. Grossman(2002) survey and research a liquid desiccant cooling system using solar energy. Since then, many studies had been conducted in this area.

To the authors' knowledge,there is little information available in literature about the separated type heat pipe heat exchanger applied in air－conditioning dehumidification of waste heat recovery .This paper aims to describe performance in solar dehumidification and heat recovery in air conditioning. A large range of parameters,including inlet air temperature and rate,solution temperature and concentration,are numerically investigated. The spatial distribution of the mass fraction of CH_2O are adopted to evaluate the indoor air quality(IAQ). Finally,the appropriate optimization suggestions have been proposed.

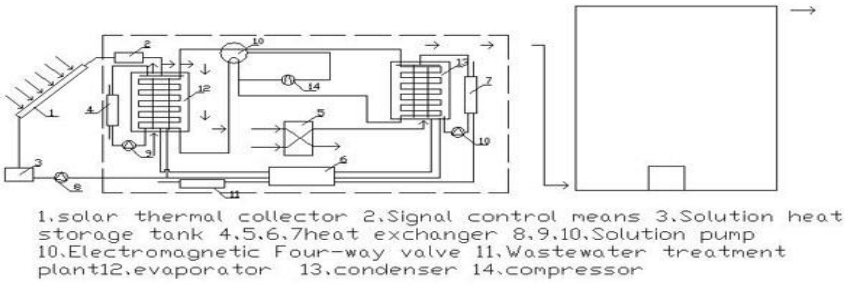


Fig3 Schematic diagram of the process

2. Description of the air conditioner unit

As illustrated in Fig.3, the new fan system(Sanjeev et al 2011)consists of the liquid desiccant system, the regenerator, thermal energy storage device,heat exchange and some auxiliary facilities, such as pump, valve. It is noteworthy that capillary backflow of working in heat pipe make device used in winter and summer with a four－way valve. Based on the ability of absorption and adsorption,heat and mass transfer between the fresh air and solution and working medium can completed in the heat pipe dehumidification unit(HPDU).Energy from solar thermal collector can been reserved in the solution tank,which balanced performance .In the hybrid system, separate type heat exchanger of heat pipe plays the role of waste heat recovery. More importantly ,it can significantly reduce costs by using solar energy and waste heat,In fact, solar energy and waste heat have been used in the device for heat recovery.

The efficiency of desiccant system depends strongly on the sensible heat ratio(SHR) and the evalution of the pollutant in the room.The heat and mass transfer between return air and fresh air has occurred in heat exchanger The SHR(AOU and WANG 2006)is defined as the follow:

$$\eta_t = \frac{t_{oa} - t_{mix}}{t_{oa} - t_{sa}} \times 100\% \tag{1}$$

$$\eta_d = \frac{d_{oa} - d_{sa}}{d_{oa} - d_{ra}} \times 100\% \tag{2}$$

Where ;

η_t , η_d thermal efficiency or dehumidification efficiency (%) －during state ,

t_{oa}: temperature(℃) for the fresh air at the inlet of the heat exchanger

t_{sa}: temperature (℃) for the supply air at the outlet of the new fan

t_{mix}: temperature(℃) for the supply air at the inlet of the new fan

d_{oa}: moisture(%) for the fresh air at the inlet of the heat exchanger

d_{sa}: moisture(%) for the supply air at the outlet of the new fan

d_{ra} : moisture(%) for the exhaust air at the outlet of the heat exchanger

the performance of the device can be described by ηt , ηd .The fresh air flows through the dehumidifier heat recovery unit,and then to the room by fan.

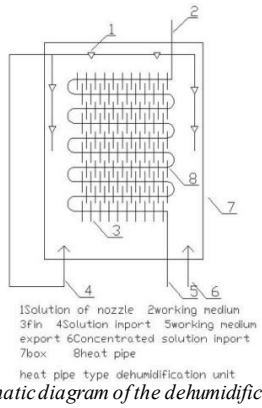


Fig4 Schematic diagram of the dehumidification unit

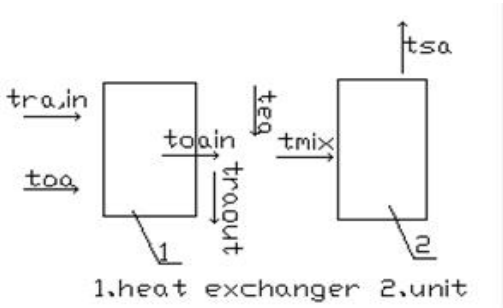


Fig5 Schematic diagram for heat recovery

2.1 Operating principle

Based on the ability of moisture absorption and vapour pressure deficit(VPD), desiccant can be completed in the HPDU. Figure 4 shows the schematic diagram of heat pipe utilization on liquid desiccant system.There exist two fluid cycles, namely,liquid cycle in HPDU.which is for dehumidifying ability. The other one is fluid cycle in heat pipe from evaporator to condenser. These two cycle enhance the heat and mass transfer efficiency and heat recovery by increasing the contact finned tube and the air steam. Heat pipe has good performance of heat recovery. At the same time fin increases the heat exchange area. Solution sprayed from the upper and both sides expand contact avoiding incomplete contact. Heat pipe evaporation and condensation correspond to HPDU, The heat and mass transfer process of the new fan in winter is similar to that in summer.A director valve can change direction. Fluid (water) in heat pipe run by capillary action .

2.2 Simulation model of Liquid desiccant units

According to the literature(ALIZADEH and SAMAN 2001； Luo et al2014;Liu 2009),The governing equations describing the flow within the unit are the simultaneous steady state heat and mass transfer and continuity equations.The following assumptions have been made for the analysis:

1 the temperature gradient between the fin and the working medium is negligibly.

2 the solution which sprays from the top and the surrounding is uniformity,Besides, it is uniformity that the solution are contact with the inlet air.

3 there is no heat and mass to the heat recovery unit and surrounding.

With this assumptions, some governing equations can be put forward for the control volume shown(Khan and Ball 1992 ； Liu 2014) as follow :

$$m_p \frac{dw_p}{dx} - \frac{dm_s}{dy} = 0 \tag{3}$$

$$m_p \frac{dhp}{dx} + \frac{dh_s m_s}{dy} + Q = 0 \tag{4}$$

$$m_a c_a \frac{dT_a}{dx} = \alpha A (T_S - T_p) \tag{5}$$

Where hp and wp are the enthalpy and moisture of air.ms and hs stand for the solution .Q is the heat remove of the cooling water.

a Boundary conditions

Liquid desiccant dehumidification (Liu and Li 2014) can give guidance to set up the parameters of the inlet. In general, the inlet and outlet of air,solution or working medium have been known. It can be shown below.

x=0 y=0, t=tin w=win m=mn (for air solution and water)

b solving method

The above governing equations are discretized by finite volume method(FVM) based on CV – face center(Patankar 1980;Tao 2002). The convection terms can simplify under center difference scheme(CDS)(Peric 2002).The SIMPLE algorithm is used to deal with the pressure and the velocity. Some good examples can used for reference.At last the all residual of the temperature velocity species should be less than 10 – 5 .

3. Air flow simulation

The serial numbers of fresh air and exhaust air in the heat recovery can be seen in Figure 5 where oa and ra stand for fresh air and exhaust air,respectively.We take a summer of Qing dao as an example and the place is in workshop of China University of Petroleum(UPC).

Energy balance of the device about the air and exhaustingair can be performed below(Liu and Zhao 2012).we can define the sensible heat efficiency as recalling the definition of fresh air ratio,r=Qra/Qsup ,Reference from literature (Liu2012) . assuming constant heat capacity for air in the range of the studied temperatures, the following relation can be obtained

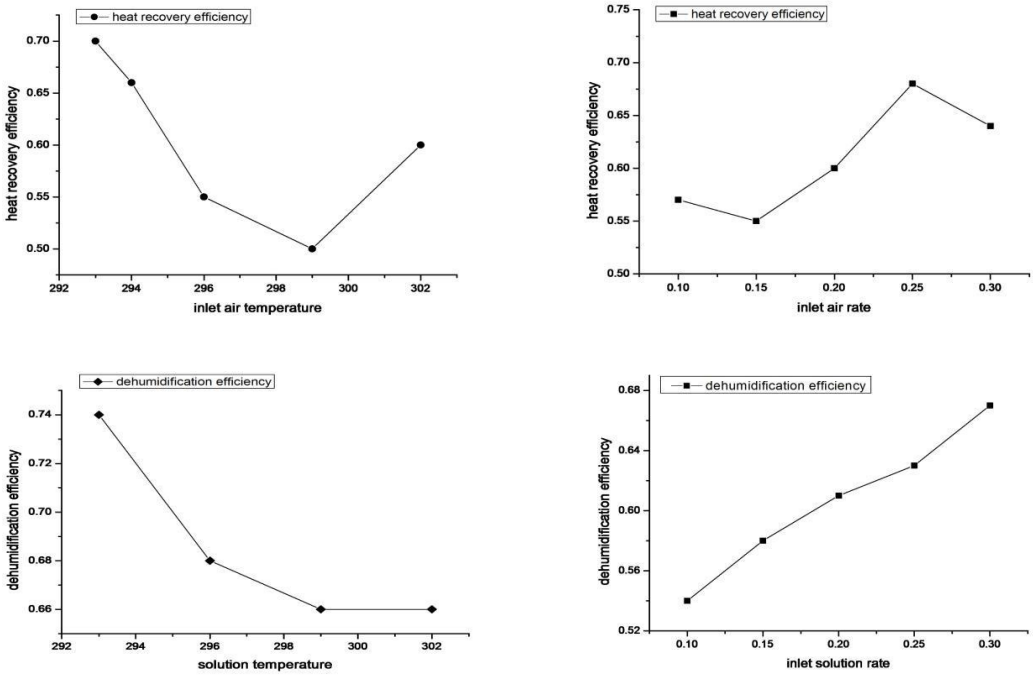
$$Q_{oa} c_p(t_{oa} - t_{oain})=Q_{ra}c_p(t_{raout} - t_{rain}) \quad (6)$$

$$Q_{oa} c_p(t_{oa,out} - t_{ref})+Q_{ra}c_p(t_{ea} - t_{ref})=(Q_{oa}+Q_{ea})c_p(t_{mix} - t_{ref}) \quad (7)$$

$$Q_{sup}=Q_{ra}+Q_{oa} \quad (8)$$

When Q_{sup} stands for mixture air ,mix stand for mixture air. $t_{ra,in}$ $t_{ra,out}$ for the exhaust air at the inlet or outlet of the heat exchanger. t_{oain} t_{ea} for the fresh air and recirculating air at the inlet of the unit. The last of the loading(refrigeration capacity)formula can be obtained

$$Q_{loading} =(Q_{oa}+Q_{ra}) \rho c_p r(1 - \varepsilon)(t_{mix} - t_{sa}) =Q_{sup} \rho c_p r(1 - \varepsilon) \triangle t \quad (9)$$



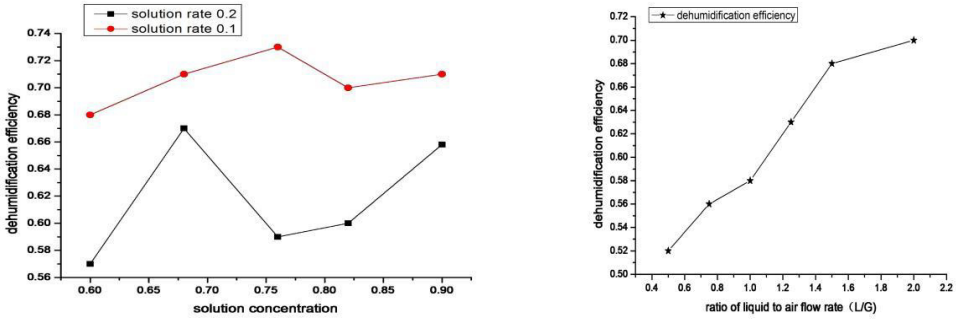


Fig 6 Effct of inlet parameters on the heat recovery and dehumidification efficiency

When Qsup stand for mixture air ,mix stand for mixture air. tra,in tra,out for the exhaust air at the inlet or outlet of the heat exchanger. toain tea for the fresh air and recirculating air at the inlet of the unit. The last of the loading(refrigeration capacity)formula can be obtained

$$Q_{loading} = (Q_{oa}+Q_{ra})\ \rho\ c_p r(1-\varepsilon)(t_{mix}-t_{sa}) = Q_{sup}\ \rho\ c_p r(1-\varepsilon)\ \triangle\ t \quad (10)$$

4. Result and discussion

The effect of several of the operating parameters on the performance of the new fan unit can be considered, Some results are presented below:

4.1 Result analysis

A comparative parametric analysis of dehumidification and heat recovery was carried out for this new fan.Some results have been obtained from Fig 5 and Fig 7.

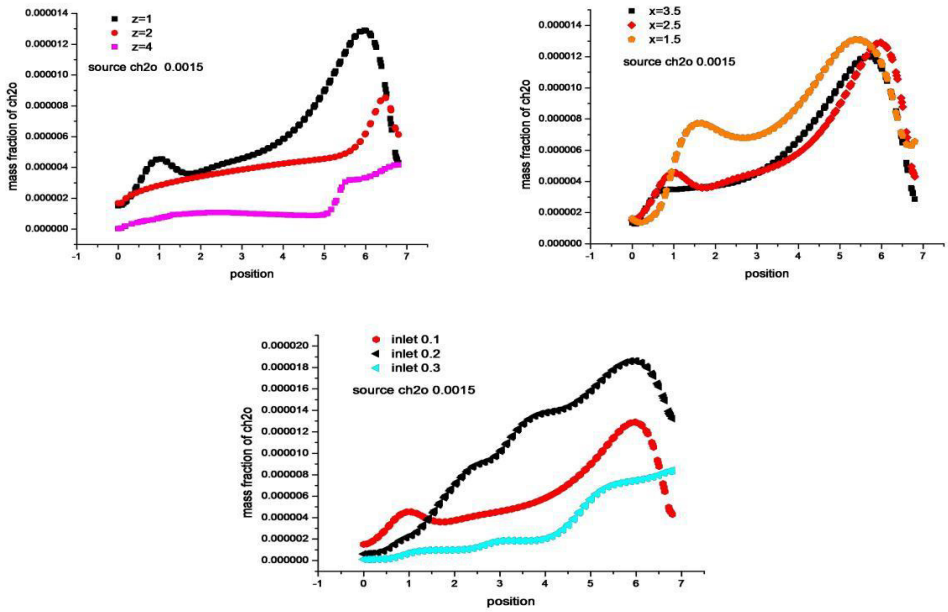


Fig7 Comparison with various height,inlet rate and horizontal position

a heat recovery unit

Fig 6 show the influence of the inlet parameters of air on the heat recovery.For the inlet air section,the inlet velocity is in a range of 0.1ms − 1 − 0.3 ms − 1,and the inlet temperature is 292k − 302k.As the inlet air rate increases,the heat recovery efficiency increases to 0.7 and then gradually tapers off. In contrast,the heat recovery efficiency decreases substantially as the temperature increases from 292k − 302k with the same heat recovery efficiency.It is a fact that the efficiency increases when supply air temperature difference is at least as big.

b evaluate the indoor air quality

Diffusion of the various conditions of mass fraction of CH2O is shown in Fig 7. From the Simulation results, we can see that, concentration decreases with height and the high concentration influenced by wind pressure airflow appears at the outlet.

4.2 Optimization of the unit

1 Supply air temperature

Qloading = Qsup ρ cpr(1 − ε)△t , △t (supply air temperature difference)is a influence argument,which is one of main deciding economy of air conditioning(Huang 2013) .Based on the premises of the established technical requirements,increasing supply air temperature has a prominent economic significance.When supply air temperature doubles,air volume can be reduced by half. As a result,material loss system and investment have about fourteen percentage reduction. In the air conditioning design,how to control the supply air temperature is an important issue.Places which need strict temperature control can get a little supply air temperature,like the hospital,otherwise,it can take opposite. The different way of supply air has a disparate result. To conclude,in the design of air conditioning it should consider various factors in selecting the optimal temperature difference. A suitable proposal is to maintain supply air temperature between 5℃ to 8℃, which is on the foundation of energy onsumption and comfort. Of course, some special circumstances required other consideration

2 air inlet rate

Fig 6 and Fig 7. show the heat recovery efficiency and the diffusion of mass fraction of CH2O. A suitable proposal about the inlet air rate ranges from 0.1ms − 1 to 0.2ms − 1 .The aim is to get the low concentration and the high efficiency.

3 design of unit

Every evaporator and condenser constitute some thermal tubes closed by two aluminum plate and four aluminum strips as plate − fin heat exchanger(Liu 2005). It is worth mentioning that reversing valve changes the use in the winter and summer and fins staggered arrangement not only increase the contact solution with air also enhance the heat transfer the working medium and air.As the research shown from Lou(2014),it is recommended to set this size around 0.3m (Lou and Shao 2014) for optimization but need further validation before their application in the daily practice.Different heat exchanger have different effects.

5. conclusion

A design of new fan was developed in this study to predict the new fan performance of a heat recovery and heat pipe type cold dehumidification unit. Meanwhile,increasing flow rate can reduce indoor pollutant level.from the performance predictions presented,this design shows a very good agreement with some experimental data available in the literature.

1. Simulation shows the liquid to gas flow rate ratios(L/G)(Sanjeev Jain 2007) have a great influence on dehumidification effectiveness and heat recovery effectiveness.When the L/G exceed three , efficiency curve gradually flat out.

2. The influence of solution temperature on the efficiency is not be greater than the speed of the solution.It is interesting to note that a decrease in the speed of the solution increases the predicted effectiveness,which is conspicuous to temperature.

3. Based on consideration of indoor comfort,referring to the design standard for energy efficiency of public buildings and civil heating, ventilation and air conditioning design,the indoor air rate is limited to 0.3, supply air temperature difference is limited from five to ten degree.

6.Acknowledgement

The authors gratefully acknowledge the financial support of National Natural Science Foundation of China, (NSFC, Grant No. 51208192 and 51306210, Instability theory and inverse convection design of air flow patterns in the large space; NSFC, Grant No. 51304233, Multi - physics diffusion of leaked natural gas and backward time inverse identification of leakage sources), Hong Kong Scholar Program (XJ2013042), China Postdoctoral Science Foundation (2014M560593), Qingdao Postdoctoral Science Foundation, and Fundamental Research Funding Programme for National Key Universities in China(14CX02103A).

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