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Exergy analysis of a solar regenerated liquid desiccant assisted air conditioning system for hot and humid climates

Exergy analysis of a solar regenerated liquid desiccant assisted air conditioning system for hot and humid climates

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Abstract. In this paper, exergy analysis of a novel solar powered liquid desiccant assisted air conditioning system is presented and simulated. The system aims to provide suitable thermal comfort conditions inside large office buildings with high internal loads situated in the hot and humid tropical/subtropical countries of the world. The system consists of process and regenerating air streams, a liquid desiccant solution loop and a cooling water loop. The primary objective of this study is to present the exergy of cooling capacity along with the overall exergy efficiency of the proposed system. The study helps to quantify the optimum operating and design parameters for system operation based on the second law of thermodynamics. For the base case, which is representative of a hot and humid climatic condition, the proposed system is able to maintain the room air conditions within the moderate thermal acceptability criterion. The exergy of cooling capacity and exergy efficiency for the base case is about 2900 W and 2 % respectively. Parametric analyses show that the system performs the best under conditions of high ambient insolation and temperature, low ambient humidity and a process air to desiccant solution mass flow rate of about 3 in the dehumidifier.

1. Introduction
In the modern world, building energy consumption is significantly augmented by the use of air conditioning systems, thereby placing a huge demand on the electricity supply network. A 6.2 % per year rise is estimated in the heating, ventilation and air conditioning load of the world [1]. Conventional vapor-compression based air conditioning systems are still the most prevalent technology in use for providing thermal comfort inside commercial and office buildings situated in the hot and humid climatic regions. However, they are associated with large primary energy consumption and global warming. This gives rise to a pressing need to develop air conditioning systems which can provide suitable thermal comfort in buildings without harming the environment.

Liquid desiccant cooling systems (LDCS) are a viable substitute to conventional vapor compression based air conditioning systems. They have the ability to treat latent and sensible heat loads independently, are easy to store and consume much less energy compared to that of vapor compression based systems. Regeneration of the liquid desiccant solution (LDS) by making use of the heat energy obtained from solar radiation has gained momentum recently owing to the system's effectual usage of low grade heat and coincidental matching of peak cooling load with maximum solar insolation.

Previous research on LDCS have been mainly focused on energy analysis, with studies based on second law of thermodynamics being limited [2]. Of the studies that are available on exergy analysis, such a novel configuration with the use of direct solar regenerator and dew point indirect evaporative cooler has not been previously reported in literature. Thus, this paper attempts to bridge that gap by carrying out both energy and exergy analysis of the proposed solar powered liquid desiccant air conditioning system for application in large office buildings with high internal loads situated in the hot and humid climatic regions. The present work focuses exclusively on designing the proposed system optimally on the basis of the second thermodynamic law.

2. Methodology

2.1. Description of the system

The scheme of the proposed system is shown in figure 1. The system is composed of the process and regenerating air streams, a LDS loop and cooling water loop. An aqueous solution of $CaCl_2$ is used as the desiccant.

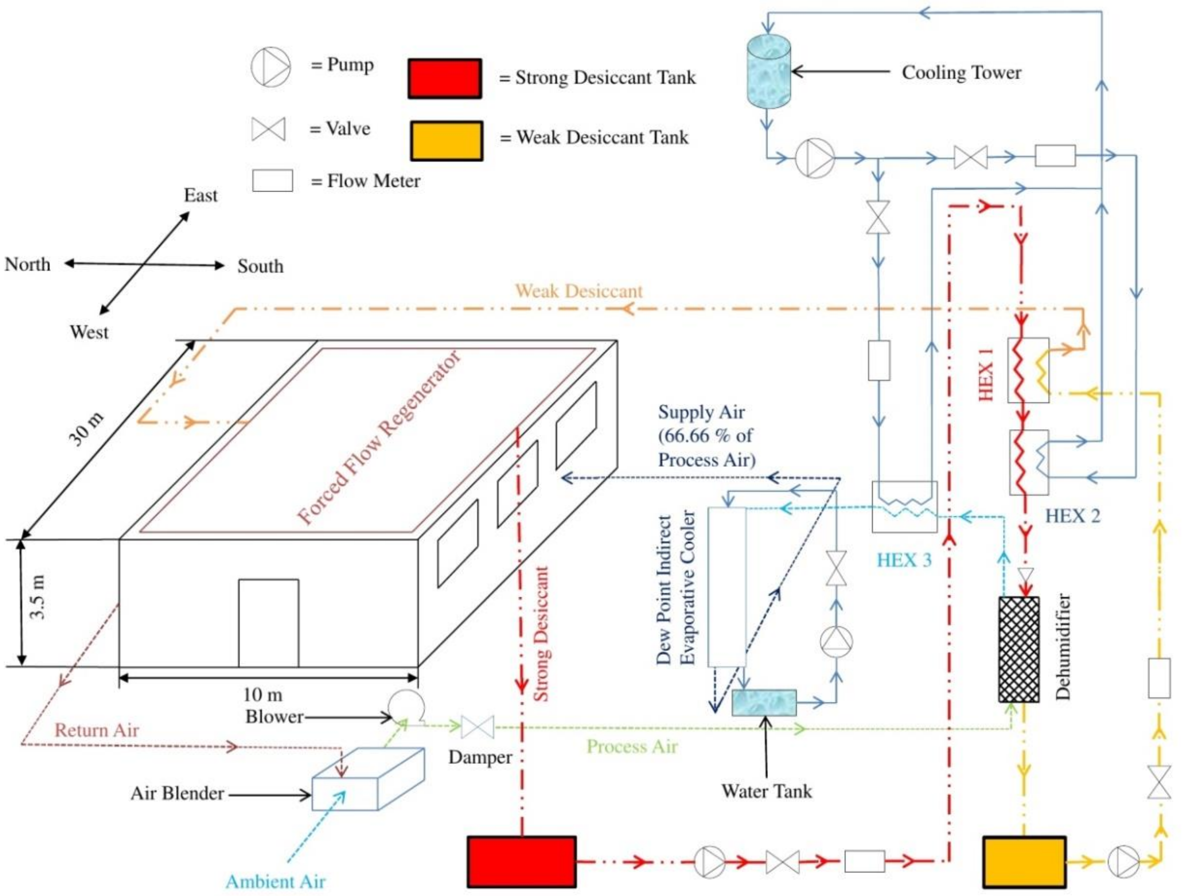

Figure 1. Scheme of the proposed system.

The process air is a mixture of ambient and conditioned return air from the room, mixed in the air blender. The process air is dehumidified in a dehumidifier (DEH), cooled sensibly in the air-water heat exchanger (HEX 3) and dew point indirect evaporative cooler (DPIEC) and finally supplied to the room. The DPIEC uses a fraction of the intake air as an evaporative sink and supplies the rest to the building [3]. This supply air aims to maintain suitable thermal comfort conditions inside the room with high internal load [4]. Finally, the cycle is completed by bringing the conditioned air inside the room back to the air blender. It may be noted that the amount of air used as an evaporative sink in the DPIEC is equal to the amount of ambient air at the air blender inlet. The regenerating air stream consists of ambient air which is drawn into the regenerator (REG) to absorb moisture from the dilute LDS in order to re-concentrate it. Finally, this REG outlet air is exhausted to the atmosphere.

The LDS after dehumidifying the process air becomes diluted due to the absorption of moisture and heats up due to the released latent heat of condensation. It is then preheated in the desiccant-desiccant heat exchanger (HEX 1) before being supplied to the forced parallel flow direct solar REG [5] where it loses moisture to the regenerating air stream. Thereafter, the hot concentrated LDS at the REG outlet is pre-cooled in the HEX 1 and desiccant-water heat exchanger (HEX 2) before being supplied to the DEH. Cooling water to the HEX 2 and HEX 3 is supplied from a natural draught wet cooling tower.

2.2. Thermodynamic modelling

The DEH is modeled based on the works of Gandhidasan [6] and Chung [7]. The governing equations are given in equations (1) through (5).

$$t_{a,i,deh} = t_{a,ret} \times f + t_{a,amb} \times (1-f) \tag{1}$$

$$H_{a,i,deh} = H_{a,ret} \times f + H_{a,amb} \times (1-f) \tag{2}$$

$$m_{da,i,deh} \times h_{a,i,deh} + m_{s,i,deh} \times h_{s,i,deh} = m_{da,o,deh} \times h_{a,o,deh} + m_{s,o,deh} \times h_{s,o,deh} \tag{3}$$

$$\varepsilon_{deh} = \frac{1-\dfrac{[0.205\times(\dfrac{m_{a,i,deh}}{m_{s,i,deh}})^{0.174}\exp(0.985\times\dfrac{t_{a,i,deh}}{t_{s,i,deh}})]}{(a_p Z)^{0.184}\pi^{1.68}}}{1-\dfrac{[0.152\times\exp(-0.686\times\dfrac{t_{a,i,deh}}{t_{s,i,deh}})]}{\pi^{3.388}}} = \frac{p_{a,i,deh}-p_{a,o,deh}}{p_{a,i,deh}-p_{s,i,deh}} \tag{4}$$

$$\beta = \frac{t_{a,i,deh}-t_{a,o,deh}}{t_{a,i,deh}-t_{s,i,deh}} \tag{5}$$

where, t, f, H, m, h, ε_{deh}, a_p, Z, π, p_a and p_s refer to the temperature (°C), fraction of return air in process air, specific humidity, mass flow rate (kg/s), specific enthalpy (J/kg), effectiveness of DEH, specific area of packing (m^2/m^3), packed bed height (m), ratio of vapor pressure depression of the LDS to the vapor pressure of pure water, partial pressure of water vapor in the air (Pa) and vapor pressure of the LDS respectively.

The REG is modeled based on the work of Alizadeh and Saman [8]. The governing equations are given in equations (6) through (12).

$$I_{reg} = I_{amb} \times (1-\rho_s) \times (\tau\alpha) \tag{6}$$

$$I_{reg}x - U_{reg}(t_s - t_{amb})x - m_{a,i,reg}C_{pa}(t_a - t_{a,i,reg}) - m_s C_{ps}(t_s - t_{s,i,reg}) - m_{evap}h_{fg} = 0 \tag{7}$$

$$I_{reg}dx - U_{reg}(t_s - t_{amb})dx - m_{a,i,reg}C_{pa}dt_a - m_s C_{ps}dt_s - dm_{evap}h_{fg} = 0 \tag{8}$$

$$m' = \frac{dm_{evap}}{dx} = \frac{0.622 \times hc_{s-a}}{950 \times P_{atm}} \times (p_s - p_a) \tag{9}$$

$$p_s = a + bt_s + c / C_s \tag{10}$$

$$hc_{s-a}(t_s - t_a)dx = m_{a,i,reg} C_{pa} dt_a + hc_{a-amb}(t_a - t_{amb})dx \tag{11}$$

$$\frac{1}{hc_{a-amb}} = \frac{1}{hc_{a-g}} + \frac{1}{hc_{g-amb}} \tag{12}$$

where, I, ρ_s, $(\tau\alpha)$, x, U, C_p, h_{fg}, hc, C_s refer to the solar insolation (W/m^2), reflectivity of the LDS, transmissivity-absorptivity product, length (m), overall heat loss coefficient (W/m^2°C), specific heat capacity (J/ kg °C), heat of evaporation of water from LDS (J/kg), convective heat transfer coefficient (W/m^2°C) and mass concentration of the LDS (%) respectively.

The effectiveness of the heat exchangers and the wet bulb effectiveness of the DPIEC are defined as follows.

$$\varepsilon_{HEX1} = \frac{t_{s,i,reg} - t_{s,o,deh}}{t_{s,o,reg} - t_{s,o,deh}} \tag{13}$$

$$\varepsilon_{HEX2} = \frac{t_{s,i,HEX2} - t_{s,i,deh}}{t_{s,i,HEX2} - t_{w,o,ct}} \tag{14}$$

$$\varepsilon_{HEX3} = \frac{t_{a,o,deh} - t_{a,o,HEX3}}{t_{a,o,deh} - t_{w,o,ct}} \tag{15}$$

$$\varepsilon_{DPIEC} = \frac{t_{a,o,HEX3} - t_{a,i,room}}{t_{a,o,HEX3} - t_{wb,a,o,HEX3}} \tag{16}$$

The Merkel approach [9] is used to calculate the water temperature at the cooling tower outlet. By solving equations (1) through (16), the conditions of supply air at the inlet to the room are found out. The heat load calculation for the office building is done by assuming a sensible heat factor of 0.92 using the approach provided in [4,10].

For the exergy analysis, the saturated state of the ambient atmosphere is selected as the reference state [2]. The specific exergy of moist air (J/kg of dry air) is expressed as:

$$e_{da} = (C_{pa} + wC_{pv})T_{amb}\left[\frac{T_a}{T_{amb}} - 1 - \ln\left(\frac{T_a}{T_{amb}}\right)\right] + R_a T_{amb}\left[(1+1.608w) \times \ln\frac{(1+1.608w_{sat})}{(1+1.608w)} + 1.608w \times \ln\left(\frac{w}{w_{sat}}\right)\right] \tag{17}$$

where, w, T, R_a refer to the humidity ratio, temperature (K) and ideal gas constant for dry air (J/kg K) respectively.

The exergy of cooling capacity (W) is expressed as [11]:

$$E_{CC} = m_{da,i,room} \times (e_{da,i,room} - e_{da,i,deh}) \tag{18}$$

The system's exergy efficiency is expressed as the ratio of the exergy of cooling capacity to the absorbed solar radiation exergy rate by the REG (W) [12]. Analyses based on exergy efficiency take into account the energy quality and reversible (idealized) version of the system which the first law analyses miss out on. It is an important tool which can be used to quantify the irreversibilities within the system.

$$\eta_{ex} = \frac{E_{CC}}{I_{amb} \times (1-\rho_s) \times (\tau\alpha) \times A_{reg} \times (1 - \frac{T_{amb}}{T_{apparent,sun}})} \tag{19}$$

A computer code is written in Java to simulate the proposed thermodynamic model. It takes the ambient insolation, temperature, humidity, wind speed and radiation tilt factor as inputs and predicts the room air conditions along with the system's exergy of cooling capacity and exergy efficiency. The place under consideration in this study is Kolkata [13], India which is representative of a location having hot and humid ambient climate.

3. Results and discussions

The validation of the proposed thermal model is carried out by comparing the results predicted by our model with a similar reference model study [14]. Figure 2 shows the comparison between the present and reference model for a typical day in August (16th of August). We can observe that the proposed model agrees well with the reference model, having mean absolute errors of 3.2 % and 6.5 % on comparing the room supply air temperature and humidity ratio respectively, for system operation in complete ventilation mode.

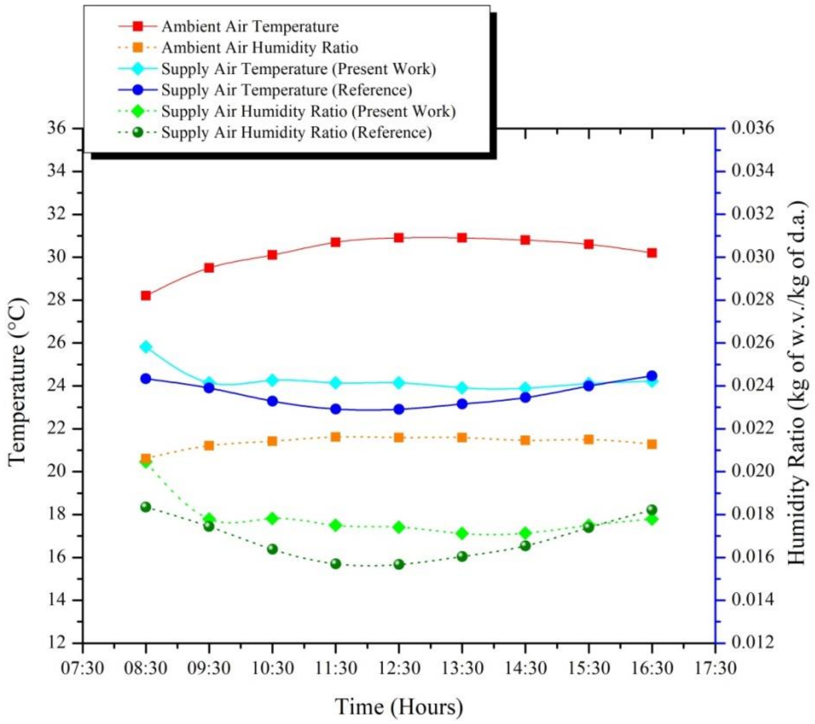

Figure 2. Comparison of the proposed model with the reference model study for a typical day in August.

The operating and design conditions for operation of the re-circulating air system for a base case, which is typical of the climatic condition for a hot and humid environment, is provided in Table 1. It can be seen from the simulations that the room temperature and humidity ratio is maintained at 24.3 °C and 9 g/kg respectively, which is suitable for moderate thermal comfort condition. The system's exergy of cooling capacity (E_{CC}) and exergy efficiency (η_{ex}) is about 2900 W and 2 % respectively. The low value of exergy efficiency is due to the large area of the regenerator along with moderately high ambient insolation which increases the absorbed solar radiation exergy rate, and low value of supply air flow rate into the room (or process air flow rate at DEH inlet) which decreases the exergy of cooling capacity. For the parametric analyses, individual parameters are varied keeping others the same to estimate the influence of the varied parameters on the second law performance of our system.

Table 1. Operating and design conditions for the base case.

Parameter	Unit	Value
Ambient temperature	°C	34
Ambient relative humidity	%	75
Ambient insolation	W/m^2	700
Ambient wind speed	m/s	2.5
Wall radiation tilt factor		0.5
Desiccant flow rate	kg/s	1.875
Process air flow rate	kg/s	3.75

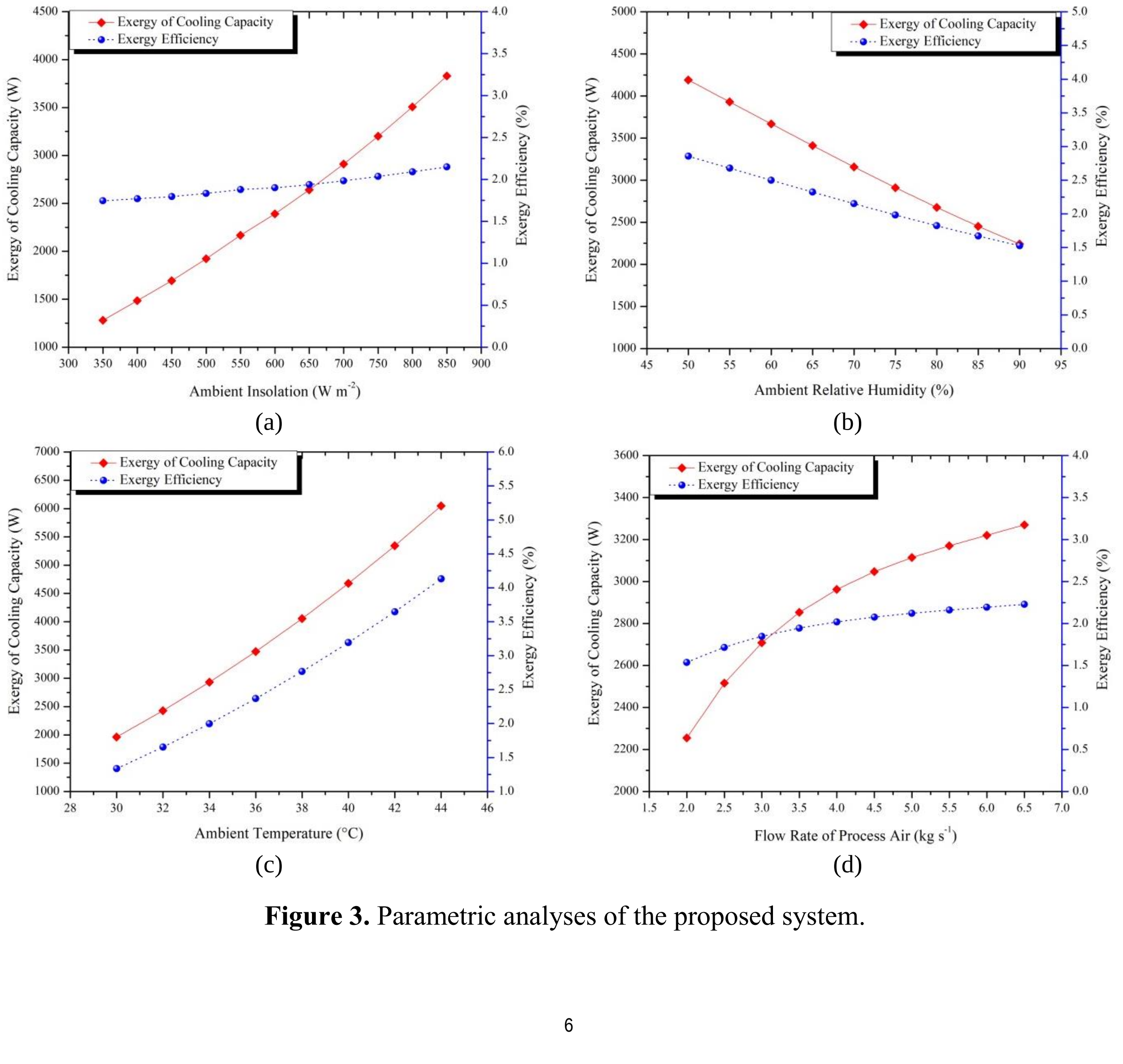

Figure 3. Parametric analyses of the proposed system.

Figure 3 presents the variation of the system's E_{CC} and η_{ex} with the variation of various design and operating parameters. Figure 3(a) presents the aforesaid parameters' variation with the variation of ambient insolation. A higher ambient insolation increases the evaporation rate of water vapor in the REG. Thus, the LDS supplied to the DEH is more concentrated. This increases the mass transfer of water vapor in the DEH, which correspondingly increases the temperature depression of the air in the DPIEC. Consequently, there is an increase in the system's E_{CC}. We observe that with the increase of ambient insolation, the η_{ex} increases slightly. This is due to the fact that the effect of increase in E_{CC} supersedes the increase in absorbed solar radiation exergy rate in the REG.

Figure 3(b) shows us that both the E_{CC} and η_{ex} decrease with the corresponding increase of ambient relative humidity. This occurs because at higher ambient humidity levels, the performance of the REG shrinks and thus, the LDS supplied to the DEH is more dilute. Thus, the air at the inlet to the DPIEC has higher humidity ratio. This decreases the temperature depression of the air in the DPIEC which in turn decreases the system's E_{CC}. As the absorbed solar radiation exergy rate is constant, the system's η_{ex} correspondingly decreases with the decrease in E_{CC}.

Figure 3(c) shows us that both the E_{CC} and η_{ex} increase with the increase in ambient temperature. An ambient air humidity ratio of 25 g/kg is assumed for this analysis. As a result of the increase in ambient temperature, the air temperature at the REG inlet increases. This correspondingly enhances the mass transfer of water vapor in the REG. Thus, the LDS supplied to the DEH is more concentrated which increases the mass transfer of water vapor in the DEH and consequently the temperature drop of the air in the DPIEC. Correspondingly, the system's E_{CC} and η_{ex} is augmented.

Figure 3(d) shows us that both the E_{CC} and η_{ex} increase with the increase in mass flow rate of process air. The mass transfer in the DEH increases with increase in the air flow rate which augments the system's E_{CC}. The system's E_{CC} and η_{ex} show a rapid increase when the process air mass flow rate is low, but increases at a slower rate at air flow rates more than 5.5 kg/s. This indicates that the optimum mass flow rate of air to LDS in the DEH is about 3.

4. Conclusions

The paper presents the second thermodynamic law analyses of a novel solar regenerated liquid desiccant assisted evaporative cooling system. The proposed system has the ability to provide suitable thermal comfort conditions inside large office buildings with high internals loads situated in the hot and humid climatic regions. For the base case, which is representative of a hot and humid climate, the system is able to maintain the room air temperature and humidity ratio at 24.3 °C and 9 g/kg respectively. The exergy of cooling capacity and exergy efficiency for the base case is about 2900 W and 2 % respectively. Parametric analyses show that the system performs the best under conditions of high ambient insolation and temperature, low ambient humidity and a process air to LDS mass flow rate of about 3 in the DEH.

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