Evaluation of Compressed Air Adsorption Dryer with Internal heating Regeneration, External Heating Regeneration, and Compressed Air Heating Regeneration Modes

Suresh Kannan. V *, Somasundaram. P**, Lenin. V.R *** and Vijayan. S****

Keywords：compressed air; moisture removal; adsorption dryer; heat regeneration, performance evaluation

ABSTRACT

In many engineering applications, compressed air without any moisture is critical. Heat regeneration-based adsorption dryers are commonly used to achieve ultra-low dew points in compressed air. This study experimentally examines the essential characteristics of three heat regeneration modes: internal heating, external heating, and compressor air heating. The calculations for the adsorption dryers are based on previous work, forming the foundation for a logical system comparison. The assessment is conducted over a common performance range, including a compressed air output of 1000 to 5000 m³/hr, operating pressures of 5 to 10 bar, and feed air temperatures of 25 to 45°C. The results show that the characteristic specific energy requirements for internal heating, external heating, and compressor air heating are 0.176, 0.342, and 0 kW/m³/min, respectively, for electrical energy. For dried regeneration air, the energy requirements are 0.203, 0, and 0 kW/m³/min, respectively.

Paper Received April, 2024. Revised March, 2025. Accepted March, 2025. Author for Correspondence: Suresh Kannan V. E-mail: professorskv@gmail.com

**Assistant Professor, Department of Mechanical Engineering, Kangeyam Institute of Technology, Tirupur 638108, Tamil Nadu, India*

***Professor, Department of Mechanical Engineering, Kangeyam Institute of Technology, Tirupur 638108, Tamil Nadu, India*

****Associate Professor, Department of Mechanical Engineering, Anil Neerukonda Institute of Technology & Sciences, Sangivalasa, Visakhapatnam 531162, Andhra Pradesh, India*

*****Associate Professor, Department of Mechanical Engineering, Coimbatore Institute of Engineering and Technology, Coimbatore 641109, Tamil Nadu, India*

The study concludes that, for a 3000 m³/hr flow rate at 35°C feed air temperature and 8 bar pressure, energy and capital costs are similar across all three modes, enabling practical system comparison for varying performance ranges.

INTRODUCTION

Compressed air is widely used in industries like agriculture, textiles, chemicals, and construction. However, moisture in compressed air can cause issues like rust, blockage, and increased energy use. It can also lead to operational problems such as freezing air lines and process malfunctions. To address this, moisture must be removed through a drying process, with the level of dryness measured by pressure dew point temperature. Selecting the right drying method is essential for ensuring smooth industrial operations and meeting process requirements (Venkatachalam et al.).

There are five primary methods for removing moisture from compressed air: refrigerant drying, over-compression, membrane drying, absorption drying, and adsorption drying. Among these, adsorption drying is often preferred for its ability to achieve very low pressure dew points, such as -70ºC (Saidur et al.). In the adsorption process, heat transfer plays a critical role in moisture removal from compressed air. The thermal conductivity of the adsorbent-adsorbate pair within the solid desiccant-packed tower greatly influences heat transfer during adsorption (Misha et al.). This conductivity depends on several factors, including temperature, pressure, adsorbate concentration, and the density of the desiccant's packing (Prakash et al.). Heat transfer occurs through conduction, with heat generated due to adsorption traveling in both axial and radial directions, affecting the desiccant bed's concentration profile and reducing adsorption capacity (Rezaei et al.). In real-world conditions, heat transfer within desiccant beds involves solid particle conduction, convection in void spaces, and radiation from desiccant surfaces. Given the non-homogeneous nature of desiccant beds, understanding heat transfer characteristics can be complex, as different modes of heat transfer interact. To address this, effective thermal conductivity is used in analysis. Radiation heat transfer from individual desiccant particles is absorbed by surrounding particles and feed air. However, modeling the packed bed's effective thermal conductivity remains challenging due to its non-isotropic properties (weidenfeld et al.)

In adsorption studies, temperature variation data is crucial and is typically derived from energy balance equations of the entire system. The energy balance includes the heat gain by desiccant particles, moisture in the desiccant bed, and the heat transferred to the flowing air (Polesek-Karczewska et al.). For instance, Kwapinski et al. (2010) developed a correlation considering both radial and axial thermal conductivities in the adsorption bed. Visser et al. (2008) modeled heat and airflow through the saturated bed with varying porosity and desiccant heterogeneity under steady-state and non-steady-state conditions. Fuentes et al. (1998) used a pseudo-model to describe homogeneous beds, showing that thermal conductivity varies with radial position. Béttega et al. (2011) found that thermal conductivity profiles significantly influence the adsorption system's behavior. Another critical aspect of adsorption processes is predicting the cyclic steady-state (CSS) (Nilchan et al.). Several techniques have been developed to determine CSS, often requiring repeated simulations, which are computationally demanding and sometimes yield suboptimal accuracy (Ko et al.).

A review of the literature on adsorption dryers for compressed air drying reveals that heat regeneration is commonly used for high dryness requirements. However, the complex heat and mass transfer mechanisms in moisture removal from desiccants have made theoretical analysis challenging. Energy consumption significantly impacts operating costs, but no prior studies compare energy consumption and costs across different regeneration modes. This study aims to analyse the performance of three heat-based regeneration methods—internal heating, external heating, and compressed air heating—offering a comparative analysis to guide optimal regeneration selection based on performance and cost.

MATERIALS AND METHODS

The adsorption dryer can be regenerated using three modes: internal heating, external heating, and compressor heating. In internal heat mode, regeneration uses a flow of dried compressed air and electrical heating. External heat mode uses ambient air heated by external electricity. Compressor heat mode utilizes heat generated during compression.

Layout design of three modes of heat regeneration-based adsorption dryer

The saturated desiccant in an adsorption dryer can be regenerated through three modes: internal heat, external heat, and compressor heat. Fig. 1(a) shows the schematic of the internal heat mode adsorption dryer. This setup includes flat sieves (2) and wire mesh (3) at the entry and exit to prevent desiccant carryover at high feed stream velocities. Both adsorber vessels are interconnected. A bypass valve with orifice (5) switches the feed stream from adsorption to regeneration. During regeneration, the offline vessel operates at atmospheric pressure, while adsorption occurs in the online vessel under operating pressure. The regeneration process involves heating the desiccant bed to expel moisture, followed by airflow through the bed to the atmosphere via the exhaust valve (6). After regeneration, the vessel pressure is equalized before switching back to adsorption mode. This cycle ensures efficient desiccant regeneration.

Fig. 1(b) illustrates the schematic diagram of the regeneration process using external heat mode in an adsorption dryer. In this system, an external heater and blower are used to regenerate the desiccant, requiring only a small amount of dried air for purging. The regeneration air is drawn from the atmosphere and heated by an external heater (9), with the blower (8) assisting in the airflow. Two 4/2-way valves (1) control the air flow, and the exhaust valve (6) manages the discharge during the pressure build-up phase after regeneration. During the regeneration cycle, the blower helps circulate the hot air through the desiccant bed. Once regeneration is completed, the humidified air is expelled via the exhaust valve, and the cooling stage begins. The thermostat (TS) monitors the temperature, signalling the transition to the cooling phase, where cold air is introduced to cool the desiccant bed. After the cooling stage ends, the valve (6) closes, and pressure build-up occurs, preparing the system for vessel switching. The regenerated vessel is depressurized via the exhaust valve (5) and muffler (4).

Fig. 1(c) depicts the schematic of a compressor heat mode regeneration system, utilizing a closed-loop design for continuous operation. The system is equipped with two adsorber vessels and multiple valves for switching between adsorption and regeneration phases. The regeneration air, typically more humid, requires higher temperatures to achieve the desired pressure dew point due to the closed-loop setup.

Experimental design for system evaluation

The system evaluation is carried out by describing the essential characteristic of different types of adsorption dryer. Using fundamental parameters, objective evaluation is arrived. The principal calculation schemes of adsorption dryers were established in the previous work carried out by the authors [13-17]. Such calculations form the basis of a logical comparison between systems. The system assessment is carried out with the significant process variables such as the operating pressure from 5 to 10 bar, feed air temperature from 25°C to 45°C, and the pressure dew point of product air as -40°C. The characteristic compressed air cost parameter to be established for adsorption dryers is based on INR 1000 per 1000 m^3 [18-23].

Regeneration energy establishment

In the regeneration experiments, an electric heater provided the heat energy required to raise the bed to the regeneration temperature (TRE). During desorption, the adsorber temperature moves from the entry to the exit, and the change in temperature at the bed outlet signals the completion of temperature migration, triggering the switch-off of the heating phase. This temperature is known as the switching-off temperature (TRO).

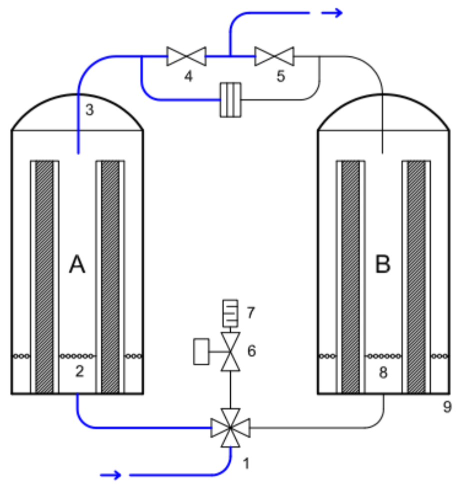
In the study, inlet air temperatures were systematically controlled and varied using a temperature-regulated air supply system equipped with a precision heater and a feedback temperature controller. This setup allowed us to maintain consistent inlet air temperatures within a range of ±0.5°C of the target value. The temperatures were varied across specific increments to simulate different operational conditions relevant to the adsorption and desorption processes.

The variation in inlet air temperature directly impacted the regeneration energy demand due to its influence on the thermal properties of the system. Higher inlet air temperatures reduced the temperature gradient required for desorption, lowering the energy needed for heating the adsorbent bed to the regeneration temperature. Conversely, lower inlet air temperatures increased this gradient, resulting in higher energy consumption.

Inlet air temperature directly affected regeneration energy demand by influencing the thermal properties of the system. Higher inlet temperatures reduced the temperature gradient for desorption, lowering energy consumption, while lower temperatures increased the gradient, requiring more energy. The study showed that optimizing inlet air temperature is crucial for minimizing energy demand while ensuring effective TSA system performance.

At a maximum water vapour loading in the desiccant bed, the ratio between Q_{H2O} and Q_S to be greater than 0.7 has to be taken account while calculating the amount of regeneration air at suitable temperature. Due to low desiccant load (Q_{H2O} / Q_S > 0.7), a substantial part of heat duty is used to heat the desiccant bed and machinery.

The regeneration air requirement is calculated through LMTD-Desorption and LMTD-Cooling. The LMTD-Desorption is calculated using regeneration temperature (T_{RE}), switching off temperature (T_{RO}), and desiccant bed temperature (T_{BC}) using equation (1).



(a)Regeneration by internal heat

$$LMTD_{Desorption} = \frac{T_{RO}-T_{BC}}{ln\frac{T_{RO}-T_{BC}}{T_{RE}-T_{RO}}} \qquad (1)$$

The $LMTD_{Cooling}$ is calculated using regeneration temperature (T_{RE}), the entry temperature (T_O), and the outlet temperature of cooling air (T_{CO}) using equation (2). The exit temperature T_{CO} of the cooling air is not to exceed 70 to 80°C so that the product dry air moisture content is low.

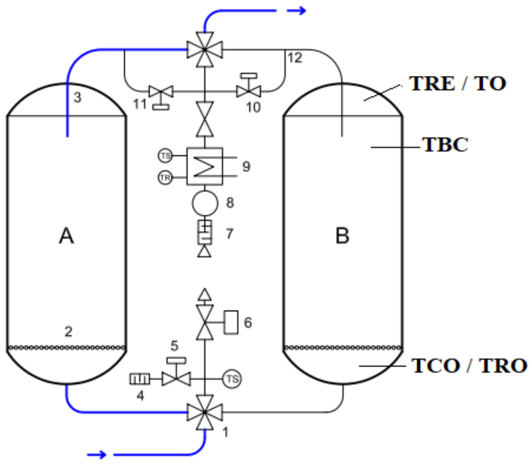
$$LMTD_{Cooling} = \frac{T_{RE}-T_{CO}}{ln\frac{T_{RE}-T_{O}}{T_{CO}-T_{O}}} \qquad (2)$$

During the cooling phase, the desiccant bed and adsorber vessel are brought to low temperature.

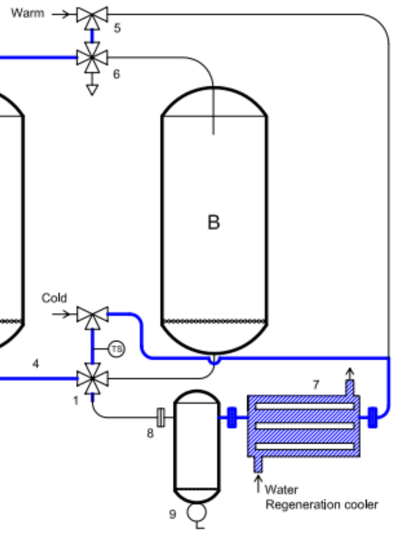
The energy requirement to heat desiccant material Q_{dr} in Joule is calculated with the specific heat value using the equation (3).

$$Qdr=mdr\times Cdr\times (T_{RE} - T_O) \qquad (3)$$

The load factor value essentially quantifies the desiccant's moisture removal performance for the specified conditions of Dwell time (3 seconds), Relative humidity (60%), and Feed air temperature (35°C). It bridges the gap between theoretical adsorption behavior and practical system performance by incorporating the real-world inefficiencies and interactions inherent in the adsorption process. The value of Kl is multiplied by the moisture load (hc) to calculate the total energy required for desorption, thereby helping design and optimize the adsorption system's energy consumption and operational efficiency.



(b)Regeneration by external heat



(c)Regeneration by compressor heat

Fig.1 Schematic diagram of adsorption system heat regeneration modes

Equation (4) considers the heat of adsorption Q_H in J/kg, which is dependent on a load factor value of 0.13 (Kl) corresponding to a dwell time of 3 seconds and a feed air relative humidity of 60%. This value is then multiplied by the moisture load hc, which is 0.0440 kg at a feed air temperature of 35°C to calculate heat energy required to remove moisture load in desiccant bed in Joule.

$$Q_{H2O}=h_{c}\times QH \tag{4}$$

The amount of heat required for adsorber vessel heating to the required regeneration temperature (Q_{vessel} in Joule) can be calculated using equation (5).

$$Q_{vessel}=m_{vessel}\times C_{vessel}\times \Delta T \tag{5}$$

The total energy is arrived by adding the heat duty for desiccant Q_{dr}, moisture loading Q_{H2O} and adsorber vessel Q_s in Joule using equation (6).

$$Q_s=Q_{dr}+Q_{H2O}+Q_{vessel} \tag{6}$$

The energy required due to radiation Q_{ra} in Joules of an adsorber tower depends on its position. The higher the LMTD-Desorption, the larger the loss of heat through radiation. For rough estimates, Q_{rad} is taken as 8% of the total calculated energy Q_S using equation (7).

$$Q_{ra} = Q_s \times \frac{8}{100} \tag{7}$$

The obtained Q_{ra} value in Joule is based on theoretical calculation using equation (8). General tolerance of 5% is to be accounted to meet out safety factor to calculate total heat energy required for regeneration in Joule.

$$Q_{tot}=Q_s+Q_{ra}+(5\%)\,Q_{ra} \tag{8}$$

The required energy for cooling in Joule is comparatively lower than for regeneration. The desiccant bed and adsorber vessel need to be cooled.

$$Q_C=[(m_{vessel}\times C_{vessel})+(m_{dr}\times C_{dr})]\,\Delta T_{cooling} \tag{9}$$

To calculate volume flow rate of regeneration air $V_{regen\ air}$ in m^3 taking heat allowance into account, the heat content q_{dr} (in J/m^3) of the desiccant has to be accounted. First, the equation (10) calculates the desorption air quantity for the very first part of the moisture load, and then, excluding water vapour load. The effective values for heat quantity q_{dr}=130000 J/m^3 and the heat capacity c_{dr}=1297 $J/m^3{}^{\circ}C$.

$$V_{regen\ air} = \frac{Q_{H2O}}{q_{dr}} + \frac{Q_{ges} - Q_{H2O}}{c_{dr} \times \Delta T_R} \tag{10}$$

To determine volume flow rate of cooling air $V_{cooling\ air}$ in m^3 the energy requirement $Q_{cooling\ air}$ is taken into account and is calculated using equation (11).

$$V_{cooling\ air} = \frac{Q_c}{c_{dr} \times \Delta T_c} \tag{11}$$

Considering the regeneration air quantity V_{rh} in m^3 regeneration time t_r in second and cooling air quantity

V_{rc} in m^3 ,the total air volume V_t in m^3/sec is calculated using equation (12).

$$V_t = \frac{V_{rc}}{t_r} \tag{12}$$

Regeneration time t_h in seconds is calculated based on total air volume (V_r in m^3/sec), and the amount of regeneration air V_{rh} in m^3 is calculated using equation (13).

$$t_h = \frac{V_{rh}}{V_r} \tag{13}$$

Cooling time t_c in seconds is calculated based on total air volume (V_c in m^3/sec) and volume flow rate of cooling air V_{rc} in m^3 using equation (14).

$$t_c = \frac{V_{rc}}{V_c} \tag{14}$$

The total energy requirement Q_{tot} in Joule is calculated and can be used to calculate heating power P_e, considering heating system efficiency using equation (15).

$$P_e = \frac{Q_{tot}}{Heater\ efficiency} \tag{15}$$

The heater energy in Joule/sec is calculated using heating power (Pe) and heating time t_h in seconds using equation (16).

$$P_h = \frac{P_e}{t_h} \tag{16}$$

The mean power requirement during heating phase for heater P_{hm} in Joule/second and the blower power in Joule/second is calculated using the equation (17) and equation (18) respectively. In equation (18), V_b represents air volume flow delivered by the blower in m^3/sec, D_p represents total pressure increase in blower in N/m^2 and blower efficiency is taken as 0.30. The value of 0.30 (30%) could be an intentionally conservative assumption to ensure that the energy requirements are overestimated, providing a safety margin in the system's design. By assuming a lower efficiency, the calculations for energy requirements (such as power consumption for the blower) are likely to be higher, ensuring that the system is not under-designed or under-resourced. This may be especially useful when modeling or analyzing initial system performance before tuning it based on more precise, real-world conditions. The heating phase of the temperature swing adsorption system might require the blower to work under less optimal conditions (e.g., moving air at a high temperature or handling fluctuating air pressures), which could result in significant energy losses.

$$P_{hm} = P_e \times \frac{t_h}{t_c} \tag{17}$$

$$P_b = \frac{V_b \times D_p}{Blower\ efficiency} \tag{18}$$

The mean power requirement for blower P_{bm} in Joule/Second and mean total power requirement P_m considering cooling period during regeneration (t_B represents cycle time in seconds) can be calculated using the equation (19) and equation (20).

$$P_{bm} = P_e\left(\frac{t_h + t_c}{t_B}\right) \tag{19}$$

$$P_m = P_{hm} + P_{bm} \tag{20}$$

Economic Analysis of heat regeneration-based adsorption dryer

From the known calculation systems of the heat regeneration mode adsorption dryer was known, and it forms the basis for evaluation. The developed regression equation forms the basis to conduct an objective study of the dryer.

The calculation primarily focused on the energy consumed during the regeneration cycle, measured in Joules or kilowatt-hours (kWh), and converted into monetary cost using local energy pricing. However, to provide a comprehensive assessment, operational costs such as blower or compressor power and heat recovery systems were incorporated into the total energy analysis.

While maintenance costs (e.g., replacement of adsorbents, heater elements, or blower components) were not directly included in the energy cost calculations, they are considered in lifecycle cost assessments.

A significant error is to be included in a comparison of other modes of dryers later unless the correctly specified actual and different cost of energy. Only using the right parameters of compressed air can achieve an objective evaluation of different adsorption dryers. This depends on assigning uniform energy costs to the energy requirements for all types of regeneration.

To calculate the total pressure drop in bar, the pressure drop occurs in adsorption, regeneration, and cooling steps are considered.

$$\Delta p_{Total} = \Sigma \Delta p_n \tag{21}$$

If added, the sum of individual component pressure losses offers pressure loss of adsorption dryer setup, so that the product air operating pressure in bar is calculated by

$$p_o = p_i - \Delta p_{Total} \tag{22}$$

This theoretical pressure drop is compared with a loss in compressor performance. The total performance of the compressor is assigned the requirement of power through pressure drop. Here p_i represents feed air pressure in bar. This also allows determining the pressure loss as an objective value element of similar electrical performance.

$$\Delta p_i = \left(1 - \frac{{p_o}^2}{{p_i}^2}\right) \times p_c \tag{23}$$

Performance of compressor has arrived through the energy required value based on compressor type is multiplied by the regeneration volume flow rate, purge air quantity, and cooling air quantity to arrive performance product by equation (24).

$$p_c = V \times E_{fl} \tag{24}$$

The average performance requirement value is arrived by taking adsorption time and regeneration time and is calculated using the equation (25), (26), and (27).

For the purging air by,

$$p_{rs} = V_{fl} \times E_{fl} \times \frac{t_{sp}}{t_B} \tag{25}$$

The average performance requirement value is calculated by,

$$p\Delta p_l = \Delta p_l \times p_v \tag{26}$$

For each adsorption dryer, the total of average performance is calculated using the equation (27).

$$p_{vtotal} = \Sigma p_{vn} \tag{27}$$

The cost of energy per unit time is calculated using the input energy cost data and average performance data using the equation (28).

$$K_{Eh} = K_E \times \ P_{vges} \tag{28}$$

To form a common uniform parameter, the cost of energy per 1000 m^3 is calculated using equation (29) in INR.

$$C_{o,1000} = C_{oh} \times \frac{1000}{V_e} \tag{29}$$

For a dryer comparison taking investment into account, capital costs are to be included and capital amortization (C) is calculated using the equation (30) in INR. Here A represents equipment cost, n represents writing off period, z represents interest amount.

$$C = \frac{z \times (1 + z)^n}{(1 + z)^{n-1}} \times A \tag{30}$$

The operational cost of running time and differential flows at the exit of dryers is calculated by converting the capital service dispersed over the yearly service life and expressed in capital amortization (C_h).

$$C_h = C \times \frac{t_{Bh}}{t_{Ba}} \tag{31}$$

System evaluation data

Fig. 2 illustrates the cycle time sequence for internal heat, external heat, and compressor heat regeneration mode adsorption dryers. The chart includes the time required for each phase, such as adsorption, regeneration, heating, cooling, flushing, holding, and pressure build-up, all measured in seconds. The adsorption period is 3600 seconds for all types. External heat regeneration has the longest regeneration period (2952 seconds) and the shortest cooling time (396 seconds). Compressor heat regeneration has the shortest regeneration period (1440 seconds) and does not require pressure build-up time. Internal heat regeneration has the longest heating (2340 seconds) and cooling time (1116 seconds), but does not require flushing time.

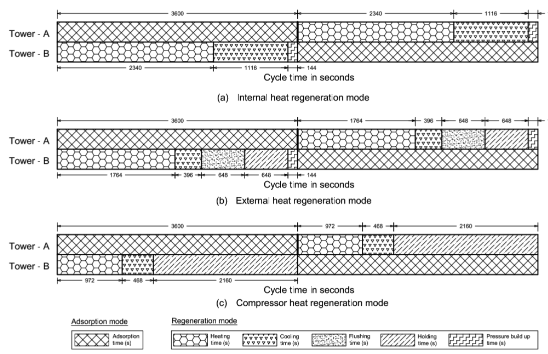


Fig.2 Cycle time sequence for heat regeneration modes

Fig. 3 illustrates the process of preparing compressed air through heat regeneration. When air is compressed to 8 bar, impurities, including moisture, become concentrated in the compressed air. As the air cools to ambient temperature, the dew point rises, causing excess moisture to condense. For example, at 20°C and 60% humidity, a compressor with 100 m^3/hr intake carries 1040 g/hr of moisture. After compression, 378 g/hr of water vapor is added, and 824 g/hr of moisture is removed by the aftercooler.

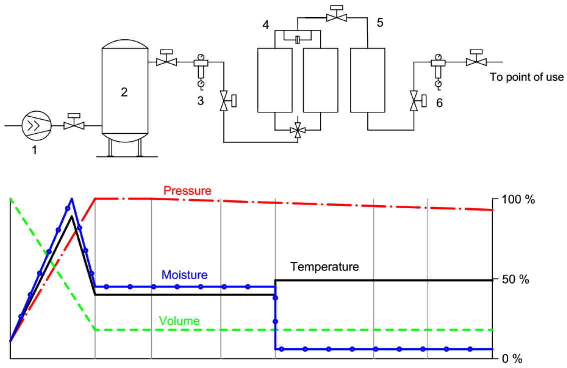


Fig.3 Property change of compressed air in heat regeneration mode adsorption dryer

The data presents a comparison of key parameters for three types of heat regeneration-based adsorption dryers: internal heat regeneration, external heat regeneration, and compressor heat regeneration. In terms of regeneration air requirements, internal heat regeneration does not require purge air during heating and cooling, while external heat regeneration requires 5% purge air during heating time and 4% during cooling time. The regeneration air power requirement for internal heat regeneration is 0.697 kW/m³/min. Regarding dryer performance, the heating power requirement for external heat regeneration is the highest at 0.61 kW/m³/min, while the blower power requirement is 0.13 kW/m³/min for external heat regeneration. For product air, compressor heat regeneration achieves the highest outlet air at 100%, with the lowest air loss of 1.5%, whereas internal heat regeneration results in 95% outlet air with a 5% loss. The operating pressure is approximately 7.74 bar for internal and external heat regeneration, and 7.43 bar for compressor heat regeneration. The characteristic specific energy consumption for electrical energy is lowest for internal heat regeneration at 0.176 kW/m³/min, while external heat regeneration has a higher value at 0.342 kW/m³/min. For flush air, external heat regeneration consumes 0.053 kW/m³/min.

The data highlights the differences in performance among the three types of heat regeneration-based adsorption dryers. It highlights that the choice depends on factors such as energy availability, process requirements, and the trade-off between capital and operational costs. Applications requiring frequent regeneration often benefit from internal or external heating, while simpler setups may favor compressed air heating. Environmental goals, such as reducing carbon footprints, can influence the selection when renewable or waste energy sources are available. Standards like ISO 8573-1 (compressed air quality) and ASHRAE guidelines provide frameworks for system design and efficiency.

Modeling tools like thermodynamic analysis, lifecycle cost analysis, and simulation software namely ASPEN Plus and MATLAB help compare options under specific conditions. The decision-making process ensures an optimal balance between energy efficiency, cost, and operational feasibility.

RESULTS AND DISCUSSION

The regeneration characteristics in all three modes are influenced by factors like desorption temperature, switch-off temperature, and the logarithmic temperature differences during desorption and cooling. These optimal values help determine the energy cost for regeneration. The system assessment uses a common performance range for all modes, which includes a compressed air output of 1000 to 5000 m³/hr, operating pressures from 5 to 10 bar, and feed air temperatures between 25 and 45°C.

Effect of desorption and switching off temperature

Figure 4 shows the variation of pressure dew point temperature in relation to desorption and switching off temperatures in a TSA system. Desorption temperature and pressure dew point are indirectly related. For a -40°C pressure dew point, the desorption and switching off temperatures are 180°C and 130°C, respectively. During desorption, the temperature inside the adsorber moves towards the outlet, with the speed of this movement remaining consistent at different regeneration temperatures. This is due to the heat capacity ratio between the gas and solid phases. A change in temperature at the outlet signals the end of the heating phase, affecting the pressure dew point.

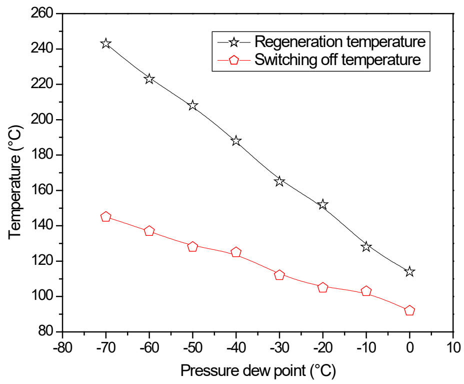


Fig.4 Variation of PDPT corresponding to desorption temperature

Effect of LMTD-Desorption and LMTD-Cooling

Figure 5 shows the logarithmic temperature variations during desorption and cooling phases in relation to the pressure dew point temperature. LMTD-Desorption is calculated using equations for switching off temperature, desiccant bed, and desorption temperature, while LMTD-Cooling is calculated from desorption, inlet, and cooling air outlet temperatures. The product air dew point temperature varies from -10°C to -70°C with increasing desorption temperatures. The cooling air outlet temperature stays below 70-80°C to maintain low humidity on the desiccant bed's dry side, completing the cooling phase after a specified time [14].

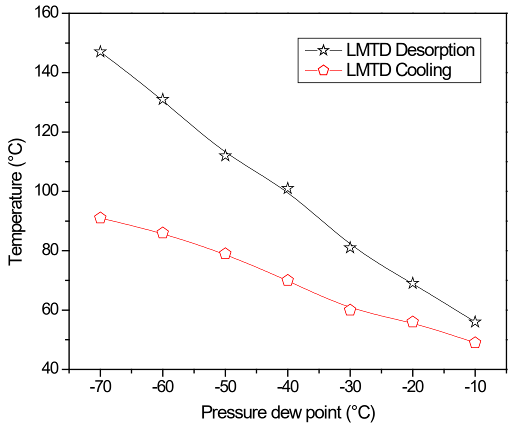


Fig.5 Variation pressure dew point with respect to logarithmic temperature difference

Energy cost diagrams

Figure 6 illustrates the energy costs per 1000 cubic meters for different regeneration modes at a constant volume flow rate and feed temperature. As operating pressure increases, heat mode regeneration dryers become more cost-effective than heatless regeneration dryers. Heat mode dryers excel in controlling secondary relative humidity, which is pressure-dependent, optimizing the correlation between flow rate and working state. The graph shows that internal heating mode becomes equally energy-efficient to heatless mode at 6.5 bar, suggesting they can be interchangeable at this pressure based on application and resources. Beyond this point, internal heating mode becomes more efficient than heatless mode but remains less efficient than external and compressor heat modes. Overall, the graph highlights the advantages of heat mode regeneration at higher operating pressures.

Figure 7 illustrates the energy costs (in INR) for four regeneration modes (heatless, internal heat, external heat, and compressor heat) at various feed air temperatures (25°C to 45°C). The graph reveals that as feed air temperature increases, energy costs rise for all modes. Heatless mode incurs higher costs, while compressor heat mode remains the most energy-efficient. At 33°C, internal and external heat modes have the same energy costs, while at 42.5°C, internal heat and heatless modes are more expensive than external heat and compressor heat modes. This data aids in selecting the most efficient regeneration mode based on operating temperatures.

Fig. 8 shows the energy cost at constant pressure and feed temperature for different volume flow rates (1000 m³). The energy cost for heatless regeneration adsorption dryers remains constant, with changes in volume flow rate affecting only the energy cost. In contrast, heat mode regeneration dryers exhibit lower energy costs per 1000 m³ as the flow rate increases, due to the dryer's design, regeneration methods, and purge air requirements.

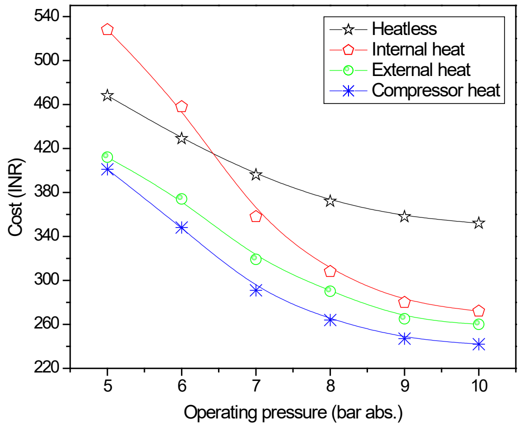


Fig.6 Energy cost referred to Q=3000 m^3/hr and T_i=35°C

Fig. 9 compares the capital and energy costs per 1000 m³ at constant feed pressure and temperature for varying volume flow rates. Heatless regeneration mode dryers are cheaper to produce, but capital and energy costs move in opposite directions. The data indicates that heat mode regeneration dryers are more cost-effective in terms of both capital and energy costs compared to heatless regeneration dryers.

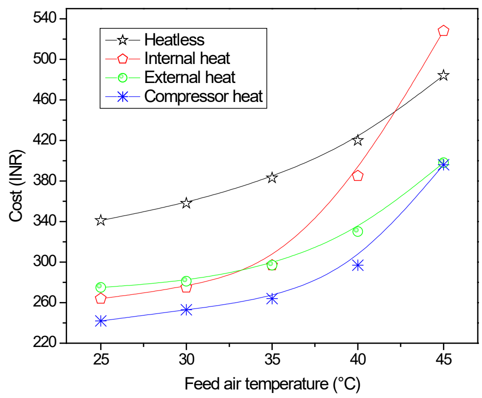


Fig.7 Energy cost referred to Q=3000 m^3/hr and p=8 bar abs

Application suitability

This section shows the summary of suitability of different adsorption systems based on key parameters such as product air flow rate, PDPT, operating pressure of air, and operating temperature of air. External heating may result in higher PDPs due to heat losses, while internal heating often achieves lower PDPs by delivering heat more efficiently to the desiccant material. Similarly, compressed air heating, though simpler, might be less effective in removing deeply adsorbed moisture, leading to relatively higher PDPs.

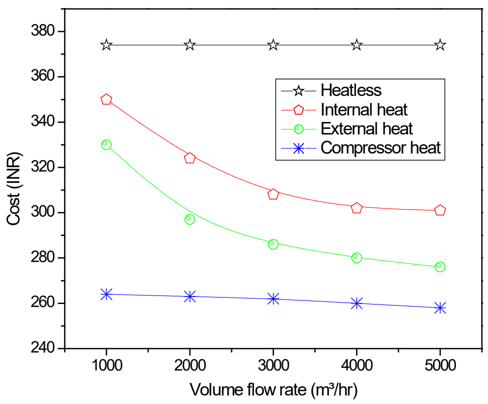


Fig.8 Energy cost referred to p=8 bar and T_i=35°C

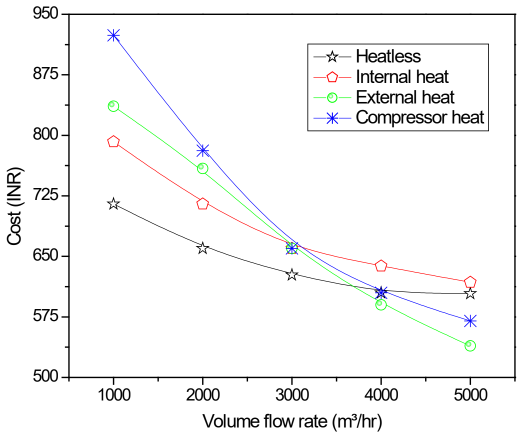


Fig.9 Energy and capital cost referred to T_i=35°C and p=8 bar abs.

These differences were quantified in our study by measuring the PDP at various operating conditions for each regeneration mode. Additionally, drying efficiency was assessed by comparing the moisture removal rate and the specific energy consumption for each mode. Product quality was further evaluated by monitoring parameters such as the air's residual moisture content and its compliance with specific application requirements.

Our findings demonstrate that the choice of regeneration mode directly influences both energy efficiency and the quality of the product air, highlighting the importance of selecting the appropriate mode for specific operational goals.

The following summarizes the application specifications for each regeneration mode:

1. For air flow, internal heat regeneration is suitable for flows between 200 to 5000 m³/hr, while external heat and compressor heat are also applicable for this range. For air flow above 5000 m³/hr, external heat and compressor heat are suitable, but internal heat is not. For air flow up to 200 m³/hr, none of the modes are applicable.
2. Regarding process air temperature, all regeneration modes are suitable for temperatures between 5 to 45°C. However, none of the modes work when the process air temperature exceeds 45°C.
3. For operating pressure, all three regeneration modes are applicable for pressures ranging from 3 to 10 bar.
4. For pressure dew point temperature (PDPT), internal heat and external heat can achieve up to -40°C, but compressor heat is not suitable for this PDPT. However, all modes are capable of achieving a PDPT of up to -70°C.

CONCLUSIONS

The performance of three heat regeneration-based adsorption dryers was evaluated, leading to the following conclusions:

- Heat mode adsorption dryers are suitable for feed air inlet temperatures between 25 to 45°C, operating pressures from 5 to 10 bar, and product air dew points ranging from -40 to -70°C.
- Heat mode regeneration is ideal for strongly adsorbed species, as it allows for high desorbate recovery relative to the maximum adsorption capacity of the desiccant. However, it is not suitable for rapid cycling, which reduces adsorbent efficiency. In contrast, the heatless mode provides less energy for regeneration, resulting in a shorter cycle time but also less moisture adsorption.
- The specific energy requirements for regeneration are 0.176, 0.342, and 0 kW/m³/min for internal heating, external heating, and compressor air heating, respectively. For heatless regeneration, the energy requirement is 0.697 kW/m³/min, with regeneration achieved by purging 15% of product air [19].
- This system comparison, using reproducible dryer data across varying performance ranges, can guide practical applications.

Nomenclature

Qra	-	Additional energy requirement due to radiation, Joule
Vb	-	Air volume flow delivered by the blower,m3/s
n	-	Blower efficiency
Pb	-	Blower power, J/s
Ch	-	Capital amortization cost, INR
C	-	Capital cost, INR
Vcooling air	-	Cooling air quantity,m3
Vrc	-	Cooling air quantity,m3
Tc	-	Cooling time, sec
C0,1000	-	Cost of energy per 1000m3 in INR
m3	-	Cubic Metre
CSS	-	Cyclic steady state
°C	-	Degree Celsius

TBC - Desiccant bed temperature
t_h - Desorption time, sec
Qc - Energy requirement for cooling, Joule
TO - Entry cooling air temperature, °C
A - Equipment cost, INR
p_i - Feed air pressure, bar
Qtot - Grand total energy requirement, Joule
Qvessel- Heat duty for adsorber vessel, Joule
Qdr- Heat duty for desiccant material, Joule
QH2O- Heat duty for moisture load on desiccant bed, Joule
QH - Heat of adsorption, J/kg
ddr - Heat quantity, J/m^3
Ph - Heater energy, J/sec
Pe - Heating power, Joule
hr - hour
INR - Indian Rupees
z - Interest amount, INR
Cdr - Heat capacity, J/m3°C
kW - kilowatt
LMTD- Logarithmic Mean Temperature Difference, °C
mvessel -Mass of adsorber vessel, kg
mdr - Mass of desiccant, kg
Phm - Mean power requirement for heating device, J/s
Pbm - Mean power requirement for heating device, J/s
min - minute
hc - Moisture load in kg
TCO - Outlet temperature of cooling air, °C
po - Product air pressure, bar
Vregen air -Regeneration air quantity,m3
Vrh - Regeneration air quantity,m3
tr - Regeneration period, sec
TRE - Regeneration temperature
Cdr - Specific heat of desiccant, J/kg°C
Cvessel- Specific heat of vessel material, J/kg°C
TRO - Switching-off temperature
Vc - Total air volume,m3/sec
tB - Total cycle time, second
Qtot - Total energy requirement, Joule
QS - Total heat duty for desiccant, moisture load and vessel load, Joule
Dp - Total pressure rise in blower, N/m2

REFERENCES

Chu, Yu-Cheng, et al., "On effective design and operating conditions of desiccant dehumidification system," *Heat Transfer Engineering, vol.* 39.no.7-8, pp.598-607, 2018.

Kamar, H. Mohamed, et al., "Performance Analysis of a Solid Desiccant Air Dehumidifier," Journal of Advanced Research in Fluid Mechanics and Thermal Sciences, vol.10, no.1, pp.5-10, 2015.Venkatachalam SK, Thottipalayam Vellingri A, Selvaraj V., “Low-temperature drying characteristics of mint leaves in a continuous-dehumidified air-drying system.” Journal of Food Process Engineering, vol.43, no.4, 2020. DOI: 10.1111/jfpe.13384.

Kannan, V. Suresh, T. V. Arjunan, and S. Vijayan., "Drying characteristics of mint leaves (Mentha arvensis) dried in a solid desiccant dehumidifier system," Journal of Food Science and Technology, vol. 58, no.2,pp.777-786,2021.DOI: https://doi.org/10.1007/s13197-020-04595-z.

Kannan, V. Suresh, T. V. Arjunan, and S. Vijayan., "Experimental investigation of temperature swing adsorption system for air dehumidification," Heat and Mass Transfer, vol. 56, no.7,pp. 2093-2105,2020.DOI: https://doi.org/10.1007/s00231-020-02841-w.

Kannan, V., and T. Arjunan., "Optimization of process parameters of pressure-swing adsorption cycle in a silica gel desiccant dehumidification system," J Adv Chem, vol. 13, no.10,pp.6543-6551,2017.

Ko, Daeho, Ranjani Siriwardane, and Lorenz T. Biegler., "Optimization of a pressure-swing adsorption process using zeolite 13X for CO2 sequestration," Industrial & engineering chemistry research, vol. 42,no.2,pp.339-348,2003.DOI: https: //doi.org/ 10.1021/ie0204540.

Kwapinski, Witold, et al., "Thermal and flow effects during adsorption in conventional, diluted and annular packed beds," Chemical engineering science, vol. 65, no.14,pp.4250-4260, 2010.DOI: https://doi.org/10.1016/j.ces.2010.04.017.

Mandegari, Mohsen Ali, and Hassan Pahlavanzadeh., "A study on the optimization of an air dehumidification desiccant system," Journal of Thermal Science and Engineering Applications, vol.5, no.4, pp.041002, 2013.

Misha, S., et al., "Review of solid/liquid desiccant in the drying applications and its regeneration methods," Renewable and Sustainable Energy Reviews,vol.16, no.7,pp.4686-4707, 2012.DOI:

https://doi.org/10.1016/j.rser.2012.04.041.

MM, Matheswaran. "Performance Comparison of Heat Regeneration Modes in Adsorption Dryers for Ultra-Low Dew Point Compressed Air Production." *中國機械工程學刊* 44.5 (2023): 473-480.

Nilchan, Sujinda, and C. C. Pantelides., "On the optimization of periodic adsorption processes," Adsorption,vol. 4,no.2,pp.113-147,1998. DOI: https://doi.org/10.1023/A: 1008823102106.

Polesek-Karczewska, S, "Effective thermal conductivity of packed beds of spheres in transient heat transfer," Heat and mass transfer, vol.39, no.5,pp.375-380,2003.DOI: https://doi.org/10.1007/s00231-002-0343-6.

Prakash, M. Jose, Madhu Prasad, and K. Srinivasan, "Modeling of thermal conductivity of charcoal–nitrogen adsorption beds," Carbon,vol.38,no.6,pp.907-913,2000.DOI: https://doi.org/10.1016/S0008-6223(99)00202-X.

Rezaei, Fateme, and Mattias Grahn., "Thermal management of structured adsorbents in CO2 capture processes," Industrial & engineering chemistry research, vol. 51, no.10, pp.4025-4034,2012.DOI: https://doi.org/10.1021/ie201057p.

Saidur, Rahman, Nasrudin Abdul Rahim, and Md Hasanuzzaman., "A review on compressed-air energy use and energy savings," Renewable and sustainable energy reviews,vol. 14,no.4,pp.1135-1153, 2010. DOI:https://doi.org/10.1016/j.rser.2009.11.013.

Sawardsuk, Prueksa, Jiraporn Sripinyowanich Jongyingcharoen, and Ekkapong Cheevitsopon., "Effect of different desiccant bed designs in a desiccant column on their dehumidification performance," Agricultural Engineering International: CIGR Journal, vol. 24, no.1,2022.

Sureshkannan, V., et al., "Parametric study on pressure-based packed bed adsorption system for air dehumidification," Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering, vol.235, no.5,pp.1351-1362,2021.DOI: https://doi.org/10.1177%2F09544089211007318.

Sureshkannan, V., S. Vijayan, and V. R. Lenin., "Design and performance analysis of compressed air adsorption dryer with heatless regeneration mode," Heat and Mass Transfer, vol. 58, no.4, pp.631-641,2022.DOI:https://doi.org/10.1007/s00231-021-03136-4.

Venkatachalam SK, Thottipalayam Vellingri A, Selvaraj V., "Low-temperature drying characteristics of mint leaves in a continuous-dehumidified air-drying system. "Journal of Food Process Engineering, vol.43, no.4, 2020. DOI: 10.1111/jfpe.13384.

Venkatachalam, Sureshkannan, and Arjunan Thottipalayam Vellingri. "Optimization of process parameters of adsorption process in solid desiccant dehumidification system using genetic algorithm." *中國機械工程學刊* 40.5 (2019): 547-553.

Visser, C. J., A. G. Malan, and J. P. Meyer., "An artificial compressibility algorithm for modelling natural convection in saturated packed pebble beds: A heterogeneous approach," International journal for numerical methods in engineering, vol.75, no.10,pp.1214-1237, 2008.DOI: https://doi.org/10.1002/nme.2296.

Weidenfeld, Galit, Yeshayahu Weiss, and Haim Kalman, "A theoretical model for effective thermal conductivity (ETC) of particulate beds under compression," Granular Matter, vol. 6, no.2, pp.121-129, 2004.DOI: https://doi.org/10.1007/s10035-004-0170.