# Improvements and modifications in compressed air-drying system: A review

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**ABSTRACT**

*In recent trend, three utilities are most essential for industrial processes: water, electricity and natural gas. But, there is one utility considered nearly as important as these: compressed air. Instrument air dryers are essential part of the compressed air system in a various chemical plant. These devices and their consociate filters remove moisture and lubricant from the compressed air that supplied to sensitive instruments and controlling actuators. Such devices used for pushing down the dew point temperature of the compressed air, so the air remains dry and contamination free. The problems are related to dryer's performance and its systems which are directly affects outlet compressed air quality and may results in failure of concerned pneumatic equipment due to high concentration of moisture. To provide safe guard against such problem, it is required to improve performance of dryer and effectively remove moisture from pressurized air. The main objective of this paper is to review briefly the performance of refrigeration air dryer by maintaining inlet temperature of compressed air utilizing outlet air heat. In addition to this, use of additional moisture separator in dryer outlet can helps to remove water particles effectively and can maintain dew point temperature at desired limit.*

**Keyword:** *Air Dryer, Refrigeration Dryer, Desiccant Dryer, Moisture removal, Heat Exchanger, Condenser, Refrigerants*

## 1. INTRODUCTION

Compressed air dryers are special types of systems that are specifically designed to remove the moisture that is inherent in compressed air[1]–[6]. Due to the process of compression, the temperature of air is significantly increases and concentrates atmospheric contaminants, primarily water vapor. As a result, the compressed air is generally at an elevated temperature and 100% relative humidity. As the compressed air gets cooled, water vapor condenses into the storage vessels, pipes, hoses and components that are downstream from the compressor. Water vapor is removed from compressed air to prevent moisture from interfering in sensitive industrial processes. Excessive liquid and condensing water (moisture) in the air stream can be extremely damaging to pneumatically operated equipment, tools and processes that based on compressed air. The condensed water can cause corrosion in the air receivers and piping, takes away lubricating oils from pneumatic tools, get mixed with the grease used in cylinders, clump blasting media and fog painted surfaces. Consequently, it is desirable to remove condensing moisture from the air stream to prevent damage to equipment, air tools and processes. The function of removing this unwanted water is carried out by compressed air dryer. Now a day, there are various types of compressed air dryers available. These dryers are generally classified into two different categories: primary, which includes coalescing, refrigerated[2], [6], [7], and deliquescent; and secondary, which includes desiccant[8]–[10], absorption, and membrane. Their performance characteristics are typically defined by flow rate capacity in Standard Cubic Feet per Minute (SCFM) and dew point expressed as a temperature.

### 1.1 Nomenclature

| | | | |
|---|---|---|---|
| q | Heat flux (W/m2) | T | Temperature (°C) |
| G | Mass flux (kg/m2 s | Ma | Marangoni Number |
| Bo | Boiling Number | Re | Reynold's Number |
| h | local heat transfer co-efficient | D | Diameter (m) |
| x | quality | L | Length of tube (m) |

**Greek symbols**

| | | | |
|---|---|---|---|
| ρ | Density (kg/m3) | σ | Surface Tension (N/m) |
| ν | Kinematic Viscosity (m2/s) | λ | Thermal conductivity (w/m.K) |
| λ | Thermal conductivity (w/m.K) | | |

**Subscripts**

| | | | |
|---|---|---|---|
| l | Liquid | v | Vapor |
| sat | Saturation | sub | Subcooled |
| w | wall | in | Inlet condition |

## 2. RECENT STUDIES

The very first air refrigerated type air dryer was developed in 1974 by Joseph H Henderson.[1] In that research, a shell and tube type heat exchanger includes an inlet and an outlet port, cooling and demisting unit inside the shell was used (Fig-1). A refrigeration system was also used in which refrigerant is circulates into the tubes of the exchanger and extracts heat from the air. The moist air enters from the inlet port and passes through heat exchangers so that the air gets cooled and moisture gets condensed and trapped in the demisting unit which was removed from the shell through a drain valve provided at the bottom of the shell. The cooled dry air then passes through a heat exchanger and exits from the outlet port. Likewise, in 1980 Ralph O. Dowling[2] had made improvements in such a system and used separate heat exchangers the one was air to air heat exchanger and the second was air to refrigerant heat exchanger. Each of these heat exchangers made of tubular housing which contains bundles of tubes longitudinally and contains baffles directs passes around fin sheets. The refrigerant suitable for this system was R12 or R22 at -1°C by a conventional refrigeration compressor. Which takes the heat from moist air (100- 200 PSIG pressure and 30-40°C temperature) and the moisture gets condensed. The moisture, oils, and particulates get deposited on the fin and flow towards the bottom due to gravity and collected in the housing is removed through an automatic drain valve. The compressed air gets cooled up to 3°C after interacting heat with refrigerant and then gets heated up to 24°C at the outlet of the dryer.

Moreover, in 1985 Ronald G. Pridham[3] had made an apparatus for an air-drying system wherein inlet air is to be dried passes through the tube which contains pre-cooled air tube and refrigerant tube and guided by baffles to flow along the tortuous path over the tubes.

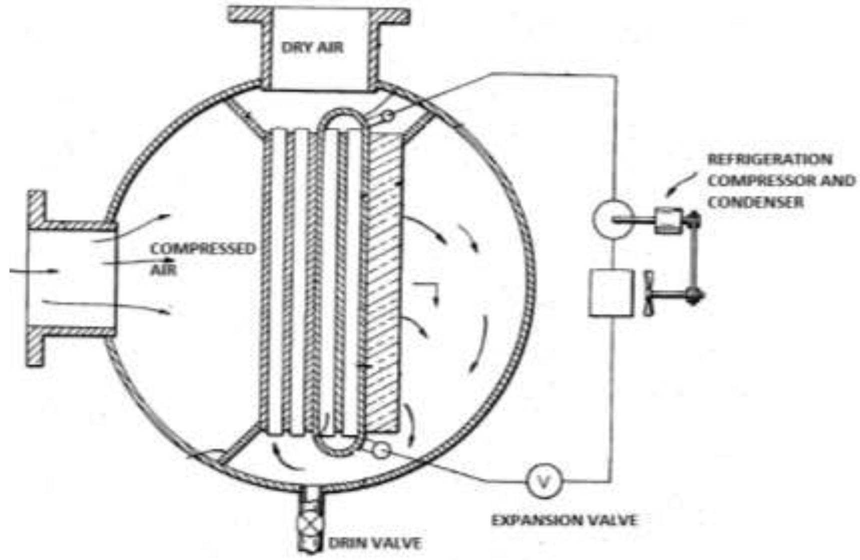


Fig-1: Shell and tube type heat exchanger for compresses air drying process [1]

These all arrangements are provided into a single tube type heat exchanger. In addition, Mantegazza et al. (1993)[4] has used a tube and fin heat exchanger which has two adjacent fluid circuits through which compressed air, refrigerant, and cooled/dry air exchanges heat with respect to each other. Also in such a system, the areas between the fins and tubes are filled with a moist material which temporarily accumulates and transfers the energy when the refrigerant is not being circulated (Fig-2). The air at the outlet of the first circuit (i.e. Refrigerant and moist air) has temperature up to 3°C so that water vapor gets condensed which is removed from the bottom of the condensate trap and is elevated in the second circuit about 20°C - 30°C.

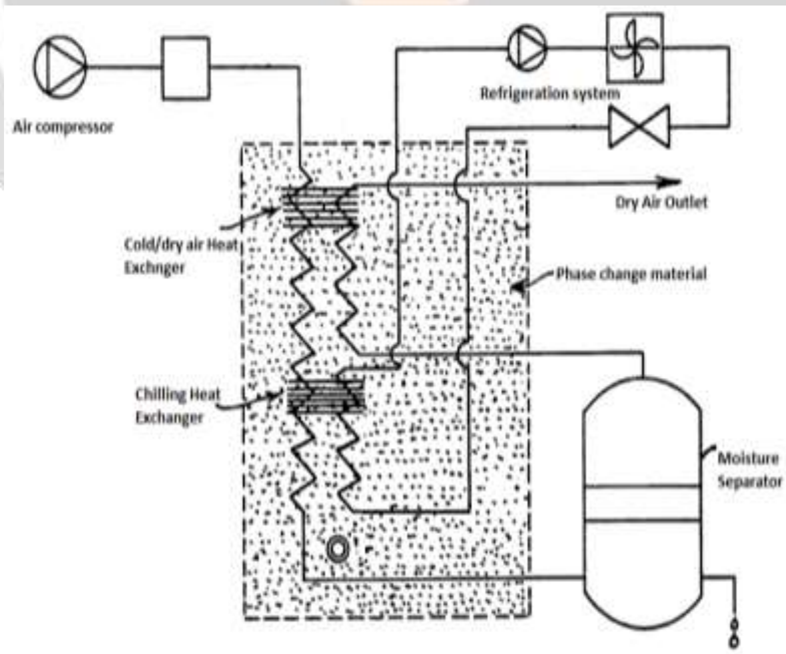


Fig-2: Tube and fin type heat exchanger with Phase change material [4]

Timothy J. Galus, et al.[5] have made some modifications to the earlier system and invented a heat exchanger which consists pre-cooler/re-heater and cooler in a single unit(Fig-3). The incoming hot and humid air enters the heat exchanger inside the pre-cooler portion where the air is precooled by dry air. Following the pre-cooler, the moist air exposed to the chillier where the compressed air further gets cooled, and the water particles present in the air get condensed and is removed through the drain point. And at the end, the dry air passes through the re-heater where it interacts heat with moist air and gains temperature up to usable range. Along with this, James W. Barnwell, et al.[6] has modified the heat exchanger. They developed air to air heat exchanger coupled with air to refrigerant heat exchanger by using a connecting tube. The air to refrigerant heat exchanger having series of stacked plates and phase change material is filled in the blank between these stack plates. The refrigeration system in this invention is designed in such a manner that the compressor gets turned off during low load or no-load condition for preventing icing on surfaces but due to frequent starting & stopping of compressor leads to wear and tear and increases maintenance cost. In solution, the phase change material is used in this system which stores the heat by using its heat-storing capacity and makes the system more reliable. The phase change material used in such a system are can maintain the cooling temperature from +4 °C to - 29°C as per requirements. Rami M. Saeed, et al.[11] had made an experimental investigation of a thermal energy storage vessel for load-shifting purposes. The new heat storage vessel invented is a plate-type heat exchanger unit with water taken as the working fluid and a phase change material as the energy storage medium. The characteristics of the heat exchanger such as heat transfer coefficient, efficiency, effectiveness, water outlet temperature, heat storage rate, total energy storage capacity and storage time were experimentally evaluated as a function of various inlet conditions. The compact parallel plate type design showed an enhanced the performance compared to conventional storage systems with an effectiveness up to 83.1% even when a phase change material of low thermal conductivity is used. The proposed phase change energy storage system not only can deliver substantial benefits as a thermal energy storage medium, but also provides cost savings in maintenance/operations, equipment, and infrastructure compared to conventional systems.

Apart from modifications in heat exchangers, Schmid, et al.[7] have installed the temperature sensor between the evaporator and the internal heat exchanger, and they are operated in an operational mode which is called a "semi flood" condition. "Semi flood" condition refers to the condition of the evaporator which instead of complete evaporation of the refrigerant in the evaporator, provides a mixture of gaseous and liquid refrigerant at the outlet which has a very low superheat. The internal heat exchanger will rise the superheat of this saturated refrigerant, thus evaporating the remaining liquid refrigerant and securing the safe operation of the compressor to which the refrigerant is directed after the internal heat exchanger as a liquid refrigerant at the inlet of the compressor can cause severe damage of compressor.

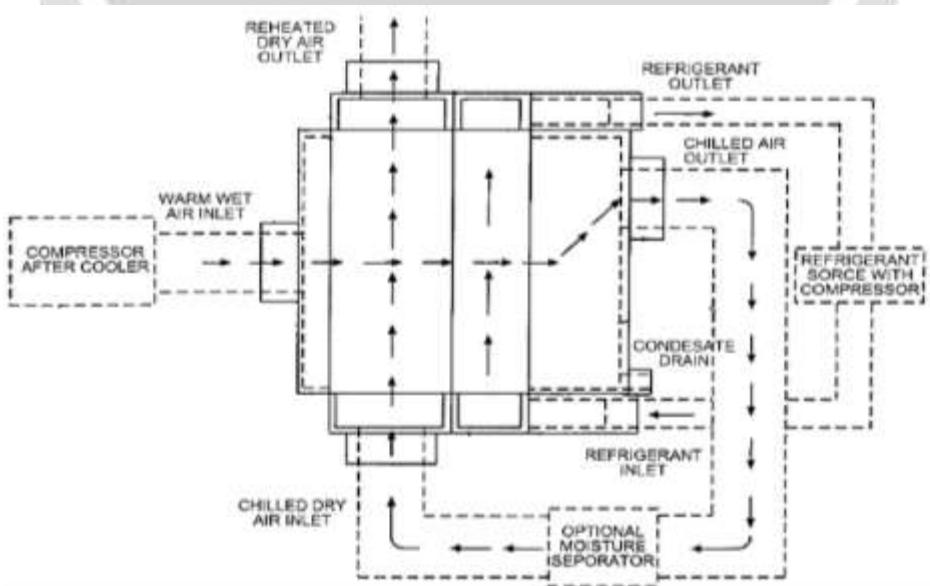


Fig-3: Pre-cooler/Re-heater/chiller type compact arrangement [5]

The refrigerants used in refrigeration systems are also plays important role in refrigeration type air dryers. Chao Dang, Li Jia, et al.[12] had made an experiment for finding out the flow boiling properties of pure refrigerant R134a and zeotropic mixture R407C are experimentally in a single visualized rectangular micro-channel heated on three sides with the cross-sectional area of 1 mm x 1 mm and length of 106 mm. Boiling heat transfer coefficients are attained at the saturation temperature of 21 °C under the heat flux and mass flux ranging from 30–150 kW/m2 and 35–1400 kg/m2 s, respectively. The boiling curves of the two different refrigerants (i.e. R134a & R407C) are also discussed (Fig-4).

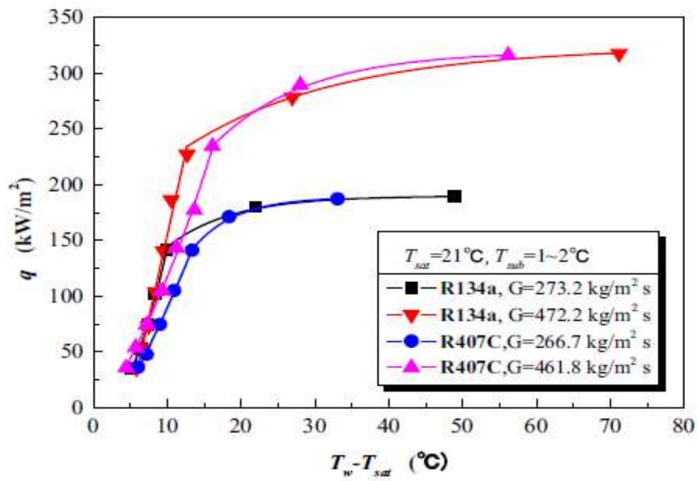


Fig-4: Boiling curves of R134a & R407C[12]

Based on the visualization results, seven flow types are determined, and the flow pattern maps are plotted. Through the comparative study, the situation of advance into churn-annular flow and coexistent bubbles with the flow patterns from confined bubble to annular are noticed for R407C. The boiling heat transfer coefficient of R407C was a bit higher than that for R134a at lower vapor quality while the opposite situation appears with the increasing vapor quality after that. During churn-annular to annular flow stage, the boiling heat transfer coefficient of R407C presents a downslide trend which is normally different from the relatively stable value of R134a. The nucleate boiling heat transfer of R407C is quelled during bubbly flow but encouraged in confined bubble to slug flow stage compared with R134a. The CHF of R407C is more than that of R134a. A relation for the flow boiling heat transfer coefficient of mixtures are proposed in consideration of Ma number and predicts satisfactorily the database of R407C and R404A.

$$Ma = \frac{\Delta\sigma}{\rho_l v_l^2}\left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]$$

Kh.R. Dione, et al.[13] showed experimental results concluded during tests on the evaporation heat transfer inside tube of a coaxial heat exchanger for residential and industrial geothermal heat pumps (Fig-5). The experimental device and instruments of the test section using evaporation inside a water heated double tube, have been described. The results obtained for evaporation local heat transfer of refrigerants R134a and R407C have been analyzed to the existing correlations. The aim of that comparison was to obtain the validation of the well-known correlations to calculate evaporation heat transfer in the geothermal operating conditions. A complete experimental analysis of the local evaporation of the R134a and R407C was carried out as a function of vapor quality inside a smooth horizontal tube. Various refrigerant mass fluxes ware tested. Additionally, new correlations for local heat transfer and pressure drop considerations have been found and validated based on the measurements.

$$\text{Ratio R} = \frac{h_x D_{in}}{\lambda_l} Re_l^{-0.857} Bo^{-0.87}$$

Most of the conventional refrigeration dryer systems are working based on evaporator temperature and accordingly compressor gets loaded/unloaded, but the disadvantage of such systems is the evaporator temperature can become too low that can occur freezing in evaporator. A method for cool drying gas has been developed by DE HERDT, et al.[14], whereby the cool dryer is characterized by curves (Fig-6b) that show the setpoint for the evaporate or temperature/evaporator pressure for a load (C) as a function of the lowest gas temperature (LAT). The method comprises the following steps: - the determination of a curve and saturated temperature or saturated pressure as a function of the load that is required to cool the gas to the lowest gas temperature: - the control of a supply of coolant from the compressor (6) to an injection point (P) downstream from the expansion means (8) and upstream from the compressor (6) to make the evaporator temperature or evaporator pressure equal to saturated temperature or saturated pressure.

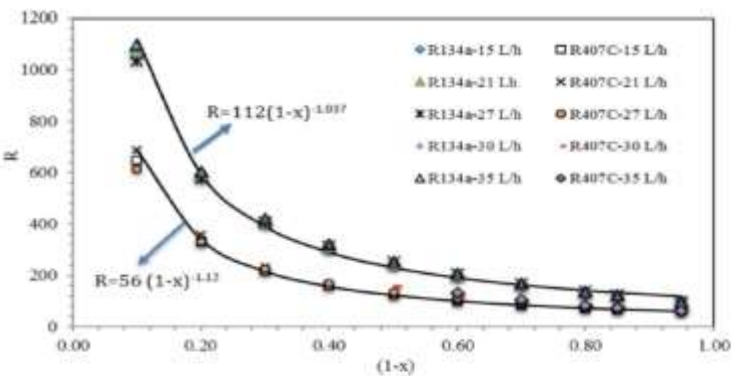


Fig-5: R134a and R407C evaporation heat transfer coefficient: Ratio R versus liquid quality[13]

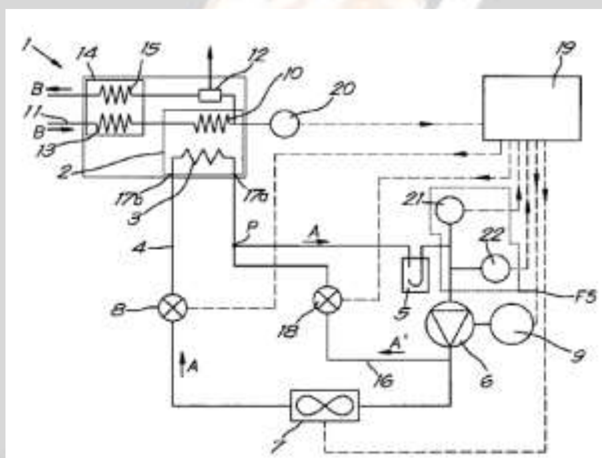


Fig-6a: method for cool drying gas[14]

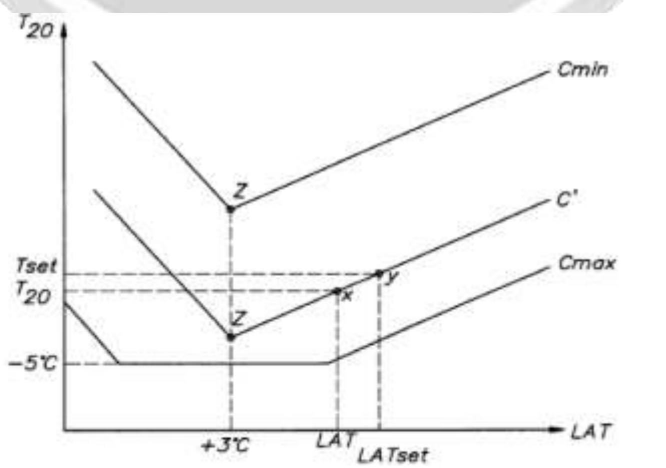


Fig-6b: Saturation Temperature v/s lowest gas temperature [14]

For a multi stage refrigeration air dryer operation into which evaporators are connected in parallel connection, Frits Cornelis A. Baltus[15] has developed a method for controlling a cooling circuit. A cooling circuit is equipped with a coolant, a compressor, a condenser and evaporator expansion valve combinations, where the outlets of the evaporators are connected to a collection pipe connected to the compressor. The cooling circuit contains a control unit connected to a temperature sensor and a pressure sensor affixed in the collection pipe and connected to the expansion valves for the control of them. The control unit is provided with an algorithm for controlling the expansion valves based on the temperature sensor and pressure sensor to control the superheating in the collection pipe. In such system, each evaporator has its separate expansion valve which operates by control system according to pressure and temperature sensor which protects the compressor and works according to load.

Aside from the Refrigeration air dryers, desiccant type air dryers are the second largely using air dryer. Such dryers work based on adsorption process and can reduce air dew point up to -40°C. J C Atuonwu, et al.[8] has established a connection between desiccant dryer energy performance and controllability using energy balance and process resiliency analysis. It is observed that using the process gain matrix, the dryer energy efficiency can be efficiently calculated with conditions for simultaneous controllability improvement. By integrating a drying rate modifying system such as a desiccant dehumidifier as an addition, these conditions are observed to be achievable due to the extra dehumidification which can be controlled using the additional degrees of freedom introduced by the adsorption system. Due to the adsorbent regulation properties which are raised by high-temperature regeneration, the resilience of energy performance to disturbances is significantly improved compared to conventional dryers. They prepared graphs of Energy efficiency (%) v/s Regeneration inlet temperature (°C) v/s MRI (Morari Resiliency Index) (Fig-7) and Percentage change in energy consumption (%) v/s Ambient temperature (°C) (Fig-8). Likewise, a desiccant dryer system performance is stipulated by the ARNEE (adsorber–regenerator net energy efficiency) and it indicates that the energy efficiency improvement is possible only if the ARNEE is greater than the energy efficiency of the single dryer.

In the same way James C Atuonwu, et al.[9] also had evaluated the energy efficiency of dehumidification drying in relation to conventional convective drying techniques. Mathematical models ware developed using which the energy efficiencies of different dehumidification type dryers are expressed in terms of that of a conventional convective dryer operating at the same temperature and prepared the graphs of dryer efficiency (%) with respect to drying air inlet temperature(°C)(Fig-9). This allows the separation of important design and operational parameters specific to each dryer type which when optimized, improve energy efficiency for the same product quality requirement and ensures better product quality for the same efficiency as a conventional dryer. Desiccant dehumidification systems have the benefit of providing further opportunities for beneficial heat integration.

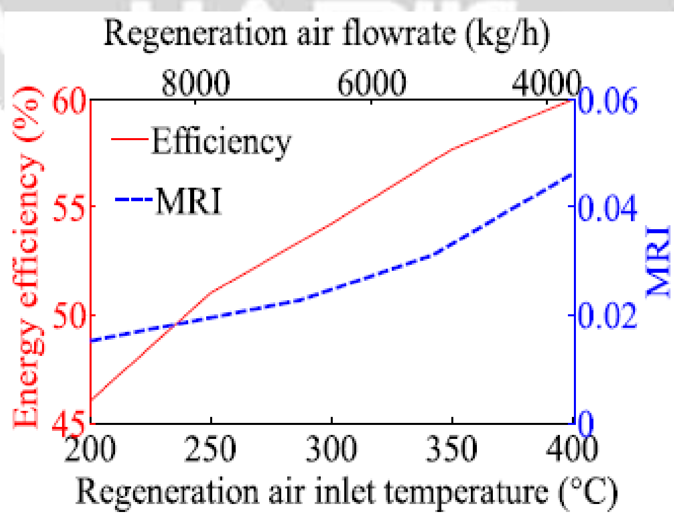

Fig-7: Variation of energy efficiency (%) and MRI with additional degrees of freedom[8]

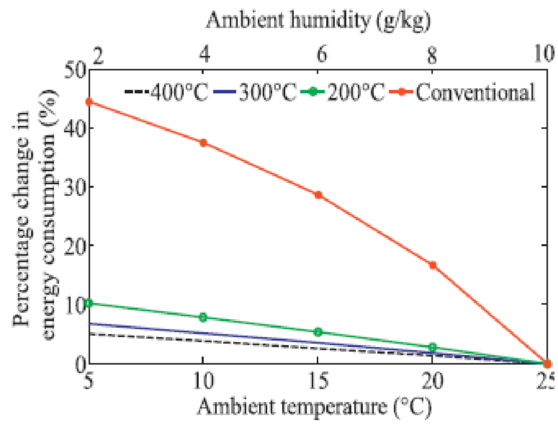


Fig-8: Percentage change in energy consumption (%) v/s Ambient temperature (°C)[8]

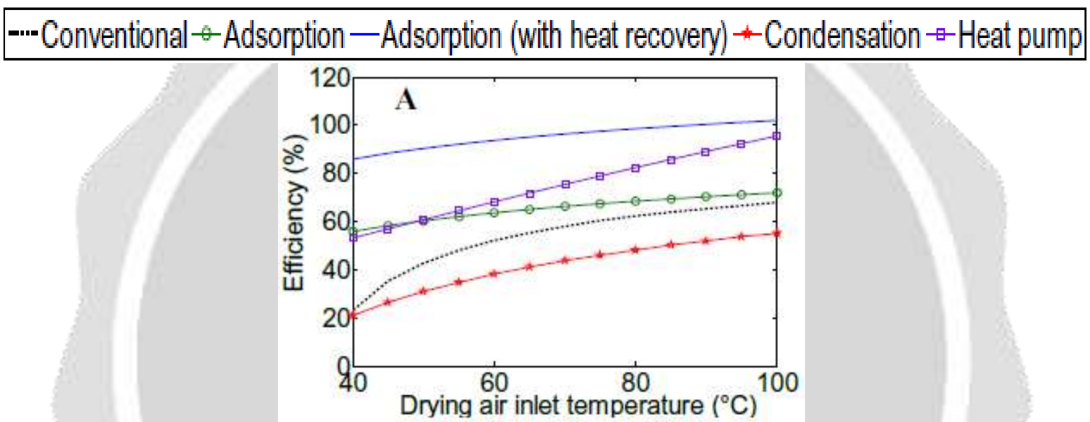


Fig-9: Energy efficiency for different dryer types[9]

S. Sundararaj, et al.[10] had carried out numerical analysis of three-dimensional flow behavior in air dryer without diffuser and with diffuser is carried out using a standard Computational Fluid Dynamics (CFD) solver. The standard k-e turbulent model is used in the analysis to anticipate the flow behavior. The air dryer is investigated with diffusers including conical, annular, and curved wall configurations to analyses the distance from which desiccant contacts with air for better air drying (Fig-10). The operating conditions of air considered for study are 0.0298 kg/s and 7 bar (gauge). The size of dryer used for study is length 1150 mm, inlet diameter 20 mm, outlet diameter 26.4 mm, tower outer diameter 100 mm, and inner diameter 93.6 mm. The diffusers modelled and analyzed are conical, annular and curved wall shape inlet diameter 20 mm, exit diameter 26.4 mm, and length 29 mm. The results showed that air dryer with conical diffuser has the low velocity standard deviation compared with annular and curved wall arrangements. Further analysis on conical diffuser with divergence angles from 9° to 47° in steps of 2° is carried and found that diffuser with 31° divergence angle shows minimum velocity standard deviation. The results disclose that the contact of air with desiccant starts at 250 mm from the inlet and the desiccant can be provided in the dryer from that location onwards.

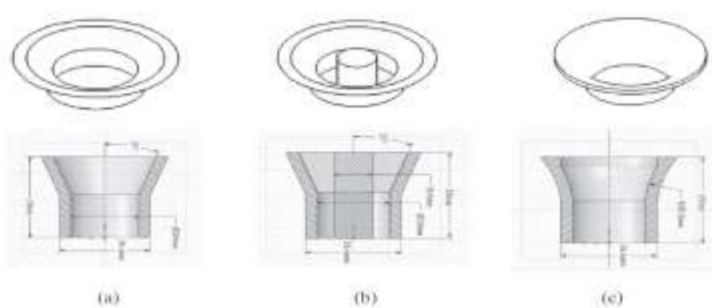


Fig-10: (a) Conical diffuser, (b) Annular diffuser, (c) Curved wall diffuser[10]

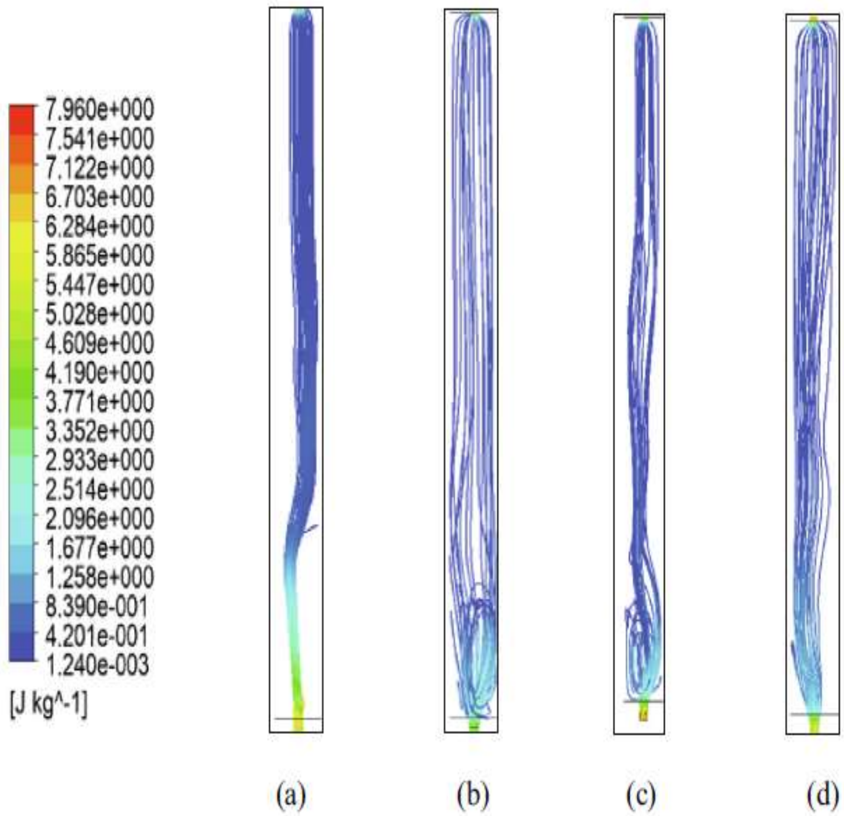


Fig-11: Turbulent kinetic energy profile of air dryer (a) without diffuser, (b) conical diffuser, (c) annular diffuser, (d) curved diffuser [10]

And concluded that:- (1) The conical diffuser has low velocity standard deviation nearly 250 mm in comparison with annular and curved wall diffusers. (2) The conical diffuser with 31°C divergences angle provides better static pressure recovery coefficient. (3) The velocity profile elicits that for the conical diffuser with 31°C divergences angle, the minimum velocity of air reaches at 250 mm from the inlet of dryer and suggested to maintain the desiccant from 250 mm distance till the exit of the dryer for effective usage of dryer. (4) The turbulent kinetic energy various broadly from inlet to the 1/3rd height of the dryer for the case of without diffuser and 1/5th height of the dryer with diffuser.

T. Kudra[16] had compared energy performance of several industrial dryers quantified in terms of the specific energy consumption and compared to the results acquired from the Baker and McKenzie's adiabatic type dryer model for convective heat transfer dryers. Examples of performance assessment are specified for indirectly heated spouted bed dryer with inert particles and spray dryer with integrated fluidized bed. Since the energy performance determination is based on temperature and humidity of the ambient and exhaust air, the calculation method is also given for gas-fired direct dryers represented by a natural-gas heated pneumatic dryer where combustion air and generated water vapor must be considered for and concluded that, The Baker and McKenzie's (B-McK) adiabatic type dryer model produced for single stage indirect spray dryer can successfully be used for energy performance of various convective dryers in both single-stage and multi-stage configurations where drying air passes through each stage and the B-McK model allows detection of faulty operation in the dryer such as clogged nozzle in the multi-nozzle system as the specific energy consumption increases drastically.

Along with air dryers, the auto moisture drain valves also plays a significant role in the air-drying systems. The function of this valves is to remove condensed water from the system. These valves work either on timer based or using mechanical system. The very first auto moisture drain valve is invented by Burton S. Aikman[17]. His invention relates to automatic drain Valve devices and more particularly to the type employed with the reservoir or receiver of a fluid pressure system to drain moisture which accumulates in the reservoir or receiver. Fluid under pressure supplied to a reservoir or receiver by a fluid compressor is usually cooled while being stored therein, as a result a substantial amount of its vapor content condenses in the reservoir. If this condensate can accumulate in the reservoir, it not only reduces the fluid pressure volume of reservoir, but may find its way into a system utilizing fluid under pressure therefrom and cause operational failure or damage to the equipment of the system. The automatic drain valve is an object or device which automatically prevents accumulation of condensate/moisture in the system. In such type of valve, it contains a chamber of certain volume and the valve is operates automatically when the chamber gets filled at certain level by condensate.

In the field of energy conservation of air dryers, there are so many analysis and researches are having been carried out. Bianca Pokorny[18] has invented a method for recovering waste heat from desiccant type dryer system. In desiccant air dryers, the air gets dried because the resins/desiccant material adsorbs the moisture presents in the air. But, after some limit, it is require to regenerate the resin/desiccant material by supplying high temperature air (hot air) so that it can dehumidified again. Some amount of heat is being utilize in heating if air and partially released the atmosphere. So, this hot air has some heat potential of supply heat to some other required system. So for utilizing this energy recovery opportunity, CARD (Compressed Air Dryer series ) is developed by FarragTech company from GmbH which has made and tested a unit made of combination of compressor and heat exchanger and concluded that that system can save energy efficiently and along with that it can reduce the overall cost of air dryer operation. This system can utilize 70% of energy present in the exhaust hot air.

In HVAC systems which are works on principle of refrigeration cycle (VCRS), the heat dissipated at the outlet of the condenser is having some potential and gives opportunity of energy conservation. This air can be used for supplying heat to another system. The recovered heat can also be utilizing in heating, drying purposes. Mohamad Ramadan, et al.[19] has developed an in-house code, simulated it into different condition and validated experimentally through different conditions. The results of analysis showed the accuracy of numerical and experimental analysis which are found almost same. Moreover, this experiment claims that the HVAC system of 1 TR has potential to complete the drying needs of 4 people if the system operates for 6 hours. This analysis is amplified by performing environmental & economical studies and concluded that, this system can save 228.8$/year and reduce the emission CO2 by 1249.6 kg. That shows that the system can be used for energy conservation as well as it is helpful in economic sector.

Moreover, K.A. Manske, et al.[20] has optimized an existing industrial system of large cold storage of Wisconsin for energy conservation. The system considered in research work is consist of single screw and reciprocating compressor, a condenser and direct expansion (dx) evaporator. A mathematical model of that system was developed and was legalized using experimental data recorded from the system. this model was used as a tool for finding out the alternative design for system and to achieve optimum performance of the system. The research was focused on the condenser sizing and pressure control. The head pressure is linear function of wet bulb temperature is minimizing the energy cost. A strategy is provided for implementing the optimum control. Along with that, the performance of the system was observed and concluded that, the energy consumption is reduced by 11% because of recommended changes.

In a modern era, researchers are focusing towers renewable energy sources to overcome the crises of crude oil. Elias M. Salilih, et al.[21] has made research on Photovoltaic (PV) solar type refrigeration system. the main objective of this research work is to observe the effect of condenser and evaporator pressure of solar refrigeration system. The performance parameters considered in that research are speed & power consumption of compressor, refrigerant mas flowrate, and capacity of cooling. The working pressure of the system at evaporator and condenser were assumed with the saturation temperatures. In that study, a sensitivity analysis was carried out by considering 2 cases. In the 1st case, the condenser saturation temperature was maintained constant at 49 ◦C and at the same time the evaporator temperature was varied from -13°C to -34°C. Similarly, In the 2nd case, the saturation temperature of the evaporator was fixed to -24◦C and the saturation temperature of the condenser was varied from 38 ◦C to 60 ◦C. In the 1st case, it was concluded that all the decided performance parameters were changes according to variation in evaporator saturation temperature. In the 2nd case, the power consumption of the compressor and compressor speed has negligible sensitivity to variation in condenser saturation temperature, but the other two performance parameters, mass flow rate of refrigerant and cooling capacity are sensitive to the condenser saturation temperature.

## 4. CONCLUSIONS

Through all the research, it is found that the various methods, modifications and attachments were developed for improve the performance, reduce operation cost and energy conservation in compressed air dryer. Earlier the researchers were more focused on improving heat exchange, surface area and finding the heat storage medium which is known as phase change material. But, in recent trends, the researchers are more fascinated about reduction in operation cost, energy consumption and use of non-conventional energy sources. Along with that various attachments are also invented to remove moisture from the system time by time like auto moisture drain valves and improved desiccant materials in case of desiccant dryers. The solar power refrigeration is also invented, and its improvement is also under progress which can reduce the operation cost up to 100% during day time. By conducting pinch analysis in the system, we can find the loop holes of the system and can utilize the energy effectively. It is possible to remove water vapor/moisture effectively by using modern auto moisture drain valve. Use of modern refrigerants can also reduce the operation cost of compressor and increase refrigeration effect in evaporator. Some specially designed nozzles and desiccants can also be used for improving efficiency if the desiccant type air dryer. There is some scope of improvement come out in field of refrigeration type air dryers as outlet air of such system has potential to absorb heat energy from other source which must be utilize in energy conservation. It is required to find best method/attachment/device to improve the performance of existing compressed air dryers.

## 5. REFERENCES

[1] J. H. Henderson, “AIR DRYING DEVICE,” 3541807, Nov. 24, 1970.

[2] Dowling Ralph O., “COMPRESSED AIR DRYER,” 4235081, Nov. 25, 1980.

[3] Pridham Ronald G., “APPARATUS FOR DRYING AIR,” 4646819, Mar. 03, 1987.

[4] Mantegazza Mario and Bellemo Luciano, “HEAT EXCHANGER,” 5228504, Jul. 20, 1993.

[5] Galus Timothy J. and Fijas David F., “PRECOOLER/CHILLER/REHEATER HEAT EXCHANGER FOR AIR DRYERS,” 6085529, Jul. 11, 2000.

[6] Branwell James W., “REFRIGERATION-TYPE DRYER APPARATUS AND METHOD,” US7134483B2, Nov. 14, 2006.

[7] Heinbokel Bernd and Schmid Urs, “Control of Refrigeration circuit with an internal Heat Exchanger,” AU2005327828B2, Aug. 24, 2005.

[8] J. C. Atuonwu, G. van Straten, H. C. van Deventer, and A. J. B. van Boxtel, “Improving dryer energy efficiency and controllability simultaneously by process modification,” Computers and Chemical Engineering, vol. 59, pp. 138–144, Dec. 2013, doi: 10.1016/j.compchemeng.2013.06.006.

[9] J. C. Atuonwu, X. Jin, G. van Straten, H. C. van Deventer Antonius, and J. B. van Boxtel, “Reducing energy consumption in food drying: Opportunities in desiccant adsorption and other dehumidification strategies,” Procedia Food Science, vol. 1, pp. 1799–1805, 2011, doi: 10.1016/j.profoo.2011.09.264.

[10] S. Sundararaj, B. A. Abraham, and P. Krishnakumar, “Computational fluid dynamics analysis of flow in diffuser of a desiccant type air dryer,” in Materials Today: Proceedings, 2020, vol. 37, no. Part 2, pp. 1517–1523, doi: 10.1016/j.matpr.2020.07.120.

[11] R. M. Saeed, J. P. Schlegel, R. Sawafta, and V. Kalra, “Plate type heat exchanger for thermal energy storage and load shifting using phase change material,” Energy Conversion and Management, vol. 181, pp. 120–132, Feb. 2019, doi: 10.1016/j.enconman.2018.12.013.

[12] C. Dang, L. Jia, M. Xu, Q. Huang, and Q. Peng, “Experimental study on flow boiling characteristics of pure refrigerant (R134a) and zeotropic mixture (R407C) in a rectangular micro-channel,” International Journal of Heat and Mass Transfer, vol. 104, pp. 351–361, Jan. 2017, doi: 10.1016/j.ijheatmasstransfer.2016.08.067.

[13] K. R. Dione, H. Louahlia, M. Marion, and J. L. Berçaits, “Evaporation heat transfer and pressure drop for geothermal heat pumps working with refrigerants R134a and R407C,” International Communications in Heat and Mass Transfer, vol. 93, pp. 1–10, Apr. 2018, doi: 10.1016/j.icheatmasstransfer.2018.02.008.

[14] de Herdt Johan Hendrik R., Baltus Frits Cornelis A., KOOYMAN Maarten, and Roelants Frank Jacques E., “METHOD AND DEVICE FOR COOL-DRYING A GAS WITH CIRCULATING COOLING LIQUID WITH BYPASS LINE,” WO2015/168755AI, Nov. 12, 2015.

[15] Baltus Frits Cornelis A., “COOLING CIRCUIT, COLD DRYING INSTALLATION AND METHOD FOR CONTROLLING A COOLING CIRCUIT,” US10060663B2, Aug. 28, 2018.

[16] T. Kudra, “Energy Performance of Convective Dryers,” Varennes, 2012.

[17] AIKMAN B. S., “AUTOMATIC DRAIN VALVE,” 2508132, May 16, 1950.

[18] B. Pokorny, “Saving energy made easy: 90% utilization thanks to threefold combination of compressed air dryer, compressor and heat exchanger,” Reinforced Plastics, vol. 63, no. 1, pp. 46–49, Jan. 2019, doi: 10.1016/j.repl.2017.11.019.

[19] M. Ramadan, R. Murr, M. Khaled, and A. G. Olabi, “Air dryer using waste heat of HVAC systems – Code development and experimental validation,” Applied Thermal Engineering, vol. 147, pp. 302–311, Jan. 2019, doi: 10.1016/j.applthermaleng.2018.10.087.

[20] K. A. Manske, D. T. Reindl, and S. A. Klein, “Evaporative condenser control in industrial refrigeration systems,” 2001. [Online]. Available: www.elsevier.com/locate/ijrefrig.

[21] E. M. Salilih, Y. T. Birhane, and N. H. Abu-Hamdeh, “Performance prediction of a solar refrigeration system under various operating pressure of evaporator and condenser,” Solar Energy, vol. 209, pp. 485–492, Oct. 2020, doi: 10.1016/j.solener.2020.09.033.